2011

Investigation of Thermal Integration in a Coal-Fired Power Plant with MEA Post-Combustion Carbon Capture

Erony Whyte Martin
Lehigh University

Investigation of Thermal Integration in a Coal-Fired Power Plant with MEA Post-Combustion Carbon Capture

by

Erony Whyte Martin

A Thesis

Presented to the Graduate and Research Committee

of Lehigh University

in Candidacy for the Degree of

Master of Science

in

Mechanical Engineering

Lehigh University

May 2011

Thesis is accepted and approved in partial fulfillment of the requirements for the Master of Science in Mechanical Engineering.

Investigation of Thermal Integration in a Coal-Fired Power Plant with MEA Post-Combustion Carbon Capture

Erony Whyte Martin

____
Date Approved

____
Dr. Edward K. Levy
Advisor

____
Dr. D. Gary Harlow
Department Chairperson

Acknowledgements

I would like to thank my advisor, Dr. Edward K. Levy, for providing the opportunity to do research at the Energy Research Center. I would also like to thank Ursla and Jodie for their incredible guidance and support during my time there. In addition, I thank Dr. Ian Laurenzi, and my fellow graduate students, Elaine Aiken, Josh Charles and Gordon Jonas, for their assistance with *Aspen Plus* and other challenges. Thank you to all of my family and friends, especially my husband, Jeff, for supporting me during my time at Lehigh University.

Table of Contents

List of Tables

List of Figures

Abstract

One option for capturing CO_2 from pulverized coal power plants is to use a MEA scrubber, with the captured CO_2 then being compressed to high pressure. Between them, the capture and compression processes will result in approximately 33% less net unit power. The compression process generates heat which can be recycled within the plant to reduce the energy penalty. This report describes the effect of compressor selection and thermal integration on heat rate. The power plant, MEA scrubber and compressors were modeled with *Aspen Plus* software and Ramgen, an inline and two integrally geared compressors and five thermal integration cases were evaluated.

1.0 Introduction

Climate change has become a growing concern. CO_2 emissions, believed to be one of the primary causes of climate change, are rising around the world. It is likely that CO_2 emissions will be regulated to reduce the effects of climate change. If CO_2 production does not decrease, the best method for controlling emissions is to capture the CO_2 before it is released to the atmosphere. Once it is captured, the CO_2 will most likely be compressed and stored underground in geological formations.

The following study examines a pulverized coal, supercritical steam cycle power plant using a sub-bituminous coal. There are several options for capturing the CO_2 emissions. This analysis examines post-combustion carbon capture using an amine-based scrubber. The chemical monoethanolamine (MEA) selectively absorbs CO_2 out of the flue gas. However, the MEA requires thermal energy to release the CO_2. This energy will most likely be supplied by extracting steam from the turbine cycle, which in turn will reduce the power produced. In addition, the captured CO_2 exiting the MEA scrubber will be compressed to high pressures, for which the compressor will also require power. Overall, the energy penalty for the MEA system and the compressor is predicted to be approximately one third of the net power.

Plant owners will be seeking technologies and techniques to reduce the energy penalty of carbon capture. One option is to utilize the heat generated in the MEA scrubber and compressor, which would otherwise be rejected as waste heat. Use of this heat within the power plant would result in improved power plant efficiency.

This thesis describes the use of the software program *Aspen Plus* to model the various components of a pulverized coal, supercritical power plant with MEA carbon capture. The predicted performances of various components in the power plant are compared to published results to demonstrate the reliability of the model. Then, four compressor options were modeled. Finally, thermal integration cases using each compressor and the MEA scrubber are modeled and the predicted benefits of each thermal integration case are compared.

2.0 Aspen Plus Modeling

The process modeling software, *Aspen Plus*, was used to predict the effects of carbon capture and heat integration on plant performance. The models were not designed for a specific plant so the results may be applied to multiple plants. The generic plant concept is illustrated in Figure 1. A sub-bituminous coal was used in the boiler system for all of the analyses. In addition, the turbine cycle used supercritical steam. The boiler system, turbine cycle, CO_2 capture system, CO_2 compressor, and the corresponding *Aspen Plus* models are explained below. The *Aspen Plus* model results are compared to first principle calculations and CO_2 property values from *Aspen Plus* are compared to values from the National Institute of Standards and Technology's Reference Thermodynamic Properties (REFPROP).

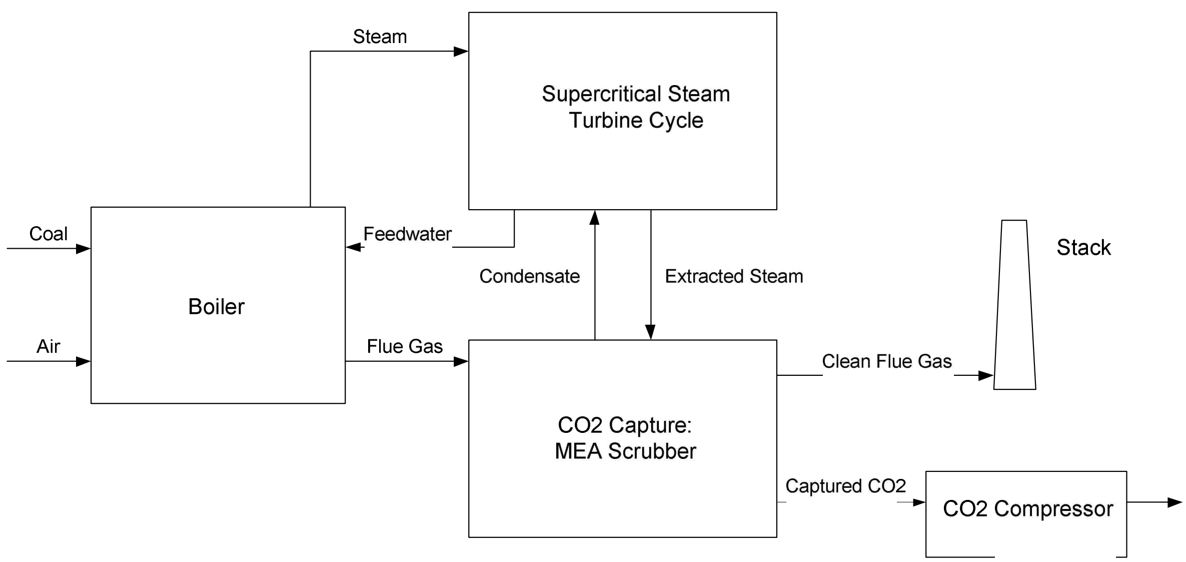


Figure 1. Conceptual diagram of power plant.

2.1 Steam Turbine Cycle

A supercritical steam turbine cycle was modeled from the vendor steam turbine kit shown in Appendix A. The term supercritical refers to the fact that the water is at pressures above its critical pressure (1070 psia) in parts of the steam cycle. The turbine cycle essentially consists of turbines, a condenser, and feedwater heaters along with other accessory equipment (Figure 2).

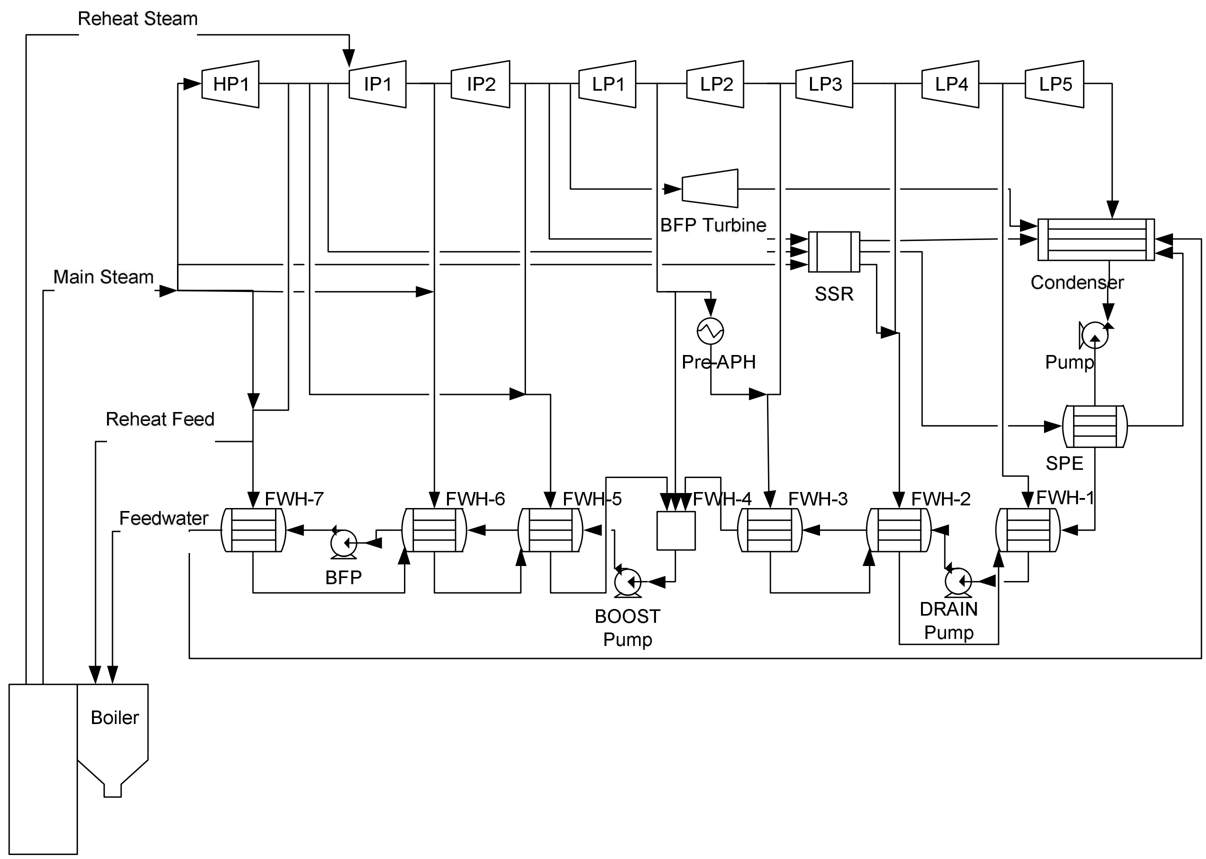


Figure 2. Supercritical turbine cycle layout.

Water is heated in the boiler to form high pressure steam, which is then expanded in the high pressure (HP) turbine in one stage. Then the steam is re-heated in the boiler before being expanded in the intermediate pressure (IP) turbine in two stages followed by the low pressure (LP) turbine in five stages. After the steam goes through the low pressure turbine, it is condensed and routed through a series of feedwater heaters. The boiler feedwater is heated with extractions from the higher pressure turbines to “pre-heat” the water before it reaches the boiler.

2.1.1 Turbine Calculations

The inlet and outlet pressures and the flow rates are given on the turbine kit diagram in Appendix A. The isentropic efficiencies of the various turbine stages were calculated based on the difference between the actual enthalpy and the isentropic enthalpy of the steam. The actual inlet and outlet enthalpies are given on the diagram, while the isentropic enthalpy was calculated using steam tables from Cengel and Boles (2008). The calculation for the isentropic efficiency is shown in equation (1). The efficiencies of each turbine stage are shown in Table 1.

$$\eta = \frac{h_1 - h_{2a}}{h_1 - h_{2s}} * 100 \ [\%] \tag{1}$$

Table 1. Turbine stage specifications.

Turbine	Inlet Pressure	Outlet Pressure	Steam Flow Rate	Isentropic Efficiency
	[psia]	[psia]	[lb/hr]	[%]
HP1	3689.7	740	4,122,221	85.08%
IP1	666	295	3,730,357	83.54%
IP2	295	165.1	3,545,824	86.48%
LP1	165.1	87.4	3,145,724	87.50%
LP2	87.4	24.9	2,797,689	89.69%
LP3	24.9	11.96	2,688,685	89.87%
LP4	11.96	4.68	2,574,150	89.73%
LP5	4.68	0.61	2,395,203	67.45%
BFP	160.1	0.61	233,214	79.98%

The amount of work performed by each turbine stage was calculated according to equation (2). The constant is a conversion factor to convert the units of the equation from $\frac{Btu}{hr}$ to kW.

$$W_{turbine} = \eta * \dot{m}_{steam} * (h_{2s} - h_1) * 2.931 * 10^{-4} \ [kW] \tag{2}$$

Table 2 shows the total work produced by each turbine stage at design conditions. The calculated results and the Aspen Plus results are compared to demonstrate the reliability of the Aspen Plus model. The table shows that the Aspen Plus model is reasonably accurate in predicting the actual turbine work produced. The booster feed pump (BFP) turbine is not included in the total power because it is dedicated to driving the booster feed pump.

Table 2. Turbine stage work.

Turbine	Work Out Calculated	Work Out Aspen	Diff.
	[kW]	[kW]	[%]
HP	183,881	183,314	-0.31%
IP1	105,981	106,167	0.18%
IP2	65,783	65,889	0.16%
LP1	57,438	57,586	0.26%
LP2	84,865	85,016	0.18%
LP3	39,164	39,200	0.09%
LP4	42,852	42,927	0.17%
LP5	55,738	55,798	0.11%
Total Power	635,703	635,897	0.03%
BFP	22,542	22,544	0.01%

2.1.2 Condenser and Feedwater Heaters

The turbine cycle includes one steam condenser at the end of the last LP stage, and seven feedwater heaters (FWHs). The steam condenser is modeled as a condenser at the same pressure as the exit of the last LP stage turbine. Then its pressure is increased to 88 psia in a pump to complete the cycle.

On its way back to the boiler to become steam, the feedwater is pre-heated through a series of FWHs. This reduces the amount of coal required to make steam at the specified boiler exit temperature. Out of the seven FWHs, six are closed heat exchangers, and one is a mixing chamber. The feedwater is heated by steam extractions after each turbine stage. Although extracting steam from the turbines reduces the total power produced by all the turbines downstream of the extraction, the reduction in the amount of coal burned offsets the turbine losses and makes the plant more efficient overall. The feedwater represents the "cold side" of the heat exchanger, while the steam extraction represents the "hot side."

The concept of the flow in a typical closed FWH heat exchanger is shown in Figure 3. The "cool steam extraction" from the following heat exchanger (bold line) is recycled to the heat exchanger preceding it to continue heating the feedwater. However, in FWH-1, which does not have a preceding heat exchanger, the "cool steam extraction" is combined with the "hot feedwater" flowing into FWH-2. Also, FWH-4 is unique in that it is an "open" heater in which the feedwater and the steam are mixed together in a chamber. The temperatures of the feedwater leaving each heater are shown in Table 3. However, the temperature and pressure leaving FWH-7 are the most important values since the enthalpy of this stream will determine how much additional heat must be provided by coal combustion in the boiler.

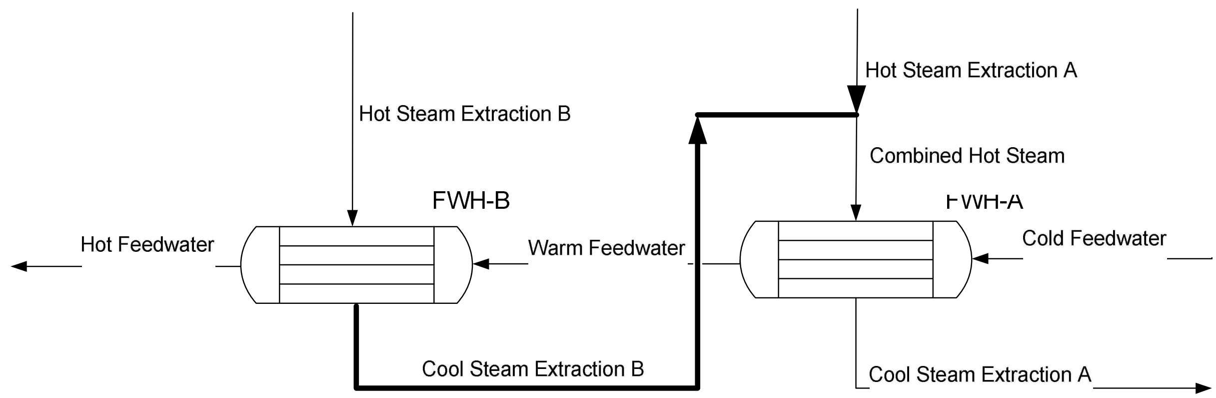


Figure 3. Diagram of feedwater and extraction steam flow in closed feedwater heaters.

Table 3. Feedwater heater specifications.

	Turbine Cycle Design Feedwater Outlet Temp	Aspen Plus Feedwater Outlet Temp	Diff.
	[F]	[F]	[%]
FWH-1	151.9	151.9	0.00%
FWH-2	193.8	193.8	0.00%
FWH-3	231.4	231.4	0.00%
FWH-5	363.6	363.6	0.00%
FWH-6	414.2	414.2	0.00%
FWH-7	506.9	506.9	0.00%

Aspen Plus calculates the duty of the FWHs based on input given by the user. An *Aspen Plus* heat exchanger model requires the input of one of the following variables:

- Heat exchanger area
- Heat exchanger duty
- Outlet temperature of the hot or cold stream
- Temperature approach at either end of the exchanger
- Degrees of superheating or subcooling for the hot or cold stream
- Vapor fraction of the hot or cold stream
- Temperature change of the hot or cold stream

In this analysis, the FWHs were given a cold stream outlet temperature (Table 3). The feedwater outlet temperature and the heat exchanger duty were most important for modeling the turbine kit. The overall heat transfer coefficient (U) was assumed constant with a value of 790 $\frac{Btu}{hr*sqft*R}$. A constant value was selected because the only known specifications of the heat exchangers are the inlet and outlet stream temperatures. Typical values range from 200 to 1050 $\frac{Btu}{hr*sqft*R}$ for steam condensers that use water on the tube-side (Incropera 2007). The assumed value in this analysis is an estimated average of the typical values and is the base value recommended by *Aspen Plus* (Incropera 2007).

In *Aspen Plus,* the "Shortcut" calculation method was selected, which determines the duty of a FWH based on three known temperatures (hot inlet, cold inlet and given cold outlet) and then determines the fourth temperature (hot outlet). Since each FWH is connected to other FWHs, *Aspen Plus* must iterate all of the FWHs until all given cold outlet temperatures are achieved. Without *Aspen Plus,* the FWH duty can be estimated by either equation (3) or (4). On the cold side, the feedwater remains a liquid through the heat exchanger. On the hot side, the extraction steam continuously releases heat until it becomes a saturated liquid. Therefore, the heat of vaporization must be included to account for the isothermal phase change (equation 4). The calculated duty of each closed FWH is compared to the *Aspen Plus* model (Table 4). The results demonstrate that the Aspen model is reasonably accurate.

$$Q = \dot{m}_{feedwater} * C_{p,feedwater} * \Delta T_{\text{feedwater}} = \dot{m}_{feedwater} * (h_{out} - h_{in}) \quad \left[\frac{Btu}{hr}\right] \qquad (3)$$

$$Q = \dot{m}_{ext.steam} * \left[\left(C_{p,ext.steam} * \Delta \mathrm{T}_{\text{ext.steam}}\right) + \left(h_{fg\,@\,P_{steam\,out}}\right)\right] \left[\frac{Btu}{hr}\right] \qquad (4)$$

Table 4. Feedwater heater performance.

	m_{cold}	$h_{in, cold}$	$h_{out, cold}$	Duty – Calculated	Duty - Aspen	Diff.
	[lb/hr]	[Btu/lb]	[Btu/lb]	[Btu/hr]	[Btu/hr]	[%]
FWH-1	2,675,463	55.356	120.05	173,086,403	173,332,835	0.14%
FWH-2	3,175,031	120.05	162.02	133,256,051	130,636,468	-1.97%
FWH-3	3,175,031	162.02	199.88	120,206,674	120,270,962	0.05%
FWH-5	4,226,581	285.11	336.37	216,654,542	216,278,387	-0.17%
FWH-6	4,226,581	336.37	390.43	228,488,969	228,584,230	0.04%
FWH-7	4,226,581	408.7	495.59	367,247,623	366,241,973	-0.27%

2.1.3 Pumps

The turbine cycle also includes four (4) pumps. There is one pump downstream of the condenser, two after feedwater heaters 1 and 4, both on the "hot side" outlet stream, and a boiler feed pump after FWH 6 on the "cold side" outlet stream. The power required for the pumps was calculated using equation (6). The quantity $\dot{m}$ is the mass flow across the pump, ΔP is the change in pressure, υ is the specific volume of the steam entering the pump and η is the pump's isentropic efficiency (calculated by equation 5). The constant is a conversion factor that changes the units of the equation from $\frac{ft*lbf}{hr}$ to kW. It is assumed that this power is drawn from the gross power produced by the plant. The performance of the booster feed pump turbine is included in the analysis of the other turbines. The calculated power and the Aspen power are shown in Table 5.

$$\eta = \frac{h_1 - h_{2s}}{h_1 - h_{2a}} * 100 \ [\%] \tag{5}$$

$$W_{pump} = \frac{1}{\eta} \dot{m} * \Delta P * \upsilon * 5.4233 * 10^{-5} \ [kW] \tag{6}$$

The boiler feed pump (BFP) is powered by a turbine using steam extracted downstream of the IP turbine. The BFP turbine is only used to power the BFP and it does not produce any additional work to the turbine generator. Therefore, its work is not included in the gross turbine cycle power or in any calculations for the turbine cycle heat rate. In addition, the power required by the BFP is not subtracted from the gross power generated by the turbine cycle since it has a dedicated source.

Table 5. Compare calculated vs. Aspen Plus pump power.

	Inlet Pressure	Outlet Pressure	Isentropic Efficiency	Power		
				Calculated	Aspen	Diff.
	[psia]	[psia]	[%]	[kW]	[kW]	[%]
Condenser	0.6	88	64.34%	283.31	283.44	0.05%
Drain	4.4	86	55.00%	65.62	65.88	0.39%
Booster	82.2	375	83.33%	1276.16	1275.87	-0.02%
BFP	290.0	4615	81.72%	22541.63	22543.13	0.01%

2.1.4 Steam Turbine Cycle Performance

The performance of the steam turbine cycle is measured by the "Turbine Cycle Heat Rate" (TCHR) which is a ratio of the total heat input to the cycle to the electrical power output. The total heat input to the cycle is the sum of the heat for the main steam and the heat for the reheat steam after the HP turbine. The calculation for TCHR is shown in equation (7).

$$HR_{turbine\ cycle} = TCHR = \frac{Q_{boiler}+Q_{reheat}}{P_{gross}} \left[\frac{Btu}{kWh}\right] \qquad (7)$$

The rate of heat transfer in the boiler to the main steam and reheat steam is calculated by equation (8). The quantity $\dot{m}$ is the mass flow of steam through the boiler, h_1 is the enthalpy of the steam entering the boiler and h_2 is the enthalpy of the steam leaving the boiler.

$$Q = \dot{m}_{\ main\ steam} * (h_2 - h_1)_{main\ steam} + \dot{m}_{\ reheat} * (h_2 - h_1)_{reheat} \left[\frac{Btu}{hr}\right] \qquad (8)$$

The gross power generated by the turbine cycle is calculated by equation (9). The gross power is the sum of the work performed by the HP, IP and LP turbines multiplied by the efficiency of the turbine generator (η_{tg}). The efficiency is assumed to be 98.5% at design conditions. The calculated values for the mass and energy of the steam flows, the turbine power, and the turbine cycle heat rate are compared to the values calculated by *Aspen Plus* (Table 6).

$$P_{gross} = \eta_{tg} * \Sigma P_{out,turbines}\ [kW] \qquad (9)$$

Table 6. Supercritical Turbine Cycle Performance.

	Unit	Turbine Kit	Aspen	Diff. [%]
Boiler Steam Flow	[lbm/hr]	4,184,734	4,184,730	0.00%
Q Boiler	[Btu/hr]	3,859,580,168	3,848,000,000	-0.30%
Reheat Steam Flow	[lbm/hr]	3,677,525	3,677,530	0.00%
Q Reheat	[Btu/hr]	915,335,973	928,219,255	1.41%
Total Heat Input	[Btu/hr]	4,774,916,141	4,776,219,255	0.03%
Gross Turbine Power	[kWh]	635,021	635,768	0.12%
Generated Power	[kWh]	625,496	626,232	0.12%
TCHR	[Btu/kWh]	7,634	7,627	-0.09%

2.2 Boiler

The boiler model used in this analysis is illustrated in Figure 4. It was assumed that the coal has the same composition entering the boiler as it did when it was delivered to the plant. The coal first enters the mill to be pulverized. Then the pulverized coal flows into the boiler to produce steam for the turbine cycle and the resulting flue gas flows through the air pre-heater (APH). In the APH, the flue gas pre-heats the incoming combustion air after the air is compressed in the forced draft (FD) fan. The flue gas then flows through the ESP to remove ash, the induced draft (ID) fan, and the FGD to remove sulfur before exiting through the stack. Note that there is no carbon capture system between the FGD and the stack in this base model. A diagram of the corresponding Aspen Plus model of the boiler system can be found in Appendix B.

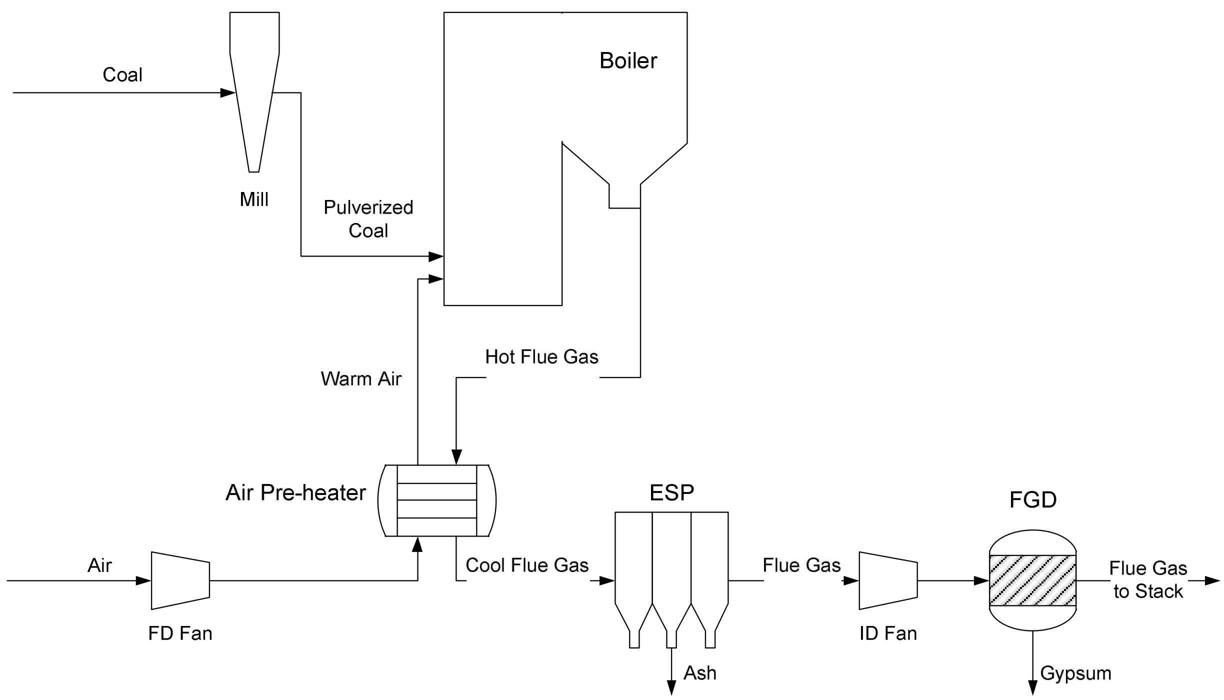


Figure 4. Diagram of boiler and accessory equipment.

2.2.1 Coal Combustion

A sub-bituminous, Powder River Basin coal was used for this analysis. Coal is composed of carbon, hydrogen, sulfur, oxygen, nitrogen, ash and moisture. Individual samples of coal are analyzed for their exact composition. The composition of the coal used in this study is shown in

Table 7. The Proximate analysis measures the relative amounts of fixed carbon, ash, volatiles and moisture, while the Ultimate analysis measures the specific amount of each component by weight. The Sulfur analysis measures the various forms of sulfur present in the coal. PRB coal has relatively high moisture content compared to other types of coal. The Higher Heating Value (HHV) is the amount of energy released from burning one pound of coal and includes the energy for the latent heat of vaporization of water.

Table 7. Composition of PRB coal.

Sub-bituminous (PRB) Coal	
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

The carbon in the coal reacts with oxygen in air to form carbon dioxide and heat. The reaction between coal and air can be calculated to predict the composition of the flue gas. First, the chemical formula of the coal is determined using the Ultimate analysis (Table 8).

Table 8. Chemical formula of PRB coal.

Ultimate Analysis: PRB Coal				
Component	As Received Weight	Molecular Weight	AR	**Molar Composition**
	[lb/100 lb coal]	[lb/mol]	[mol/ 100 lb coal]	[mol/mol C]
C	49.21	12.0107	4.0972	1.0000
H	3.51	1.0079	3.4825	0.8500
S	0.45	32.0650	0.0140	0.0034
O	11.67	15.9994	0.7294	0.1780
N	0.73	14.0067	0.0521	0.0127
H_2O	28.09	18.0152	1.5592	0.3806
Ash	6.31	----	----	----
TOTAL	99.97			

The second column shows the coal composition by weight (from a 100 pound coal sample). The molecular weight of each component, the third column, is divided by the "as received weight" to obtain the moles of the component per 100 pounds of coal. Chlorine is neglected in this analysis since it is present in a very small amount. Since carbon is the main reactant, the individual molar compositions, the fourth column, are divided by the molar concentration of carbon, which yields the fifth column. This results in a unique chemical formula for the coal sample of ($C_1H_{0.85}S_{0.0034}O_{0.178}N_{0.0127}$) + 0.3806 H_2O + Ash.

Air is composed of 20.58% oxygen, and 77.39% nitrogen by mole (and by volume) and moisture. Therefore, there are 3.76 nitrogen molecules for every oxygen molecule. In calculations, the composition of one mole of air is (O_2 + 3.76N_2). The amount of water is dependent on the temperature and humidity. In this study, a temperature of 77°F and a relative humidity (RH) of 65% were assumed as constant. The molar concentration of water in dry air (W) is calculated using equation (10).

$$W = \frac{RH * P_{sat}}{P_{atm} - (RH * P_{sat})} \left[\frac{mol}{mol}\right] \quad (10)$$

This equation determines that the moles of water at the given temperature and humidity equal 0.0207 moles H_2O per mole of dry air. This is equal to 0.0987 moles H_2O per mole of O_2 in air. Thus air at 77 F and 65% relative humidity has the following molar composition: O_2 + 3.76N_2 + 0.0987 mol H_2O.

The combustion of coal in air is calculated using Equation (11). A_{th} represents the *minimum* theoretical amount of air required for complete combustion of the carbon in the coal. Ash is the remaining solid material that does not react in the boiler and therefore, ash is not included in the reaction to form flue gas. From a chemical perspective, the ash is simply heated in the boiler.

$$(C_1H_{0.85}S_{0.0034}O_{0.178}N_{0.0127}) + 0.3806\ H_2O + A_{th}(O_2 + 3.76N_2 + 0.0987H_2O)$$
$$\rightarrow bCO_2 + cH_2O + dO_2 + eN_2 + fSO_2 \qquad (11)$$

The composition of the flue gas can be calculated by hand or with a computer program by performing a mass balance on the equation (Table 9).

Table 9. Mass balance of combustion with PRB coal and air.

	Reactants					→	Products
	Coal		Coal Moisture		Air		Flue Gas
C	1	+	0	+	0	=	b
H	0.8498	+	2*0.3806	+	2*0.0987*A_{th}	=	2c
S	0.0034	+	0	+	0	=	f
O	0.178	+	0.3806	+	2.0987*A_{th}	=	2b + c + 2d +2f
N	0.0127	+	0	+	2*3.76*A_{th}	=	2e

Shifting all of the constants to the right hand side (RHS) and the unknown values to the left hand side (LHS), results in the following equations:

$$C:\quad b = 1.0000 \qquad (12)$$
$$H:\quad -0.1974A_{th} + 2c = 1.6110 \qquad (13)$$
$$S:\quad f = 0.0034 \qquad (14)$$
$$O:\quad -2.0987A_{th} + 2b + c + d + 2f = 0.5586 \qquad (15)$$
$$N:\quad -7.52A_{th} + 2e = 0.0127 \qquad (16)$$

These equations determine the flue gas composition using the minimum theoretical amount of air (A_{th}), so there will be no oxygen in the flue gas, since all of the oxygen in the air reacted with the carbon in the coal. Therefore, the coefficient for oxygen in the products, d, is set equal to zero. This leaves 5 equations that can be solved for 5 unknowns (A_{th}, b, c, e, and f). Performing the required algebra provides the following results:

$$(C_1H_{0.85}S_{0.0034}O_{0.178}N_{0.0127}) + 0.3806\ H_2O + 1.1268(O_2 + 3.76N_2 + 0.0987H_2O)$$
$$\rightarrow CO_2 + 0.9167H_2O + 4.2440N_2 + 0.0034SO_2 \quad (17)$$

Thus, 1.1268 moles of air are required to react with 1 mole of coal. In actual coal-fired boilers, an excess amount of air is needed to ensure that combustion is complete. The excess air, in addition to adequate coal pulverization and mixing within the boiler, helps the coal and oxygen react. The amount of excess air can vary, but it is often achieved by specifying the amount of unreacted oxygen remaining in the flue gas. In this study, the amount of oxygen in the flue gas is specified at 3.5 mol%.

First the combustion equation changes to equation (18), where E represents the fractional amount of excess air used. The composition of the flue gas must satisfy equation (19) as well. Thus the six equations needed to determine the flue gas composition are listed below as equations (20) through (25).

$$(C_1H_{0.85}S_{0.0034}O_{0.178}N_{0.0127}) + 0.3806\ H_2O + 1.1268*(1+E)*(O_2 + 3.76N_2 + 0.0987H_2O)$$
$$\rightarrow bCO_2 + cH_2O + dO_2 + eN_2 + fSO_2 \quad (18)$$

$$\frac{d}{b+c+d+e+f} = 0.035 \quad (19)$$

C:		b					= 1.0000	(20)
H:	-0.2224E		+2c				= 1.8336	(21)
S:						f	= 0.0034	(22)
O:	-2.3648E	+ 2b	+ c	+ 2d		+ 2f	= 2.9234	(23)
N:	-8.4735E				+ 2e		= 8.4862	(24)
Excess O2:		-0.035b	-0.035c	+0.965d	-0.035e	-0.035f	= 0	(25)

The unknowns are found by solving the matrix, which results in equation (26). Thus, for complete combustion and 3.5% oxygen in the flue gas, 23.08% excess air is needed.

$$(C_1H_{0.85}S_{0.0034}O_{0.178}N_{0.0127}) + 0.3806\ H_2O + 1.1268*(1+0.2308)*(O_2 + 3.76N_2 + 0.0987H_2O)$$
$$\rightarrow CO_2 + 0.9451H_2O + 0.2601O_2 + 5.2226N_2 + 0.0034SO_2 \quad (26)$$

2.2.2 Energy balance

In addition to the mass balance, the boiler must also satisfy an energy balance. Following the first law of thermodynamics, the total energy entering the boiler must equal the total energy leaving the boiler (Figure 5). The energy into the boiler comes from the coal energy, the pulverizer energy and the combustion air energy. The energy leaving the boiler is in the form of steam energy, flue gas energy and energy losses through radiation. The energy required from the coal is shown in equation (27).

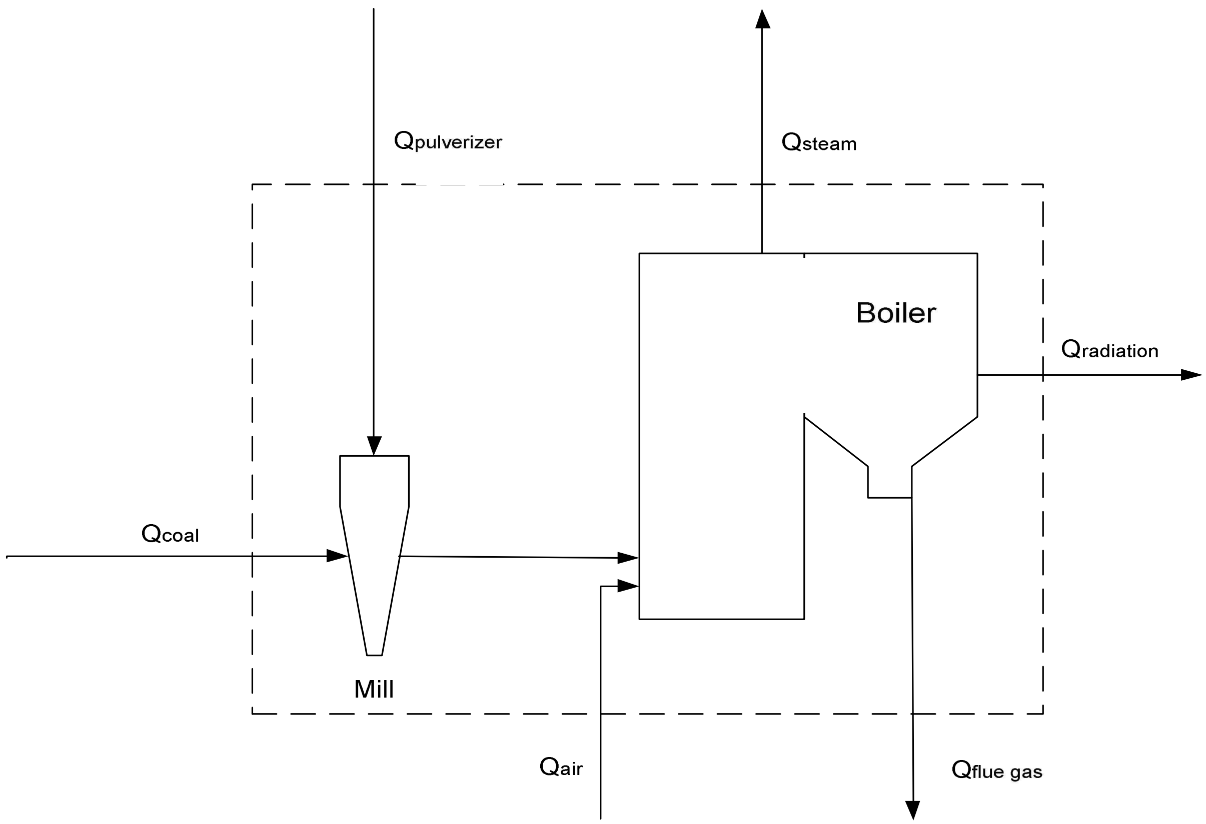


Figure 5. Boiler Control Volume and Energy.

$$Q_{coal} = Q_{radiation} + Q_{flue\ gas} + Q_{steam} - Q_{pulverizer} - Q_{air} \quad \left[\frac{Btu}{hr}\right] \tag{27}$$

The values on the right hand side of equation (27) can be calculated using equations (28) through (32). $Q_{radiation}$ is the amount of heat lost through the furnace walls, and was estimated to be 0.8% of the total coal energy. $Q_{flue\ gas}$ is the amount of energy remaining in the hot flue gas after the steam is created. Q_{steam} is the amount of energy required by the turbine cycle in the main and reheat steam flows. For this model, the total steam energy was previously calculated to be 4.775*10^9 Btu per hour (Table 6).

$Q_{pulverizer}$ is the amount of heat that is added to the coal as it is crushed in the pulverizer. The coefficient was calculated using the HEATRT code, a software program developed by the Energy Research Center (Sarunac, 1993) with the assumption that all of the power provided to the mills represented heat energy added to the coal and thus, there was no energy loss within the mill or to the environment. Finally, the energy of the air (Q_{air}) is calculated by the mass of combustion air times the enthalpy difference between inlet temperature and 77°F. The temperature of 77°F is used because the HHV of coal is measured at 77°F.

$$Q_{radiation} = 0.8\% * HHV_{coal} * \dot{m}_{coal} \quad \left[\frac{Btu}{hr}\right] \tag{28}$$

$$Q_{flue\ gas} = \dot{m}_{flue\ gas} * \left(h_{flue\ gas,out} - h_{flue\ gas,77\,°F}\right) \quad \left[\frac{Btu}{hr}\right] \tag{29}$$

$$Q_{steam} = Q_{boiler} + Q_{reheat} \quad \left[\frac{Btu}{hr}\right] \tag{30}$$

$$Q_{pulverizer} = 18.06 * \dot{m}_{coal} \quad \left[\frac{Btu}{hr}\right] \tag{31}$$

$$Q_{air} = \dot{m}_{air} * \left(h_{air,in} - h_{air,77\,°F}\right) \quad \left[\frac{Btu}{hr}\right] \tag{32}$$

The energy in the flue gas is calculated by equation (29). Via equations (11) through (26), a relationship between the amount of coal burned and the flue gas composition is known. Therefore, the mass of the flue gas and the enthalpy of the flue gas can be expressed in terms of the moles of coal burned (N_c). In equation (33), N_c represents the moles of coal, M_c represents the molecular weight of carbon (12.01 lb/lbmol), and P_c represents the mass percentage of carbon in the as-received coal.

$$\dot{m}_{coal} = \frac{N_C * M_C}{\frac{P_C}{100}} \quad \left[\frac{lb}{hr}\right] \tag{33}$$

Equations (35) through (37) show the relationship between the flue gas and combustion air mass flow rates and enthalpy proportional to the molar flow of carbon. First, the molecular weight of the flue gas ($\bar{M}$) is calculated by equation (34) which is required before calculating the enthalpy of the flue gas in equation (35). Then the mass of the flue gas is calculated in terms of the moles of coal burned in equation (36). The subscript i refers to each individual component in the flue gas mixture.

$$\bar{M}_{flue\ gas} = \Sigma(mol(\%)_i * \overline{M_i}) \quad \left[\frac{lb}{lbmol}\right] \tag{34}$$

$$h_{flue\ gas} = \Sigma\left(\frac{mol(\%)_i * h_i}{\bar{M}_{flue\ gas}}\right) \quad \left[\frac{Btu}{lb}\right] \tag{35}$$

$$\dot{m}_{flue\ gas} = N_c * \Sigma\left(\bar{M}_{flue\ gas} * \frac{mol(\%)_i}{mol_c}\right) \quad \left[\frac{lb}{hr}\right] \tag{36}$$

The energy in the combustion air can also be expressed in terms of the moles of coal burned. In the case of the example described here, the air has already gone through the APH and thus is at a temperature between 500°F and 600°F. The enthalpy of the incoming air is calculated to be 251.19 Btu/lbm or 7191.88 Btu/lbmol, using the same method as shown in equation (35). Equation (32) is now replaced with the new Q_{air} calculation, shown in equation (37). The ratio of moles of air to moles of carbon is calculated as 6.74 for PRB coal.

$$Q_{air} = h_{air} * \frac{mol_{air}}{mol_c} * N_c \quad \left[\frac{Btu}{hr}\right] \tag{37}$$

The final part of the equation is solving for the moles of coal burned. Q_{coal} can be represented according to equation (38). After substituting equations (28) through (38) into equation (27) and solving for the amount of coal (N_c), equation (39) develops. The result of this equation is shown in Table 10.

$$Q_{coal} = \dot{m}_{coal} * HHV_{coal} \quad \left[\frac{Btu}{hr}\right] \tag{38}$$

$$N_c = \frac{Q_{steam}}{\frac{M_C*(0.992*HHV_{coal}+18.06)}{\frac{P_C}{100}} + \left(h_{air}*\frac{mol_{air}}{mol_C}\right) - \left(\Sigma\bar{M}_{flue\ gas}*\frac{mol(\%)_i}{mol_C}*h_{flue\ gas}\right)} \quad \left[\frac{mol}{hr}\right] \tag{39}$$

The amount of coal required in the boiler is minimized when the energy in the flue gas and the radiation losses are small and when the energy in the combustion air is large. Therefore, it is advantageous to use as much heat as possible from the flue gas to create steam and to pre-heat the combustion air. The performance of the boiler is represented by boiler efficiency. The efficiency of the boiler is defined as the ratio of the steam energy to the coal energy. This is shown as equation (40). The summarized results of the calculations and the corresponding Aspen results are compared in Table 10.

$$\eta_{boiler} = \frac{Q_{steam}}{Q_{coal}} * 100 \ [\%] \tag{40}$$

Table 10. Boiler operation results.

	Unit	Calculated	Aspen	Diff.
Coal Flow Rate	[lb/hr]	646,937	652,976	0.93%
HHV Coal	[Btu/lb]	8,426	8,426	0.00%
Q Coal	[Btu/hr]	5,451,091,162	5,501,978,582	0.93%
Q Steam	[Btu/hr]	4,774,916,141	4,776,219,255	0.03%
Boiler Efficiency	[%]	87.60%	86.81%	-0.90%

2.2.3 Air Pre-Heater

The Air Pre-Heater (APH) elevates the temperature of the incoming combustion air. This improves the efficiency of the boiler (equations (27) and (32)). A diagram of the APH heat exchanger is shown in Figure 6.

There is a leakage stream included in Figure 6 which represents air transfer between the flows. In an actual boiler unit, there is leakage on both the cold side and the hot side. However, since there is a greater air to gas pressure difference on the cold side, only the cold side leakage was included in this analysis. The leakage was assumed to be 6% of the mass flow rate of the flue gas.

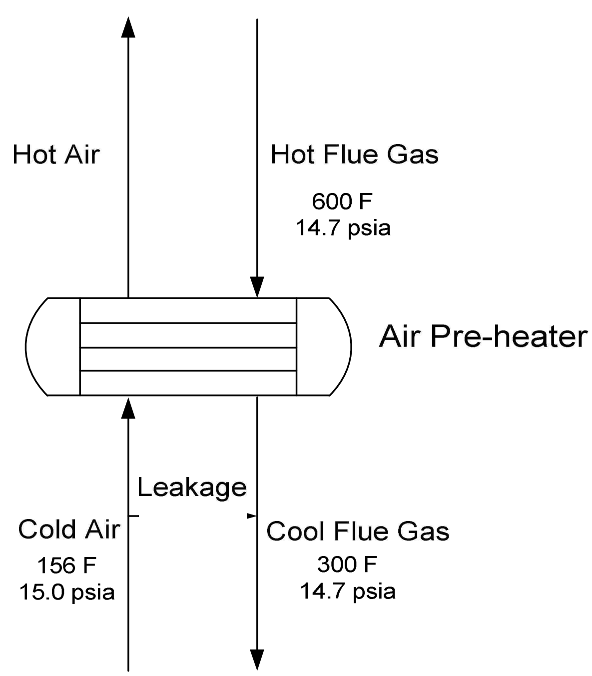


Figure 6. Diagram of APH heat exchanger flows, temperatures and pressures.

The main method for measuring the performance of the APH is by calculating its thermal effectiveness (ε). The thermal effectiveness measures the ratio of the actual heat transfer to the maximum possible heat transfer, shown in equation (41). The maximum heat transfer is determined by the minimum m*Cp (mass flow rate times specific heat) value between the two fluids, which is the air in this analysis. The actual heat transfer is the amount of heat transferred by the fluid with the minimum m*Cp value, calculated by equation (42). The effectiveness of the APH in this analysis is shown in Table 11. The energy balance is shown in Table 12.

$$\varepsilon = \frac{\dot{m}_{air} * C_{p,air} * (\mathrm{T}_{\mathrm{air,out}} - T_{\mathrm{air,in}})}{\dot{m}_{air} * C_{p,air} * (\mathrm{T}_{\mathrm{flue\ gas,in}} - T_{\mathrm{air,in}})} * 100 \ [\%] \tag{41}$$

$$Q = \dot{m} * C_p * \Delta T \ \left[\frac{Btu}{hr}\right] \tag{42}$$

Table 11. Performance of air pre-heater.

$T_{fg,in}$	$T_{air,in}$	$T_{air,out}$	Thermal Effectiveness, ε
[F]	[F]	[F]	[%]
600	156	518	81.59%

Table 12. Energy balance of air pre-heater.

Δh_{air}	m_{air}	Q_{air}	Δh_{fg}	m_{fg}	Q_{fg}	Diff.
[Btu/lb]	[lb/hr]	[Btu/hr]	[Btu/lb]	[lb/hr]	[Btu/hr]	[%]
90.05	5,197,126	468,001,196	80.11	5,808,922	465,352,741	-0.57%

2.2.4 Pollution Control Equipment

After the APH, the ash and flue gas are further treated for unit efficiency and pollution control. These include an electrostatic precipitator (ESP) and a flue gas desulfurization (FGD) unit (Figure 7). The ESP removes the ash from the flue gas stream so that only gases continue on through the plant. Then the FGD removes sulfur dioxide from the flue gas. At this point, the flue gas is considered clean enough to be exhausted through a stack into the atmosphere.

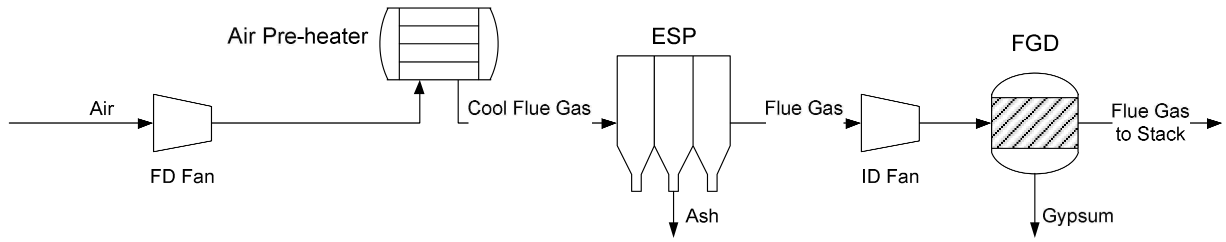


Figure 7. Flue gas pollution control equipment.

2.2.5 Station Service Power

The station service power is the power required by the plant to maintain its own operation. There are several pieces of equipment in addition to auxiliary systems such as lighting and security that require power to operate. In this analysis, the station service power is comprised of the sum of the power required by the fans, pulverizers (mills), pumps and auxiliary equipment as shown in equation (43). The net power available for sale to the electrical grid is equal to the gross power minus the station service power (equation 44).

$$P_{ss} = P_{fans} + P_{mills} + P_{pumps} + P_{aux} \ \ [kW] \tag{43}$$

$$P_{net} = P_{gross} - P_{ss} \ \ [kW] \tag{44}$$

There are two fans used near the boiler- a forced draft fan and an induced draft fan. The forced draft fan increases the pressure of the incoming combustion air. The induced draft fan increases the pressure of the flue gas after the ESP on its way to the FGD. The fans were modeled as basic compressors without consideration for a specific manufacturer or flow control. The fan efficiency was estimated as 80% for both fans. The power required to drive these fans is shown in equation (45). W_{fan} is the required power, k and R are gas constants (1.4 for air, and 0.06855 Btu/lb*R, respectively), P is the pressure, and T_1 is the temperature and $\dot{m}$ is the mass flow of the gas entering the compressor.

$$W_{fan} = \frac{1}{\eta} * \frac{k*R*T_1}{k-1} * \left(\left(\frac{P_2}{P_1} \right)^{\frac{k-1}{k}} - 1 \right) * \dot{m} * 2.931 * 10^{-4} \ \ [kW] \tag{45}$$

The pressure rise across the FD fan was assumed to be 9" H_2O and the temperature of the incoming ambient air was assumed to be 77°F. The pressure rise across the ID fan was assumed to be 35" H_2O to compensate for the pressure losses caused by the pollution control equipment (Drbal 1996). The temperature of the incoming flue gas to the ID fan was estimated to be 285°F. This was calculated by estimating the temperature after post-APH flue gas at 300°F is mixed with the APH and ESP leakage at 77°F (6% and 5% by mass, respectively). The calculated and *Aspen Plus* power for the fans are shown in Table 13.

Table 13. Fan power results.

	Unit	Calculated	Aspen	Diff.
FD Fan Power	[kW]	1,515	1,532	1.09%
ID Fan Power	[kW]	10,096	10,044	-0.52%

The energy added to the coal from the pulverization process was previously calculated using equation (31). In making these calculations, a key assumption was made that all of the power provided to the mills represented heat energy added to the coal and thus, there was no energy loss within the mill or to the environment. This assumption offsets the previous assumption that the coal mill operated at perfect efficiency. The calculated and *Aspen Plus* values for mill power are compared in Table 14. Note that the difference is equal to the difference in the mass flow rate of coal (from Table 10).

Table 14. Mill power results.

	Unit	Calculated	Aspen	Diff.
Mill Power	[kW]	3,424	3,456	0.92%

The auxiliary power is required to run lighting, internal heating and air conditioning, security systems, and other systems. It is difficult to estimate the auxiliary power without knowing the specific details of each plant. Thus, in this analysis, an auxiliary power of 15,000 kW was assumed constant for the unit in all cases. When summed with the remaining station service power components, the station service power is approximately 32 MW at design conditions. This is consistent with the general standard that station service power is approximately 5% of the gross power generated by a plant.

2.2.6 Plant Performance

The overall performance of the boiler and steam turbine cycle is defined as the net unit heat rate (NUHR). The NUHR is the ratio of the coal energy to the net power. It is calculated according to equation (46). The unit efficiency is a related performance measure obtained by the conversion of the heat rate units to a percentage by a factor of 3412 $\frac{Btu}{kWh}$ in equation (47).

$$HR_{net\ unit} = NUHR = \frac{Q_{coal}}{P_{net}} \left[\frac{Btu}{kWh}\right] \quad (46)$$

$$Unit\ Efficiency = \eta_{unit} = \frac{3412}{NUHR} * 100\ \ [\%] \quad (47)$$

The scale of NUHR and efficiency are inversely proportionate to one another – a unit has better performance with a lower NUHR or a higher efficiency. The *Aspen Plus* unit operation results for this analysis are summarized in Table 15.

Table 15. Summary of unit operation results in *Aspen Plus*.

Boiler Performance	Unit	*Aspen Plus*
Q Steam	[Btu/hr]	4,776,219,255
Coal Flow Rate	[lb/hr]	652,976
HHV Coal	[Btu/lb]	8,426
Q Coal	[Btu/hr]	5,501,978,582
Turbine Power		
HP Power	[kW]	183,314
IP1 Power	[kW]	106,165
IP2 Power	[kW]	65,888
LP1 Power	[kW]	57,559
LP2 Power	[kW]	84,977
LP3 Power	[kW]	39,182
LP4 Power	[kW]	42,908
LP5 Power	[kW]	55,774
Gross Power	[kW]	635,768
Station Service Power		
Mill Power	[kW]	3,456
Fan Power	[kW]	11,576
Pump Power	[kW]	1,658
Auxiliary Power	[kW]	15,000
Total P_{ss}	[kW]	31,689
Plant Performance		
Net Power	[kW]	604,079
Boiler Efficiency	[%]	86.81%
Turbine Cycle HR	[Btu/kWh]	7,512
Net Unit HR	[Btu/kWh]	9,108
Unit Efficiency	[%]	37.46%

2.3 Monoethanolamine (MEA) Post-combustion Capture System

Carbon dioxide was captured from the flue gas using a chemical separation process. The *Aspen Plus* model for this process was developed by Dr. Ian Laurenzi, a former Assistant Professor of Chemical Engineering at Lehigh University and Austin Szatkowski, a former Master's student at the Energy Research Center. The chemical solution used is a mixture of monoethanolamine (MEA) and water. MEA is a weak base and thus selectively binds carbon dioxide, which is a weak acid. The model is designed to have a 90 mol% carbon capture rate. There are 4 main pieces of equipment that are required for MEA-based carbon capture, which are an absorber, stripper, pump, and heat exchanger. The absorber removes the carbon dioxide from the flue gas. The pump and heat exchanger pressurize and heat the CO_2-rich amine before it enters the stripper. The stripper regenerates the MEA solution and produces a pure CO_2 stream ready for compression.

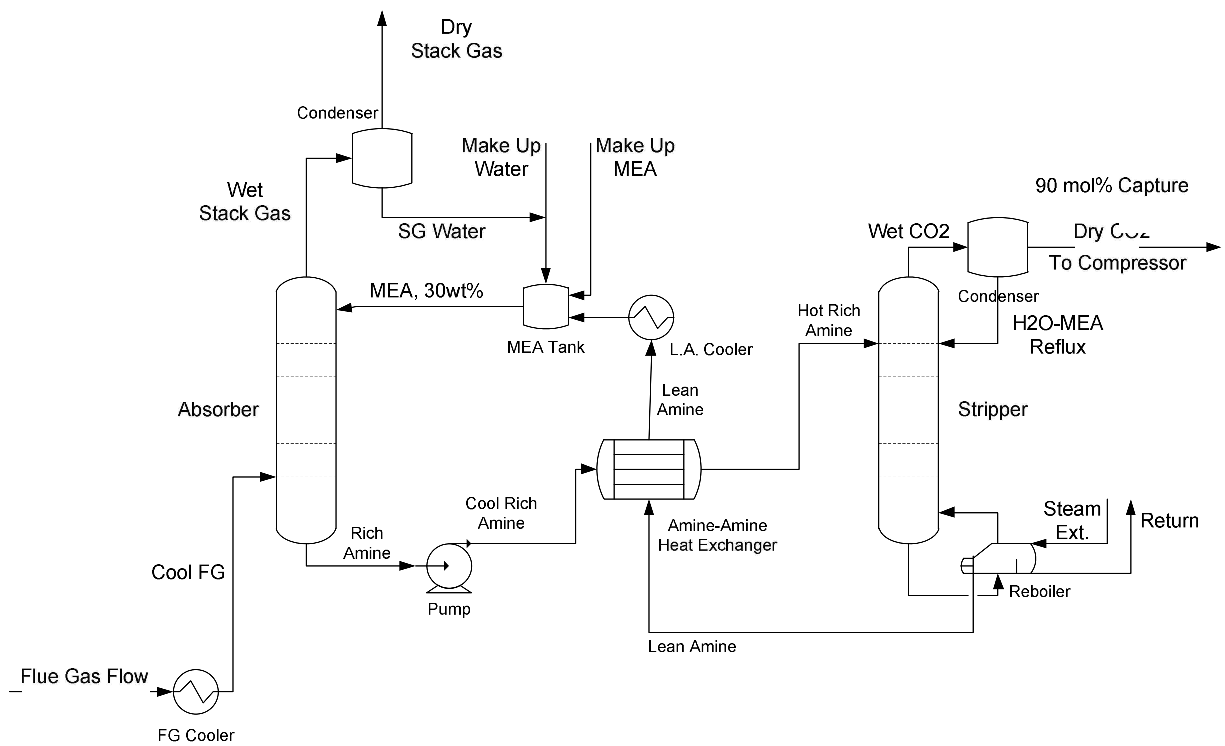


Figure 8. Diagram of MEA System Equipment.

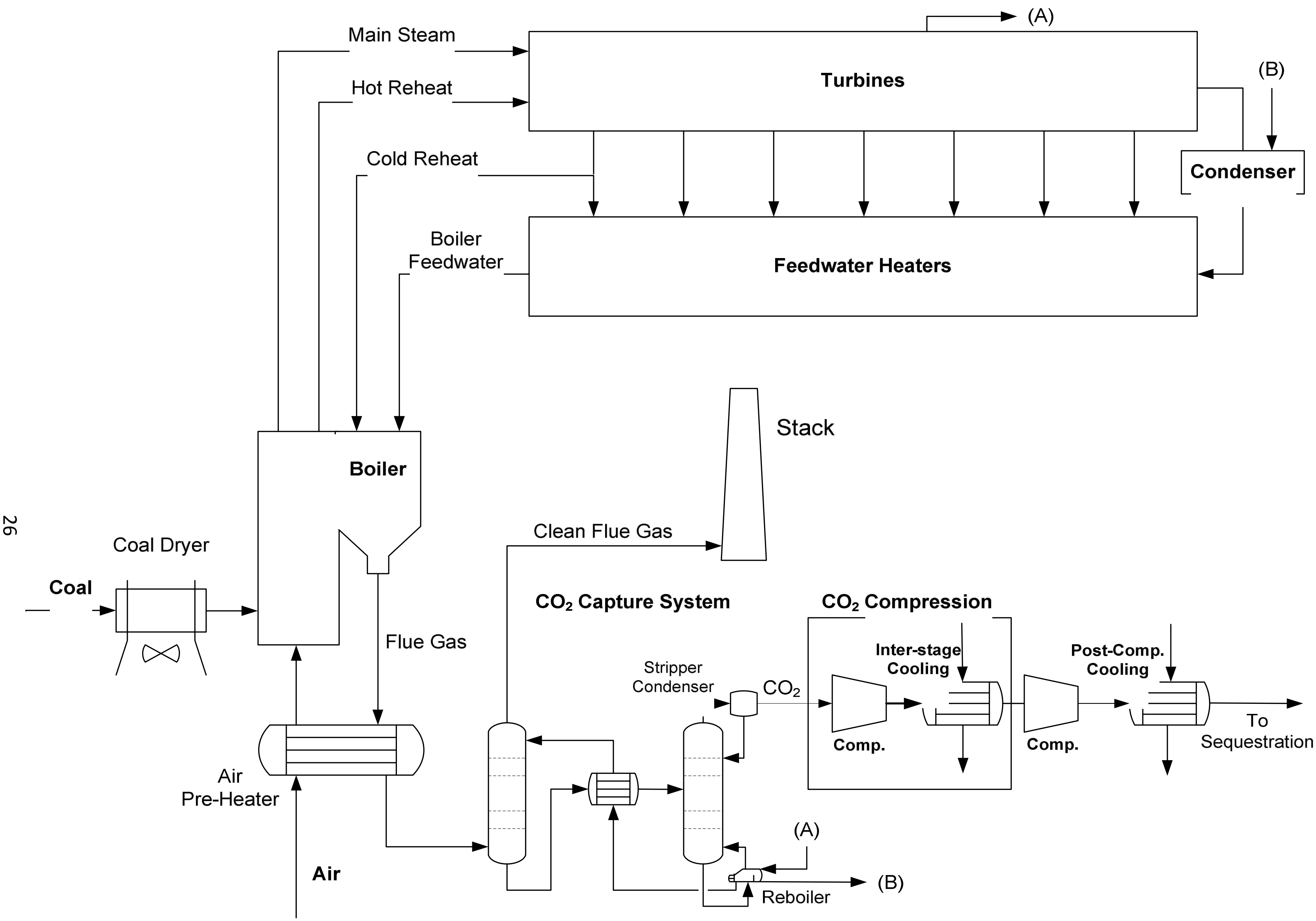


Figure 9. Conceptual Diagram of Plant Layout with MEA Based Carbon Capture.

2.3.1 Absorber Column

The absorber operates at a pressure of 1 atmosphere and at a temperature of 100°F. These settings were determined to be suitable for a 90% carbon dioxide capture rate (Szatkowski, 2009). MEA and water solution is fed into the top of the absorber column while the flue gas is fed to the bottom of the column (Figure 8). The MEA-water mixture and the flue gas flow counter-currently and the MEA solution absorbs the carbon dioxide as their molecular paths cross. The top of the column contains flue gas with minimal CO_2 content and the product at the bottom of the column is an MEA-CO_2 solution, called rich amine.

2.3.2 Stripper Column

The stripper operates at 3 atmospheres of pressure and at a temperature of approximately 268°F. The elevated pressure and temperature, compared to the absorber, is required to separate the MEA and carbon dioxide. The hot rich amine is fed to the top of the stripper column, where it sinks to the bottom and flows into the reboiler. In the reboiler, heat is added to the MEA solution to drive the separation process. The heat is usually provided by a steam extraction from the steam cycle. The contents of the reboiler are separated by partially boiling the rich amine. The liquid from this process is regenerated MEA solution, called lean amine, which is reused in the absorber. The vapor from this process is mostly carbon dioxide and water vapor with minor amounts of MEA. This vapor re-enters the column and rises to the top where it enters the stripper condenser. Then water vapor and trace MEA are condensed and fed back to the column. The condenser vapor is carbon dioxide ready for compression. The results presented here are for 90% capture (by mole) of the carbon dioxide entering the absorber.

The amount of heat required for the stripper reboiler depends on several factors, which must be optimized to reduce the cost of the MEA system. These factors include the rate of CO_2 capture, the saturation of the carbon dioxide in the rich and lean amine streams, and the temperature and pressure of the stripper column. Since there may be multiple optimized scenarios, the heat requirement is often expressed as heat per pound of carbon dioxide captured.

The literature reports a range of values between 1600 and 1900 Btu/lb CO_2 captured and the present model used 1688 Btu/lb CO_2 captured (Table 16).

Table 16. MEA model settings compared to literature values.

	Unit	Martin	Szatkowski 2009	Fisher 2005	Knudsen 2009
		Model	Model	Model	Pilot
MEA to Flue Gas Ratio	[lb/lb]	4	4	3.89	2-3.2
Lean MEA Loading	[mol CO2/mol MEA]	0.115	0.276	0.249	---
Rich MEA Loading	[mol CO2/mol MEA]	0.315	0.479	0.459	---
Delta Loading	[mol CO2/mol MEA]	0.200	0.203	0.210	---
Reboiler Temp	[F]	270	273	253	---
Reboiler Duty	[Btu/lb CO2 Cap.]	1688	1750	1872	1753
Reboiler Duty Diff.	[%]	0.00%	3.67%	10.90%	3.85%

2.3.3 Additional Equipment

The other equipment required for the MEA capture system are a pump and a heat exchanger. The pump follows the absorber and pumps the rich amine up to the stripper pressure of 3 atmospheres. Then, the rich amine enters the amine-amine heat exchanger where the rich amine is heated by the lean amine liquid leaving the stripper. The rich amine then enters the stripper at the elevated pressure and temperature and the lean amine is cooled further to be used in the absorber again.

2.3.4 Effect of MEA Carbon Capture on Unit Performance

The addition of carbon capture and CO_2 compression affects the plant performance in four ways. The low pressure steam extraction for the reboiler reduces the gross power produced by the turbines. The compressor consumes power which is assumed to come from the plant's own generation. Finally, the ID fan power increases to overcome the additional pressure drop experienced by the flue gas through the CO_2 capture system and additional pumps are required in the MEA system and in the compressor. The boiler performance remains the same as one operating without carbon capture.

This section discusses the specific impacts of the MEA capture system, operating at a 90mol% capture rate, on plant performance. The following section, Section 2.4, discusses four compressor options selected for this analysis and the impacts of each on plant performance. As a result, there are four "base cases" for the power plant, depending on the compressor selected. The performance of one base case is shown at the end of this section for illustration. The performances of all four base cases are shown in Section 2.4.3, after an analysis of each compressor option.

In this analysis, the steam used to run the reboiler is extracted between low pressure turbines LP1 and LP2 (see Figure 2). Thus, the power produced by the HP through LP1 turbines also remains the same as without carbon capture. However, the power from turbines LP2 through LP5 are all directly affected by the reboiler extraction steam since the steam is not available to go through the remaining turbines downstream. In all of the base cases, the gross turbine power is reduced by 23.5% due to the reboiler extraction.

The power required to run the fans and pumps also increases with carbon capture. The ID fan power increases to compensate for the additional pressure drop through the absorber column. The additional pressure drop in the flue gas stream is estimated to be 25" H_2O. Thus, the total pressure rise across the ID fan is now estimated to be 60" H_2O. The pump power increases with the addition of a lean amine pump in between the absorber and stripper columns and the addition of pumps used to circulate water to cool the CO_2 between compression stages. Different types of compressors have a different number of stages and each stage has a dedicated pump. The total pump power is dependent on the number of compressor stages and the amount of cooling required since this determines the size of the heat exchangers and the flow rate of cooling water.

The performance of the MEA capture plant is dependent on the type of compressor selected. When the fan and pump changes are taken together, the station service power increases 24.1 to 24.3% in the base cases, depending on the type of compressor (for complete details, see Table 30). A summary of boiler, steam turbine cycle, and station service power results reflecting the addition of MEA-based carbon capture is shown in Table 17.

Table 17. Summary of MEA-Based Carbon Capture on Plant Performance in *Aspen Plus*.

Boiler Performance	Unit	MEA Capture	No Capture	Diff.
Q Steam	[Btu/hr]	4,776,219,255	4,776,219,255	0.0%
Coal Flow Rate	[lb/hr]	652,976	652,976	0.0%
HHV Coal	[Btu/lb]	8,426	8,426	0.0%
Q Coal	[Btu/hr]	5,501,978,582	5,501,978,582	0.0%
Boiler Efficiency	[%]	86.81%	86.81%	0.0%
Turbine Power				
HP Power	[kW]	183,314	183,314	0.0%
IP1 Power	[kW]	106,165	106,165	0.0%
IP2 Power	[kW]	65,888	65,888	0.0%
LP1 Power	[kW]	57,559	57,559	0.0%
LP2 Power	[kW]	31,476	84,977	-63.0%
LP3 Power	[kW]	13,513	39,182	-65.5%
LP4 Power	[kW]	13,547	42,908	-68.4%
LP5 Power	[kW]	14,758	55,774	-73.5%
Gross Power	[kW]	486,222	635,768	-23.5%
Turbine Cycle HR	[Btu/kWh]	9,973	7,627	30.8%
Station Service Power				
Fan Power	[kW]	18,405	11,576	59.0%
Mill Power	[kW]	3,456	3,456	0.0%
Pump Power	[kW]	2,471	1,658	49.0%
Auxiliary Power	[kW]	15,000	15,000	0.0%
Total P_{ss}	[kW]	39,332	31,689	24.1%
Compressor Performance				
Ramgen Comp. Power	[kW]	45,593	N/A	N/A
Plant Performance				
Net Power	[kW]	394,003	594,543	-33.7%
Net Unit HR	[Btu/kWh]	13,964	9,254	50.9%
Unit Efficiency	[%]	24.43%	36.87%	-12.4%

2.4 CO_2 Compression

Once the carbon dioxide has been captured, the CO_2 is compressed so it is ready for geological sequestration. The specifics of geological sequestration are beyond the scope of this report. The final CO_2 pressure used in this study was 2220 psia. This study investigated three main types of compressors which included Ramgen, inline and integrally geared compressors. The primary objective of this analysis is to quantify the heat generated from the compression process and evaluate the potential to integrate this heat into the plant. Thus, each type of compressor was evaluated with each stage's discharge temperature and heat transferred during intercooling in mind.

2.4.1 CO_2 properties: REFPROP vs *Aspen Plus*

REFPROP is a software program that accesses and displays National Institute of Standards and Technology (NIST) thermodynamic properties (Levy 2010). These properties can be used to calculate compressor performance from first principles. However, the power plant is modeled in *Aspen Plus*. In order to streamline the modeling process, the compressors were modeled in *Aspen Plus* also. The properties of carbon dioxide in the "Pure22" database from *Aspen Plus* were compared to the properties in the NIST database via REFPROP. This comparison demonstrates that the compressors modeled in *Aspen Plus* have similar performance as the manufacturer's performance data.

The CO_2 properties of enthalpy, entropy and specific heat were compared in REFPROP and in *Aspen Plus*. A range of pressures (200-2200 psia) and temperatures (100-500°F) were used for these comparisons. The pressure range was chosen to encompass most pressures encountered during the compression process. The lower temperature boundary was chosen because it is above the critical temperature of 88°F. For enthalpy and entropy, most equations use two values since performance depends on the change in value (Δh or Δs), not the value itself. For specific heat, the value is only dependent on temperature and it is used as a constant in calculations.

The REFPROP and *Aspen Plus* entropy and enthalpy values were tabulated at each temperature and pressure, while the specific heat values were tabulated at each temperature. However, the *Aspen Plus* Pure 22 property database uses different baseline values for enthalpy and entropy than REFPROP. In order to compare values, the *Aspen Plus* values for enthalpy and entropy at 200 psia and 100°F was adjusted to match the REFPROP values at the same pressure and temperature, and all other *Aspen Plus* values were adjusted by the same amount. Since the specific heat is constant at a given temperature, the *Aspen Plus* values did not need to be adjusted.

Tables and graphs comparing REFPROP and *Aspen Plus* values for entropy, enthalpy and specific heat versus temperature are included in Appendix D. There is reasonable agreement between REFPROP and *Aspen Plus* values. Entropy values from *Aspen Plus* were on average 0.06% lower than REFPROP values with a range of -0.42 to 2.98%. Enthalpy values in *Aspen Plus* were on average 0.07% higher than REFPROP values with a range of -0.40 to 4.31%. For specific heat, *Aspen Plus* values were on average 0.44% higher than REFPROP values with a range of -0.10 to 0.75%. Therefore, it is reasonable to accept the performance of the compressor models in *Aspen Plus*. Note that the values are for pure carbon dioxide. The compressors used at the end of a coal-fired power plant with MEA capture will compress a mixture of approximately 98.3% CO_2 and 1.7% H_2O with trace amounts of nitrogen and oxygen.

2.4.2 Overview of Compressors and Intercoolers

In this analysis, an initial CO_2 pressure of 44.1 psia was used since it is the pressure of the MEA stripper column. A final pressure of 2220 psia was used as a representative value necessary for geologic sequestration. Each compressor stage had an isentropic and mechanical efficiency which was obtained from the manufacturer. Pressure ratios were also obtained from the manufacturer's specifications and were maintained as much as possible (Charles 2010). Occasionally, due to different inlet and outlet pressures required for this analysis, pressure ratios were adjusted slightly from the manufacturer's specifications.

Cooling between each compressor stage reduced the temperature of the CO_2 stream to 110°F unless otherwise noted. The inter-stage cooling increases the efficiency of the compressor since the compressor work is proportional to the inlet temperature of the gas (Cengel & Boles 2008). The intercooling temperature was selected to be greater than the critical point temperature (88°F) and thus potential issues with CO_2 condensation are avoided.

Intercooling was achieved with a shell-and-tube, cross-flow heat exchanger in which 90 or 100°F cooling water enters the tubes and the CO_2 is in the shell. The flow rate of cooling water was minimized in order to achieve a high cooling water outlet temperature while maintaining liquid flow, thus increasing the potential for thermal integration. A hot inlet-cold outlet temperature difference of 10°F was selected to "set" the cooling water flow rate in the heat exchanger model. More heat transfer is possible if a smaller temperature difference can be achieved by the heat exchanger design. If any water condenses out of the CO_2 stream after it's cooled, the water is removed so liquid does not enter the following compressor stage.

For this analysis, three types of compressors were evaluated. One unique compressor option is being developed by Ramgen Power Systems. Also, two configurations of centrifugal compressors, inline and integrally geared, were evaluated. One compressor of each type was modeled following the manufacturer's specifications with minor adjustments as necessary. The integrally geared compressor was modeled twice with different intercooling temperatures.

2.4.2.1 Ramgen Compressor

An innovative type of compressor has been developed by Ramgen Power Systems that can achieve high pressure ratios. The technology takes advantage of the shock effects created as the gas in the compressor exceeds Mach 1 (Levy 2010). The high pressure ratios allow the compression to take place in two stages instead of as many as eight stages with an integrally geared compressor. As a consequence, the gas discharge temperatures are much higher than typical centrifugal designs, thus increasing the potential for thermal integration. The footprint and costs of a Ramgen compressor are lower since there are fewer stages and thus less material. A diagram of the compressor is shown in Figure 10.

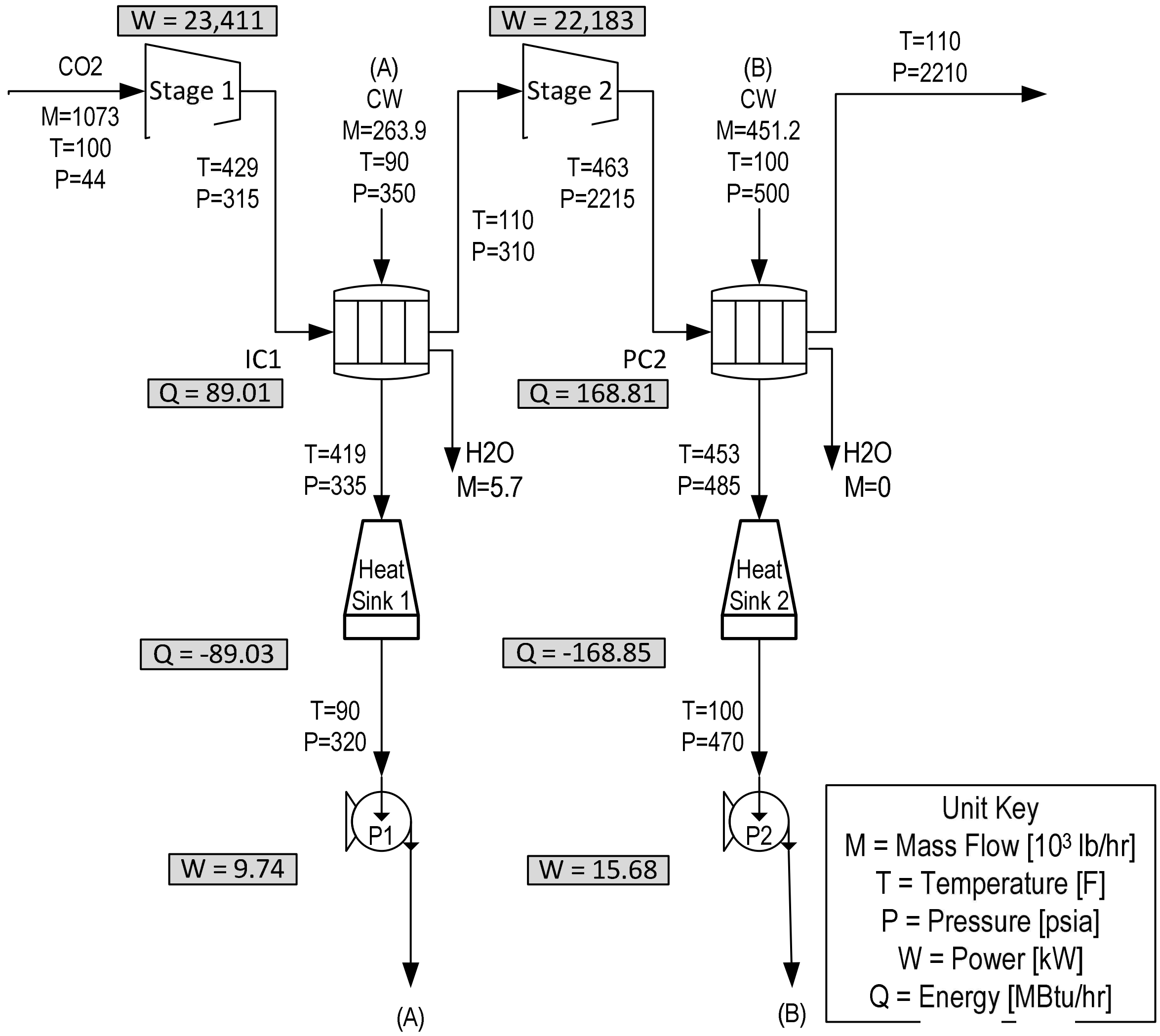


Figure 10. Ramgen Compressor Specifications.

The manufacturer's original specifications were obtained from Levy (2010). However, the manufacturer's conditions were different than those required for this analysis. This analysis used a pressure range of 44.1 to 2220 psia, so a pressure ratio of 7.145 was used for each stage. It was assumed that the isentropic and mechanical efficiencies remain constant with this change in pressure ratio. The complete inlet specifications of the compressor stages and their performance are included in Appendix E. The compressor stage powers were calculated from first principles using Equation (48) and enthalpy values obtained from REFPROP. The performance results are also summarized in Table 18.

$$W_{compressor} = \dot{m} * (h_{2a} - h_1) * 0.0002186 \ [kW] \tag{48}$$

Table 18. Power Required for Ramgen Compressor.

Stage	Aspen CO_2 Discharge Temp	Aspen Power	REFPROP CO_2 Discharge Temp	REFPROP Power	Power Diff.
	[F]	[kW]	[F]	[kW]	[%]
Stage 1	429.3	23,386	427.7	22,685	-3.00%
Stage 2	463.0	21,635	458.1	20,714	-4.26%
Total		45,593		43,399	-3.60%

The intercooler and post-compressor cooler's complete specifications and their performances are also included in Appendix E. This table includes *Aspen Plus* results compared to REFPROP-based calculations from first principles. The heat transfer in the coolers was calculated with REFPROP properties using Equation (49). The second part of the equation accounts for the latent heat of vaporization (h_{fg}), energy released from the water vapor within the CO_2 stream as it changes phase. The maximum heat available from the coolers for thermal integration is summarized in Table 19. The unit operation results for a carbon capture plant with a Ramgen compressor are summarized in Table 20. The boiler and turbine cycle information is the same for all compressors and is located in Table 17.

$$Q = \left[\dot{m}_{\,CO_2} * \left(h_{CO_2,out} - \mathrm{h}_{CO_2,in}\right)\right] + \left[\dot{m}_{\,H_2O\ knockout} * h_{fg\ @\ P_{CO2out}}\right]\ \left[\frac{Btu}{hr}\right] \qquad (49)$$

Table 19. Maximum heat available from heat exchangers used with Ramgen compressor.

Heat Source	CO_2 Inlet Temp	CO_2 Outlet Temp	Maximum Heat Available: Aspen	Maximum Heat Available: REFPROP	Diff.
	[F]	[F]	[Btu/hr]	[Btu/hr]	[%]
Intercooler (IC1)	429.3	110	89,013,445	87,838,488	-1.3%
Post-Comp. Cooler (PC2)	463.0	110	168,811,875	170,341,167	0.9%
TOTAL			257,825,320	258,179,655	0.1%

Table 20. Unit Operation Results of MEA Carbon Capture Plant with a Ramgen Compressor compared to a "No-Capture" Plant.

Station Service Power	Unit	MEA Capture & Ramgen	No Capture	Diff.
Fan Power	[kW]	18,405	11,576	59.0%
Mill Power	[kW]	3,456	3,456	0.0%
Pump Power	[kW]	2,471	1,658	49.0%
Auxiliary Power	[kW]	15,000	15,000	0.0%
Total P_{ss}	[kW]	39,332	31,689	24.1%
Ramgen Performance				
Compressor Power	[kW]	45,593	N/A	N/A
Plant Performance				
Qcoal	[Btu/hr]	5,501,978,582	5,501,978,582	0.0%
Net Power	[kW]	394,003	594,543	-33.7%
Net Unit HR	[Btu/kWh]	13,964	9,254	50.9%
Unit Efficiency	[%]	24.43%	36.87%	-12.4%

2.4.2.2 Inline Compressor

Most compressors on the market are of a centrifugal design. The centrifugal design first accelerates the gas in a rotating impeller, and then forces the gas through a diffuser at which point the velocity of the gas is converted into a higher pressure. This process can be repeated multiple times to achieve the desired final pressure. Cooling between stages is achieved by ducting the gas to an external cooler.

Inline centrifugal compressors "have all the impellers driven by the same shaft in a single casing." (Levy 2010). Therefore, each impeller rotates at the same speed. They are inflexible and do not operate well at partial loading. In addition, external cooling between stages can be difficult since the gas must be piped to the cooler after several stages of compression within a single casing. These requirements increase the footprint of an inline compressor. However, inline compressors are easier to maintain than other centrifugal compressors (Levy 2010).

The "Inline 4" compressor, from Charles (2010), was selected as the model for an inline type of compressor (Figure 11). The number 4 signifies a specific manufacturer's compressor model, the identity of which is on file at the Energy Research Center. The compressor stages' inlet specifications and performance data are included in Appendix E. The required pressures for this analysis are different from the manufacturer's specifications, but it was assumed that the isentropic and mechanical efficiencies at different pressure ratios remained valid. In this analysis, the pressure ratios for stages 1 and 2 were maintained, while the stage three pressure ratio was reduced from 3.700 to 1.294.

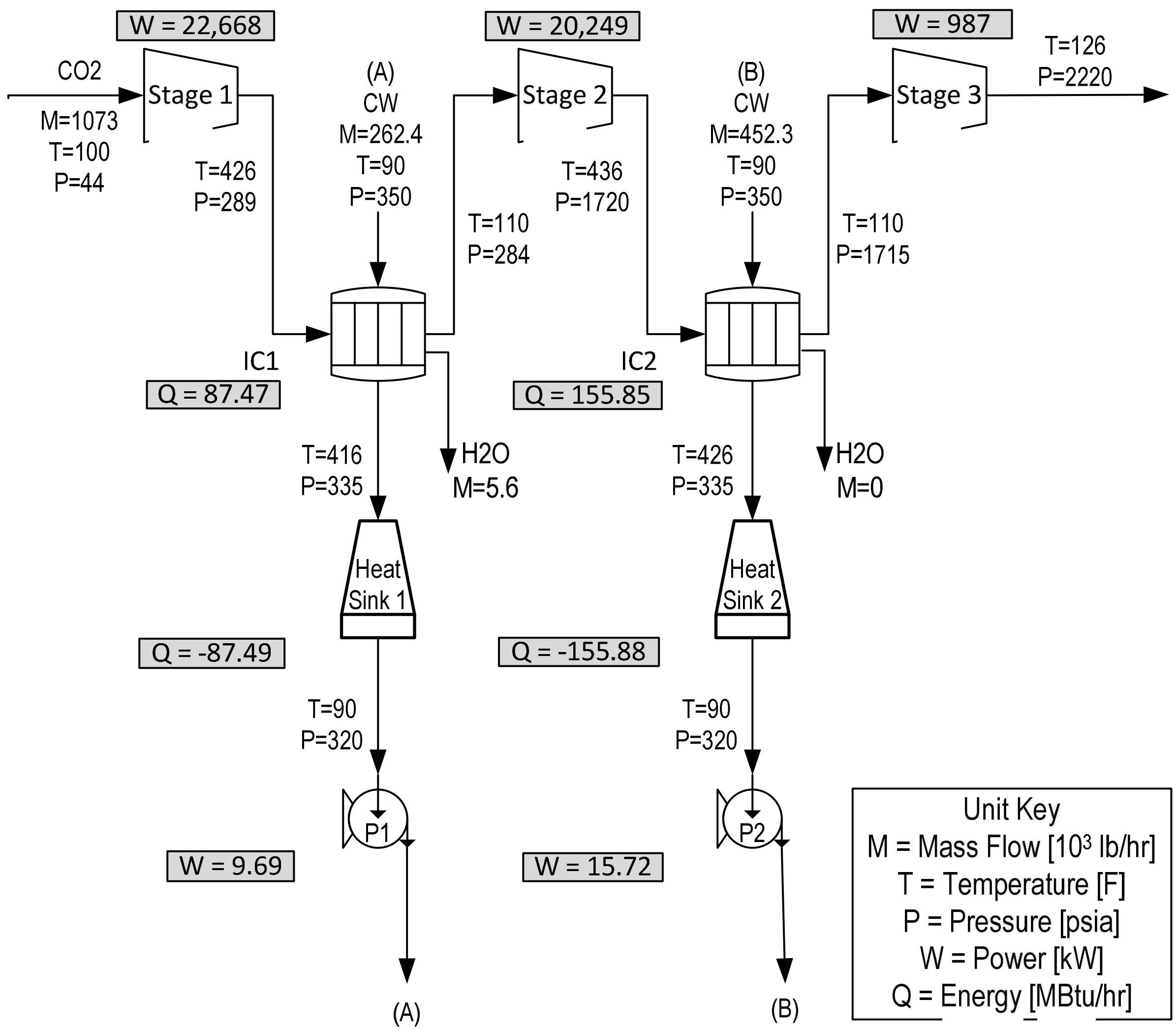


Figure 11. Inline 4 Compressor Specifications.

The compressor stages' complete inlet specifications and their performances are included in Appendix E. The compressor powers were also calculated from REFPROP values using Equation (48). The compressor power requirements are summarized in Table 21. Although the percent difference for stage 3 is high, the absolute difference between the numbers is small and the difference in total power is also small. Therefore, the *Aspen Plus* model is considered accurate.

Table 21. Power Required for Inline 4 Compressor.

Stage	Aspen CO_2 Discharge Temp	Aspen Power	REFPROP CO_2 Discharge Temp	REFPROP Power	Power Diff.
	[F]	[kW]	[F]	[kW]	[%]
Stage 1	425.9	22,668	424.3	22,464	-0.9%
Stage 2	436.1	20,249	431.2	19,753	-2.5%
Stage 3	125.9	987	123.5	842	-14.7%
Total Power		43,905		43,058	-1.9%

Intercoolers were used after stages 1 and 2. The discharge temperature from Stage 3 (125.9°F) is very close to 110°F and thus was considered an appropriate temperature for sequestration. The cost of an additional heat exchanger is not necessary. The intercoolers' complete inlet specifications and their performance are included in Appendix E. The maximum heat available for thermal integration from each heat exchanger is summarized in Table 22. The unit operation results for a carbon capture plant with an Inline 4 compressor are summarized in Table 23. The results for the boiler and turbine cycle are the same for all compressors and are located in Table 17.

Table 22. Maximum Heat Available from Heat Exchangers of Inline 4 Compressor.

Heat Source	CO_2 Inlet Temp	CO_2 Outlet Temp	Maximum Heat Available: Aspen	Maximum Heat Available: REFPROP	Diff.
	[F]	[F]	[Btu/hr]	[Btu/hr]	[%]
Intercooler 1 (IC1)	425.9	110	87,469,588	86,280,542	-1.4%
Intercooler 2 (IC2)	436.1	110	155,846,773	159,593,365	2.4%
TOTAL			243,316,361	245,873,907	1.1%

Table 23. Unit Operation Results of MEA Carbon Capture Plant with an Inline 4 Compressor compared to a "No-Capture" Plant.

Station Service Power	Unit	MEA Capture & Inline 4	No Capture	Diff.
Fan Power	[kW]	18,405	11,576	59.0%
Mill Power	[kW]	3,456	3,456	0.0%
Pump Power	[kW]	2,471	1,658	49.0%
Auxiliary Power	[kW]	15,000	15,000	0.0%
Total P_{ss}	[kW]	39,332	31,689	24.0%
Compressor Performance				
Compressor Power	[kW]	43,905	N/A	N/A
Plant Performance				
Qcoal	[Btu/hr]	5,501,978,582	5,501,978,582	0.0%
Net Power	[kW]	395,691	594,543	-33.3%
Net Unit HR	[Btu/kWh]	13,905	9,254	49.9%
Unit Efficiency	[%]	24.54%	36.87%	-12.3%

2.4.2.3 Integrally Geared Compressors

Another configuration of a centrifugal compressor is an integrally geared compressor. The integrally geared design has a central bull gear that connects to several pinions. Each pinion drives an impeller in its own casing. This arrangement allows for easier external cooling between stages. The pinions "allow each pair of impellers to rotate at different speeds while being driven by the same bull gear" (Levy 2010). A single frame can accommodate up to 10 stages of compression (5 pinions, 10 impellers) (Levy 2010). Integrally geared compressors are more flexible and operate more efficiently at partial loads than inline compressors, but they are less reliable since they are more mechanically complex (Levy 2010).

The "Integrally Geared 1" compressor, from Charles (2010), was selected as the model for the integrally geared type of compressor. Again, the number "1" signifies a specific manufacturer's compressor model, the identity of which is on file at the Energy Research Center. The manufacturer's pressure ratios were maintained except for Stage 1 and Stage 6. These pressure ratios were reduced from 2.405 to 1.503 and from 1.622 to 1.552, respectively, to achieve the desired final pressure.

The manufacturer specified using intercooling to 149°F after each stage without any cooling after stage 6, hereafter referred to as IG1-149. Cooling to 110°F was selected for the post-compressor cooler after stage 7 (Figure 12). In addition, the performance of the IG 1 compressor was evaluated using intercooling to 110°F after all stages (Figure 13), hereafter referred to as IG1-110. It is unknown why the manufacturer selected an intercooling temperature of 149°F or whether an intercooling temperature of 110°F is achievable. However, it is useful to demonstrate the difference in performance and potential for thermal integration between the two scenarios.

The compressor stages' inlet specifications and performance data is included in Appendix E. The power requirements for the IG1-149 compressor are summarized in Table 24 and the power requirements for the IG1-110 compressor are summarized in Table 25. The total power required for the IG1-110 intercooling is less than the power required for the IG1-149 since compressor power is proportional to the inlet gas temperature.

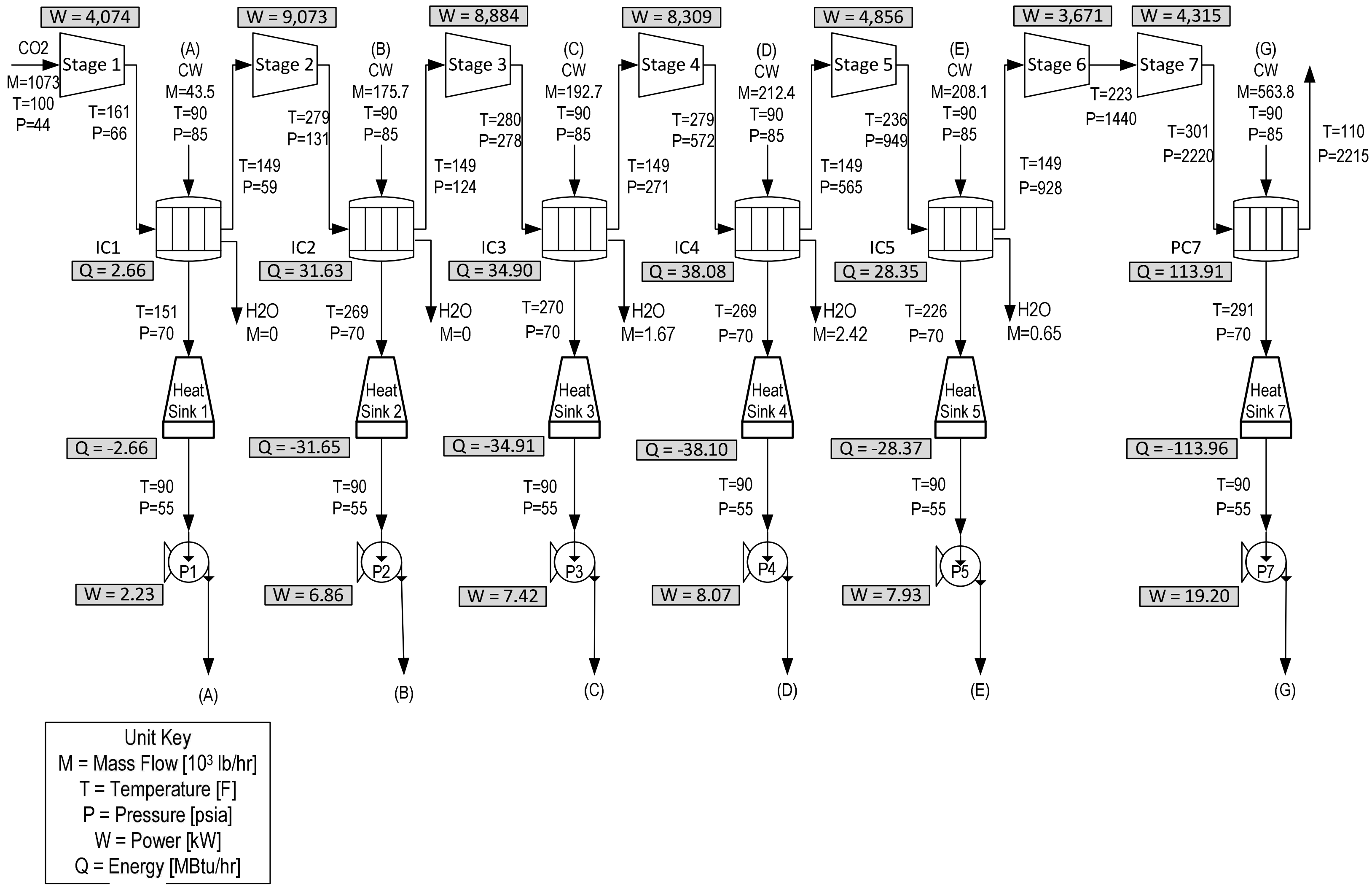


Figure 12. Integrally Geared 1 Compressor Specifications with 149F Intercooling (IG1 – 149).

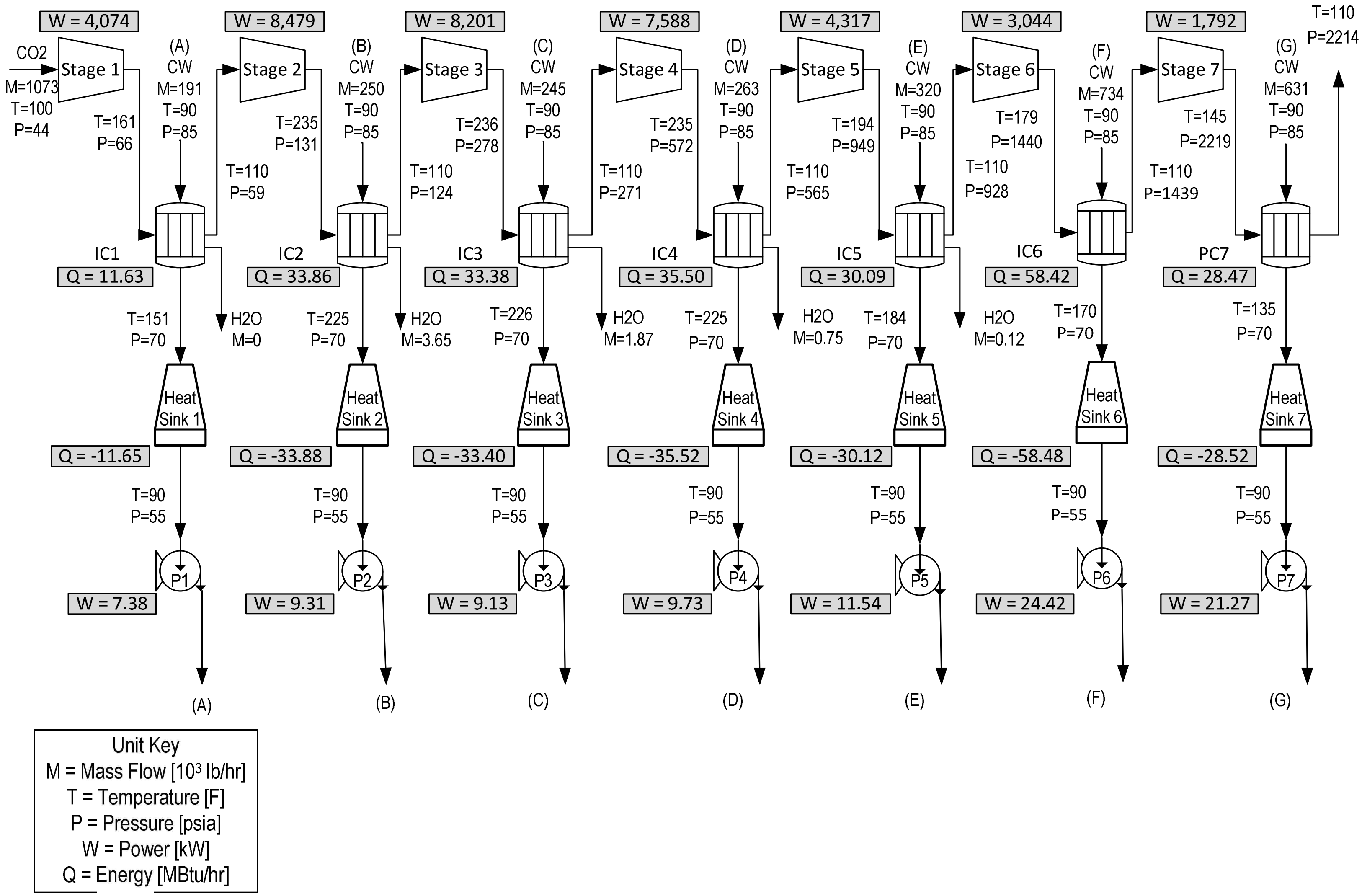


Figure 13. Integrally Geared 1 Compressor Specifications with 110F Intercooling (IG1 – 110).

Table 24. Power required for Integrally Geared 1 Compressor with 149°F Intercooling.

Stage	Aspen CO_2 Discharge Temp	Aspen Power	REFPROP CO_2 Discharge Temp	REFPROP Power	Power Diff.
	[F]	[kW]	[F]	[kW]	[%]
Stage 1	161.1	4,074	160.9	3,956	-2.9%
Stage 2	279.3	9,073	277.5	8,714	-4.0%
Stage 3	280.3	8,884	277.7	8,494	-4.4%
Stage 4	278.5	8,309	275.1	7,913	-4.8%
Stage 5	236.1	4,856	233.0	4,602	-5.2%
Stage 6	223.0	3,671	219.6	3,347	-8.8%
Stage 7	300.7	4,315	297.5	3,742	-13.3%
Total		43,182		40,768	-5.6%

Table 25. Power required for Integrally Geared 1 Compressor with 110°F Intercooling.

Stage	Aspen CO_2 Discharge Temp	Aspen Power	REFPROP CO_2 Discharge Temp	REFPROP Power	Power Diff.
	[F]	[kW]	[F]	[kW]	[%]
Stage 1	161.1	4,074	160.9	3,956	-2.9%
Stage 2	235.2	8,479	233.4	8,145	-3.9%
Stage 3	236.3	8,201	233.4	7,839	-4.4%
Stage 4	234.8	7,588	230.8	7,221	-4.8%
Stage 5	194.3	4,317	190.5	4,083	-5.4%
Stage 6	179.7	3,044	176.0	2,757	-9.4%
Stage 7	145.2	1,793	141.1	1,363	-23.98%
Total		37,496		35,364	-5.69%

The intercoolers' complete inlet specifications and their performance are included in Appendix E. The maximum heat available for thermal integration from each heat exchanger is summarized in Table 26 for IG1-149 and Table 27 for IG1-110. There is more heat available in IG1-110 since the CO_2 is transferring more of its heat to the cooling water. Of course, with the addition of Intercooler 6 in the IG1-110, there is less heat available in the Post Compressor Cooler (PC7) than in PC7 from IG1-149. The unit operation results for the capture model compared to the non-capture model for IG1-149 and IG1-110 are shown in Table 28 and Table 29, respectively. Since IG1-110 requires less power, the net power and plant efficiency are greater than the results for IG1-149 compressor.

Table 26. Maximum Heat Available from Integrally Geared 1 Compressor with 149°F Intercooling.

Heat Source	CO_2 Inlet Temp	CO_2 Outlet Temp	Maximum Heat Available: Aspen	Maximum Heat Available: REFPROP	Diff.
	[F]	[F]	[Btu/hr]	[Btu/hr]	[%]
Intercooler 1 (IC1)	161.1	149.0	2,656,018	2,672,035	0.6%
Intercooler 2 (IC2)	279.3	149.0	31,633,848	31,737,217	0.3%
Intercooler 3 (IC3)	280.3	149.0	34,898,117	34,691,126	-0.6%
Intercooler 4 (IC4)	278.5	149.0	38,079,677	37,734,393	-0.9%
Intercooler 5 (IC5)	236.1	149.0	28,348,792	28,563,733	0.8%
Post-Comp. Cooler (PC7)	300.7	110.0	113,913,503	115,955,584	1.8%
TOTAL			249,529,955	251,354,088	0.7%

Table 27. Maximum Heat Available from Integrally Geared 1 Compressor with 110-120°F Intercooling.

Heat Source	CO_2 Inlet Temp	CO_2 Outlet Temp	Maximum Heat Available: Aspen	Maximum Heat Available: REFPROP	Diff.
	[F]	[F]	[Btu/hr]	[Btu/hr]	[%]
Intercooler 1 (IC1)	161.1	110.0	11,632,909	11,686,878	0.5%
Intercooler 2 (IC2)	235.2	110.0	33,861,575	33,260,229	-1.8%
Intercooler 3 (IC3)	236.3	110.0	33,382,485	33,245,054	-0.4%
Intercooler 4 (IC4)	234.8	110.0	35,495,387	35,935,925	1.2%
Intercooler 5 (IC5)	194.3	110.0	30,093,402	31,106,383	3.4%
Intercooler 6 (IC6)	179.7	110.0	58,420,451	64,104,944	9.7%
Post-Comp. Cooler (PC7)	145.2	110.0	28,473,213	27,634,282	-2.9%
TOTAL			231,359,422	236,973,695	2.4%

Table 28. Unit Operation Results for Integrally Geared 1 – 149F Compressor Compared to a "No-Capture" Plant.

Station Service Power	Unit	MEA Capture & IG 1 – 149F	No Capture	Diff.
Fan Power	[kW]	18,405	11,576	59.0%
Mill Power	[kW]	3,456	3,456	0.0%
Pump Power	[kW]	2,497	1,658	50.6%
Auxiliary Power	[kW]	15,000	15,000	0.0%
Total P_{ss}	[kW]	39,358	31,689	24.2%
Compressor Performance				
Compressor Power	[kW]	43,182	N/A	N/A
Plant Performance				
Qcoal	[Btu/hr]	5,501,978,582	5,501,978,582	0.0%
Net Power	[kW]	396,389	594,543	-33.3%
Net Unit HR	[Btu/kWh]	13,880	9,254	50.0%
Unit Efficiency	[%]	24.58%	36.87%	-12.3%

Table 29. Unit Operation Results for Integrally Geared 1 – 110F Compressor Compared to a "No-Capture" Plant.

Station Service Power	Unit	MEA Capture & IG 1 – 110F	No Capture	Diff.
Fan Power	[kW]	18,405	11,576	59.0%
Mill Power	[kW]	3,456	3,456	0.0%
Pump Power	[kW]	2,538	1,658	53.1%
Auxiliary Power	[kW]	15,000	15,000	0.0%
Total P_{ss}	[kW]	39,399	31,689	24.3%
Compressor Performance				
Compressor Power	[kW]	37,496	N/A	N/A
Plant Performance				
Qcoal	[Btu/hr]	5,501,978,582	5,501,978,582	0.0%
Net Power	[kW]	402,033	594,543	-32.4%
Net Unit HR	[Btu/kWh]	13,685	9,254	47.9%
Unit Efficiency	[%]	24.93%	36.87%	-11.9%

2.4.3 Effect of Carbon Capture & Compression on Unit Performance

The same MEA capture system was paired with each of four separate compressor options. The MEA system affects the gross power, fan power and pump power. The compressor affects the pump power and net power. The overall unit efficiency and heat rate depend on the combination of capture system and compressor. Thus the final results are slightly different for each compressor. After the MEA system is installed, the fan power increases by 59.0% and the turbine cycle produces 23.5% less gross power than the baseline plant. The pump power increases by 49.0% to 53.1% due to additional pumps required for the MEA system, and for each compressor stage's cooler. The compressors with the most stages (IG1 – 149 & IG1 – 110) have a higher pump power than the Ramgen and Inline 4 compressors. The turbine cycle heat rate increases by 30.8% with the addition of the MEA system.

The net power is reduced by 32.4% to 33.7%, depending on the compressor selected. The Ramgen compressor uses the most power at 45.6 MW. The Inline 4 compressor uses 43.9 MW of power. The integrally geared 1 compressors use the least amount of power, with 43.2 MW for IG1 – 149 and 37.5 MW for IG1 – 110. The resulting unit efficiency is reduced by 11.9 to 12.4 percentage points, among all of the compressors. The IG1 – 110 base case has the highest unit efficiency at 24.93% with the IG1 – 149 next best at 24.58%. The Inline 4 and Ramgen base cases have unit efficiencies of 24.54% and 24.43%, respectively. The base case with the lowest compressor power has the highest unit efficiency and vice versa.

Table 30. Unit Performance Results of MEA Capture with Four Compressor Options Compared to No Capture.

Boiler Performance	Unit	MEA with Ramgen	Diff.	MEA with Inline 4	Diff.	MEA with IG1-149	Diff.	MEA with IG 1-110	Diff.	No Capture
Q Steam	[MBtu/hr]	4,776	0.0%	4,776	0.0%	4,776	0.0%	4,776	0.0%	4,776
Coal Flow Rate	[lb/hr]	652,976	0.0%	652,976	0.0%	652,976	0.0%	652,976	0.0%	652,976
HHV Coal	[Btu/lb]	8,426	0.0%	8,426	0.0%	8,426	0.0%	8,426	0.0%	8,426
Q Coal	[MBtu/hr]	5,502	0.0%	5,502	0.0%	5,502	0.0%	5,502	0.0%	5,502
Boiler Efficiency	[%]	86.81%	0.0%	86.81%	0.0%	86.81%	0.0%	86.81%	0.0%	86.81%
Turbine Power										
HP Power	[kW]	183,314	0.0%	183,314	0.0%	183,314	0.0%	183,314	0.0%	183,314
IP1 Power	[kW]	106,165	0.0%	106,165	0.0%	106,165	0.0%	106,165	0.0%	106,165
IP2 Power	[kW]	65,888	0.0%	65,888	0.0%	65,888	0.0%	65,888	0.0%	65,888
LP1 Power	[kW]	57,559	0.0%	57,559	0.0%	57,559	0.0%	57,559	0.0%	57,559
LP2 Power	[kW]	31,476	-63.0%	31,476	-63.0%	31,476	-63.0%	31,476	-63.0%	84,977
LP3 Power	[kW]	13,513	-65.5%	13,513	-65.5%	13,513	-65.5%	13,513	-65.5%	39,182
LP4 Power	[kW]	13,547	-68.4%	13,547	-68.4%	13,547	-68.4%	13,547	-68.4%	42,908
LP5 Power	[kW]	14,758	-73.5%	14,758	-73.5%	14,758	-73.5%	14,758	-73.5%	55,774
Gross Power	[kW]	486,221	-23.5%	486,222	-23.5%	486,222	-23.5%	486,222	-23.5%	635,768
Gross Power minus Gen. Losses	[kW]	478,927	-23.5%	478,929	-23.5%	478,929	-23.5%	478,929	-23.5%	626,232
Turbine Cycle HR	[Btu/kWh]	9,973	30.8%	9,973	30.8%	9,973	30.8%	9,973	30.8%	7,627
Station Service Power										
Fan Power	[kW]	18,405	59.0%	18,405	59.0%	18,405	59.0%	18,405	59.0%	11,576
Mill Power	[kW]	3,456	0.0%	3,456	0.0%	3,456	0.0%	3,456	0.0%	3,456
Pump Power	[kW]	2,471	49.0%	2,471	49.0%	2,497	50.6%	2,538	53.1%	1,658
Auxiliary Power	[kW]	15,000	0.0%	15,000	0.0%	15,000	0.0%	15,000	0.0%	15,000
Total P_{ss}	[kW]	39,332	24.1%	39,332	24.1%	39,358	24.2%	39,399	24.3%	31,689
Compressor Performance										
Compressor Power	[kW]	45,593	N/A	43,905	N/A	43,182	N/A	37,496	N/A	N/A
Plant Performance										
Net Power	[kW]	394,003	-33.7%	395,691	-33.3%	396,389	-33.3%	402,033	-32.4%	594,543
Net Unit HR	[Btu/kWh]	13,964	50.9%	13,905	49.9%	13,880	50.0%	13,685	47.9%	9,254
Unit Efficiency	[%]	24.43%	-12.4%	24.54%	-12.3%	24.58%	-12.3%	24.93%	-11.9%	36.87%

3.0 Thermal Integration Cases

This analysis was set up to model the benefits of thermal integration. There are various thermal sources in the plant that normally exhaust waste heat. They can be matched with different thermal sinks in the plant to improve plant performance. The thermal sinks normally draw on steam from the turbine cycle which reduces the gross and net power produced. Thermal integration will reduce the cost of carbon capture by recycling heat rather than using steam heat.

Possible thermal sources are the compressor coolers, stripper condenser, lean amine cooler and flue gas cooler, and the first two are evaluated in this study. The thermal sources from the compressor coolers are shown in Table 19, Table 22, Table 26, and Table 27. The thermal sources in the plant, independent of the compressor selected, are shown in Table 31. Each of these heat sources gives off heat that would normally be removed by utilizing air or water-cooled cooling towers or heat exchangers. If this heat is re-directed to a thermal sink within the plant, less steam needs to be extracted from the turbine cycle, thus increasing the gross power.

Table 31. Plant Thermal Sources: Design Temperatures and Maximum Heat Available.

Heat Source	Design Material Inlet Temp	Design Material Outlet Temp	Maximum Design Duty Available
	[F]	[F]	[Btu/hr]
Stripper Condenser	239.9	100.0	489,940,816
Lean Amine Cooler	148.1	100.0	1,036,350,050
Flue Gas Cooler	135.0	100.0	511,780,705
TOTAL			2,038,071,571

The thermal sinks considered in this study include the stripper reboiler, feedwater heaters, and a coal dryer prior to the boiler (Table 32). The locations of the steam extractions for the reboiler and FWHs are shown in Figure 14. Dried coal may be either purchased at a higher cost, or dried on site with an available heat source in a coal dryer.

In Table 32, the "thermal energy to sink" is the quantity of energy provided to the sink by the basic turbine cycle heat source in the row. The *Aspen Plus* model provides enthalpy values for the steam extractions entering and exiting the thermal sink. The thermal energy values for the heat sources were obtained using Equation (49). The design duty for each thermal sink is determined by totaling the thermal energy from each heat source.

$$Q = \dot{m}_{heat\ source} * (h_{in} - h_{out}) \ \left[\frac{Btu}{hr}\right] \tag{49}$$

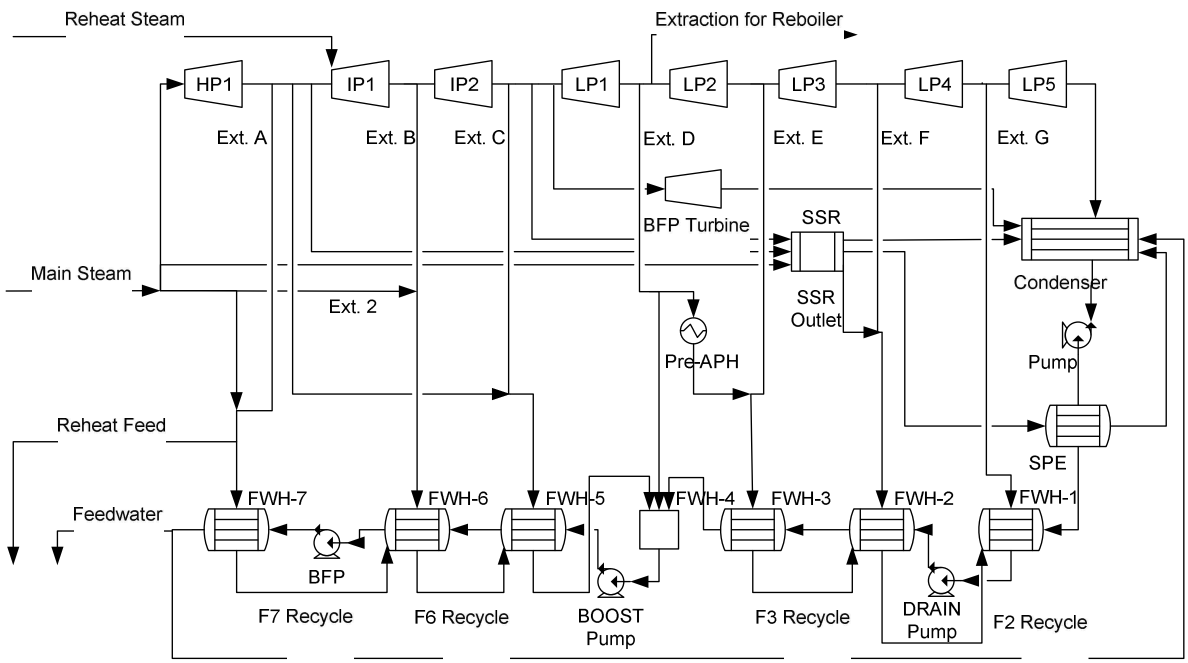


Figure 14. Supercritical Turbine Cycle Extraction Locations.

Table 32. Design Temperatures and Thermal Duties of Thermal Sinks.

Thermal Sink	Design Turbine Cycle Heat Sources	Steam Flow Rate	Steam Temp.	Thermal Energy to Sink	Design Duty of Thermal Sink	Design Temp. for Exiting Feedwater or Stripper Bottoms
		[lb/hr]	[F]	[Btu/hr]	[Btu/hr]	[F]
FWH-1	LP4 Extraction (Ext. G)	178,947	159.4	171,175,153	173,265,266	151.9
	(F2 recycle[1])			2,089,801		
FWH-2	LP3 Extraction (Ext. F)	114,535	201.8	115,328,040	130,649,607	193.8
	SSR Outlet	6,218	557.9	7,286,259		
	(F3 recycle[1])			8,035,053		
FWH-3	LP2 Extraction (Ext. E)	109,004	294.3	110,796,898	120,223,508	231.4
	Pre-APH out (LP1)	90,863	239.9	9,426,582		
FWH-5	IP2 Extraction (Ext. C)	163,004	656.8	172,739,088	215,945,146	363.6
	HP1 Extraction (5)	12,924	582.5	12,528,206		
	(F6 recycle[1])			30,677,594		
FWH-6	IP1 Extraction (Ext. B)	184,533	791.9	198,100,420	228,582,814	414.2
	Main steam (Ext. 2)	1,543	1,000.0	1,653,333		
	(F7 recycle[1])			28,828,969		
Feedwater Heater Sub-Total					868,666,341	
Stripper Reboiler	Between LP1 & LP2	1,761,410	522.0	1,798,889,282	1,798,887,740	269.7
TOTAL Design Duties of Thermal Sinks					2,667,554,081	
Coal Drying					Varies	Varies

[1] The term "recycle" refers to the condensate after a turbine steam extraction has passed through a FWH, which is directed to another FWH to be used as a thermal source.

All of the thermal sources and sinks vary in their temperature and amount of heat available or required. In order for thermal integration to occur, the thermal source must be carefully matched to the thermal sink in both temperature and quantity of heat. This analysis evaluates possible combinations between the thermal sources and sinks, to the extent allowed by the laws of thermodynamics and heat transfer. There are numerous possible combinations, and methods, for thermal integration. For each thermal sink, an educated guess was made about the best thermal source based on the source temperature and the maximum quantity of heat available. Only this match was analyzed in detail due to time constraints.

In an actual plant, many site-specific circumstances must be taken into account in the thermal integration design process including the design and size of the heat exchangers, space for the cooling water loop that connects thermal source to sink, the quantity of heat available compared to the quantity of heat required and the temperature of the thermal source compared to the sink. This analysis focused on process methods to achieve thermal integration and predict its benefits as a first step in the decision making process. The next steps in the process would be to select cases that predict large benefits, estimate the size and cost of the equipment needed and then evaluate the detailed costs and benefits. Once the appropriate case is chosen, the necessary equipment can be installed. The latter steps are beyond the scope of this study.

3.1 Compressor Coolers to Stripper Reboiler

The first thermal integration case uses heat generated during the compression process to offset the heat required for the stripper reboiler. The stripper reboiler is the single largest thermal sink in an MEA carbon capture plant, requiring more than twice as much heat as all of the FWHs combined (Table 32). Offsetting any fraction of this heat would yield benefits to the plant. In order to describe the thermal integration case, it will be helpful to describe the design of a typical reboiler. The purpose of the reboiler was described in Section 2.3.2.

A reboiler is essentially a shell-and-tube heat exchanger. The liquid stripper bottoms enter the bottom of the shell, while the heat source (i.e. steam or hot water) enters the tubes. As the stripper bottoms pass over the hot tubes, the liquid is partially boiled, causing the MEA to release the CO_2 as a gas. The gas, mostly CO_2, rises to the top of the reboiler and re-enters the stripper column. The liquid, mostly MEA and water, exits the bottom of the reboiler after which it is cooled and used again in the carbon capture process.

To achieve thermal integration, the reboiler would need to be designed with additional tube bundles, one for each thermal source used (Figure 15). In terms of process flow, the stripper bottoms would first encounter the tube bundle(s) for the integrated thermal source(s) to utilize as much recycled heat as possible. Then the stripper bottoms would pass over the tubes containing the steam extraction, making up the balance of the heat transfer. The steam extraction would be reduced by the total quantity of thermal energy provided by the other thermal source(s).

The stripper reboiler operates at 269.7°F. This temperature limits the amount of heat transfer from the thermal source(s) because only heat available above this temperature can be transferred to the stripper bottoms. Otherwise the stripper bottoms would be heating the "thermal source". There will still be heat remaining in the thermal source(s) at temperatures below the reboiler temperature, which could be used as a heat source elsewhere. The ratio of heat used in the reboiler to the total heat available in all of the compressor coolers is reported in the results.

The results for each compressor coolers-to-reboiler thermal integration case are shown and discussed below. With the Ramgen and Inline 4 compressors, thermal integration could be achieved with the hot water streams from both coolers so a total of three tube bundles were used in the reboiler model (Figure 15). For the IG1-149, thermal integration could be achieved with only one compressor heat source (PC7), so a total of two tube bundles were used in the reboiler model (Figure 16).

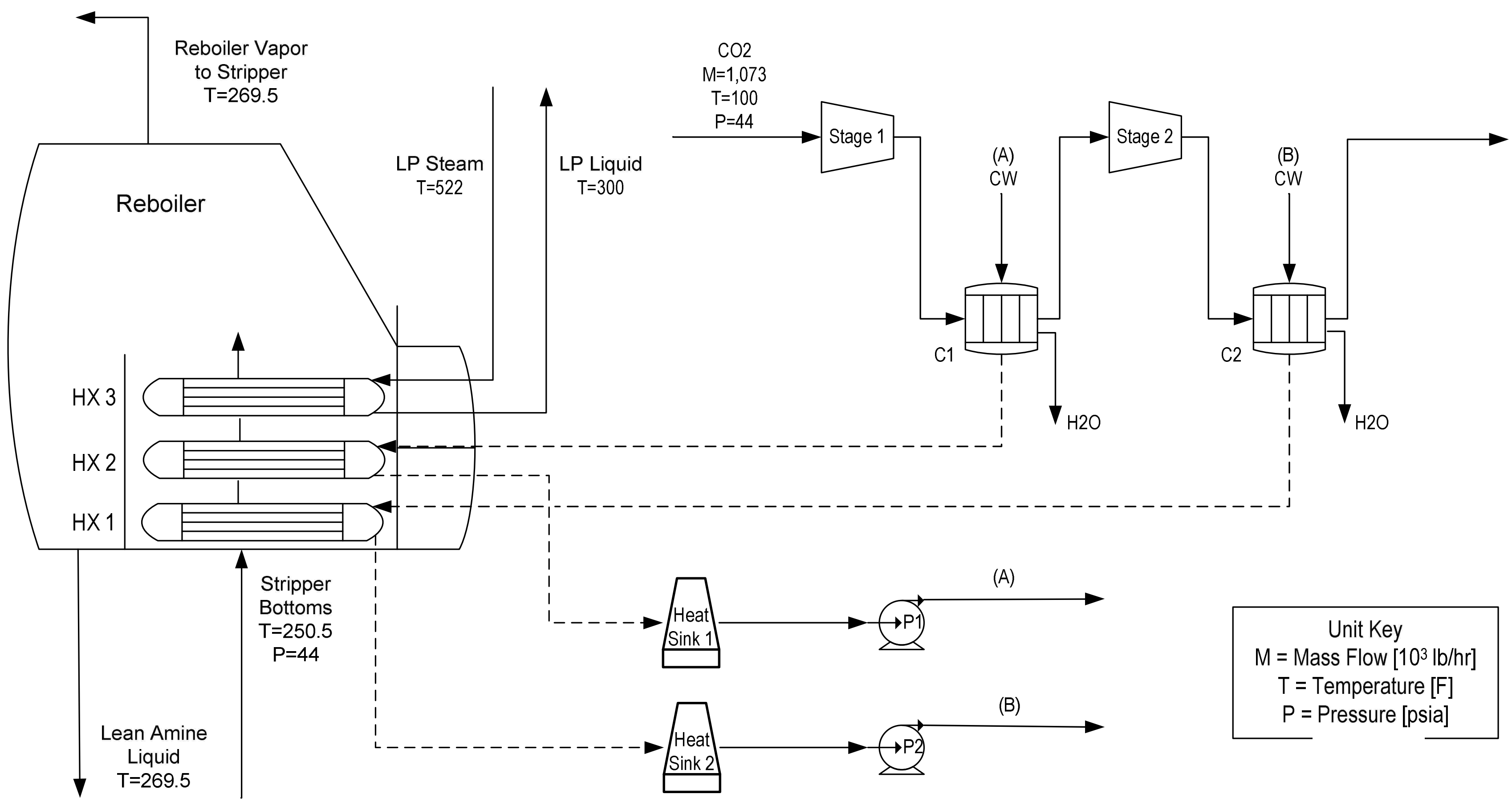


Figure 15. Compressor Coolers to Reboiler Thermal Integration for Ramgen and Inline 4 Compressors.

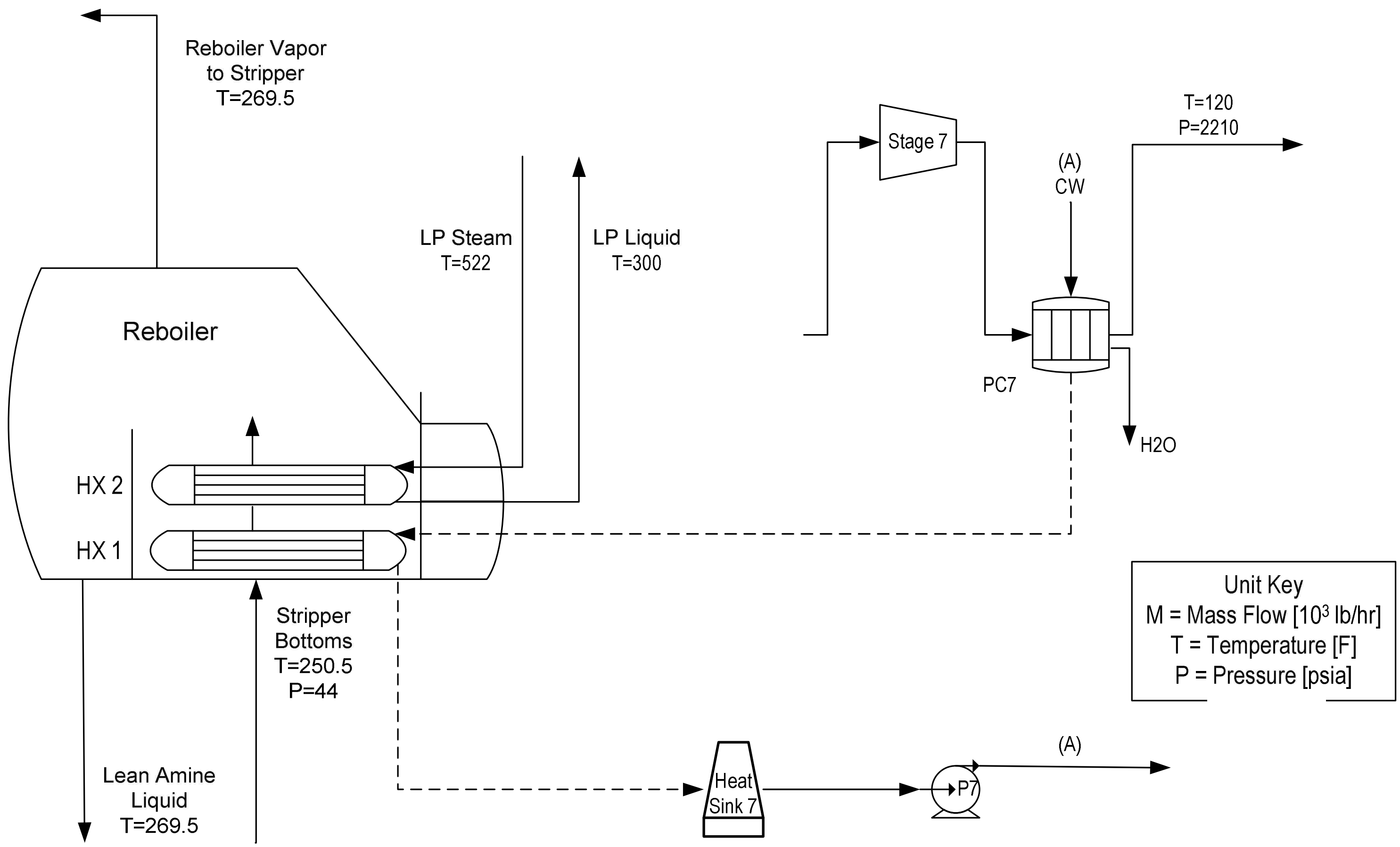


Figure 16. Compressor Coolers to Reboiler Thermal Integration for IG1 – 149 and IG1 – 110 Compressors.

3.1.1 RAMGEN Compressor to Stripper Reboiler Results

When the Ramgen compressor's coolers are used for thermal integration to the stripper reboiler, both the intercooler (IC1) and the post-compressor cooler (PC2) can be used. The integrated heat offsets 7.7% of the total reboiler thermal duty. Since the energy in the steam extraction is proportional to its mass flow, the steam extraction is also reduced by 7.7% (Table 33). This quantity of heat represents 53.5% of the total heat available in the compressor, for the case of a combined flow rate of cooling water through both coolers of 727,445 lb/hr. In other words, only 53.5% of the total thermal energy in the hot cooling water streams from both compressor coolers is able to be transferred to the reboiler due to the temperature. The remaining heat (46.5%) can be used as a thermal source elsewhere.

After the hot cooling water passes through the reboiler, it must be cooled (or integrated elsewhere) and pumped to return it to the cooler inlet specifications. This increases the pump power slightly compared to the baseline case without thermal integration. An additional 11,287 kW of net power is produced by the plant and the heat rate is decreased by 389 Btu/kWh.

Table 33. Ramgen Compressor to Stripper Reboiler Thermal Integration Results.

	Unit	MEA HI: IC1 & PC2 to Reboiler	MEA w/o heat integration
Reduction in Extraction	[%]	7.7%	0.0%
Compressor Heat Used	[%]	53.5%	0.0%
FG Flow to Compressor	[lb/hr]	1,073,080	1,073,080
Gross Power	[kW]	497,694	486,221
Gross Power minus Gen. Losses	[kW]	490,228	478,927
Turbine Cycle Heat Rate	[Btu/kWh]	9,743	9,973
Fan Power	[kW]	18,405	18,405
Pulv. Power	[kW]	3,456	3,456
Pump Power	[kW]	2,484	2,471
Aux. Power	[kW]	15,000	15,000
Total Pss	[kW]	39,346	39,332
Compressor Power	[kW]	45,594	45,593
Net Power	[kW]	405,289	394,003
Power Increase	[kW]	11,287	N/A
Net Unit Heat Rate	[Btu/kWh]	13,575	13,964
HR Improvement	[%]	-2.78%	0.00%
Unit Efficiency	[%]	25.13%	24.43%

3.1.2 Inline 4 Compressor to Stripper Reboiler Results

For the Inline 4 compressor's coolers to stripper reboiler thermal integration case, both of the intercoolers (IC1 and IC2) can be integrated. The integrated heat from the compressor intercoolers offsets 6.7% of the total reboiler heat duty and reduces the steam extraction mass flow rate by 6.7% as well (Table 34). This quantity of heat represents 49.7% of the heat available in both compressor coolers, for the case of a combined flow rate of cooling water through both coolers of 714,637 lb/hr. After the hot cooling water passes through the reboiler, it must be cooled and pumped to return it to the cooler inlet specifications. The pump power increases slightly compared to the case without thermal integration. An additional 9,873 kW of net power is produced by the plant and the heat rate decreases by 339 Btu/kWh, or 2.43%.

Table 34. Inline 4 Compressor to Stripper Reboiler Thermal Integration Results.

	Unit	MEA HI: IC1+IC2 to Reboiler	MEA w/o heat integration
Reduction in Extraction	[%]	6.7%	0.0%
Compressor Heat Used	[%]	49.7%	0.0%
FG Flow to Compressor	[lb/hr]	1,073,080	1,073,080
Gross Power	[kW]	496,284	486,222
Gross Power minus Gen. Losses	[kW]	488,840	478,929
Turbine Cycle Heat Rate	[Btu/kWh]	9,771	9,973
Fan Power	[kW]	18,405	18,405
Pulv. Power	[kW]	3,456	3,456
Pump Power	[kW]	2,484	2,471
Aux. Power	[kW]	15,000	15,000
Total Pss	[kW]	39,345	39,332
Compressor Power	[kW]	43,905	43,905
Net Power	[kW]	405,590	395,691
Power Increase	[kW]	9,873	N/A
Net Unit Heat Rate	[Btu/kWh]	13,565	13,905
HR Improvement	[%]	-2.43%	0.00%
Unit Efficiency	[%]	25.15%	24.54%

3.1.3 Integrally Geared 1 – 149 Compressor to Stripper Reboiler Results

For the IG1-149, the hot cooling water from the post-compressor cooler (PC7) was the only compressor heat source which could be used for the thermal integration case to the stripper reboiler. PC7 also has the largest maximum heat available from the compressor's thermal sources (Table 26). Thus, a total of two sets of tubes were used within the reboiler's shell.

The thermal energy in the hot cooling water stream from PC7 offset 1.0% of the total reboiler heat duty and reduced the steam extraction mass flow by 1.0% as well (Table 35). This quantity of heat represents 7.3% of the total heat available in all of the compressor coolers, including those coolers for which it is not possible to integrate the hot cooling water streams to the reboiler. The combined flow rate of cooling water through all of the compressor coolers is 1,396,326 lb/hr. This compressor, as a whole, has more thermal energy available to integrate elsewhere than the Ramgen and Inline 4 compressors.

After the hot cooling water passes through the reboiler, it must be cooled and pumped before being returned to the cooler as cooling water. The pump power increases slightly compared to the baseline case. Overall, an additional 1,492 kW of net power is produced by the plant. The heat rate decreases by 52 Btu/kWh, or 0.37% less than the baseline case. The heat rate improvement for this integration case is small because the temperature difference between the hot cooling water from PC7 and the reboiler is relatively small (31°F).

Table 35. Integrally Geared 1 – 149 Compressor to Stripper Reboiler Thermal Integration Results.

	Unit	MEA HI: PC7 to Reboiler	MEA w/o heat integration
Reduction in Extraction	[%]	1.0%	0.0%
Compressor Heat Used	[%]	7.3%	0.0%
FG Flow to Compressor	[lb/hr]	1,073,080	1,073,080
Gross Power	[kW]	487,746	486,222
Gross Power minus Gen. Losses	[kW]	480,430	478,929
Turbine Cycle Heat Rate	[Btu/kWh]	9,942	9,973
Fan Power	[kW]	18,405	18,405
Pulv. Power	[kW]	3,456	3,456
Pump Power	[kW]	2,507	2,497
Aux. Power	[kW]	15,000	15,000
Total Pss	[kW]	39,368	39,358
Compressor Power	[kW]	43,182	43,182
Net Power	[kW]	397,880	396,388
Power Increase	[kW]	1,492	N/A
Net Unit Heat Rate	[Btu/kWh]	13,828	13,880
HR Improvement	[%]	-0.37%	0.00%
Unit Efficiency	[%]	24.67%	24.58%

3.1.4 Integrally Geared 1 – 110 Compressor to Stripper Reboiler Results

It was not possible to integrate heat from the IG1-110 compressor to the reboiler because the CO_2 inlet temperatures to the coolers were lower than the reboiler temperature (Table 27). However, the heat rate of the plant using IG1-110 without thermal integration (13,699 Btu/kWh) is still lower than the heat rate of IG1-149 with thermal integration to the reboiler (13,832 Btu/kWh).

3.2 Compressor Coolers to FWHs

The second thermal integration case evaluated the use of heat from the compressor coolers to offset steam extractions for the feedwater heaters (FWHs). As discussed in Section 2.1, the FWHs are heat exchangers which use steam extractions from different turbines in the steam turbine cycle to heat the boiler feedwater. Integrating heat from the compressor coolers to the FWHs allows some of these steam extractions to be reduced or eliminated. This allows more steam to flow through the low pressure turbines, which generates more power. The first three

FWHs (FWH-1, FWH-2 and FWH-3) have low enough feedwater exit temperatures that heat from the compressor coolers can be used to offset the steam extraction requirements. The locations and mass flow rates (in units of 10^3 lb/hr) of the steam extractions in the turbine kit design for FWH-1 through FWH-4 are shown in Figure 17. The exact values are also listed in Table 32.

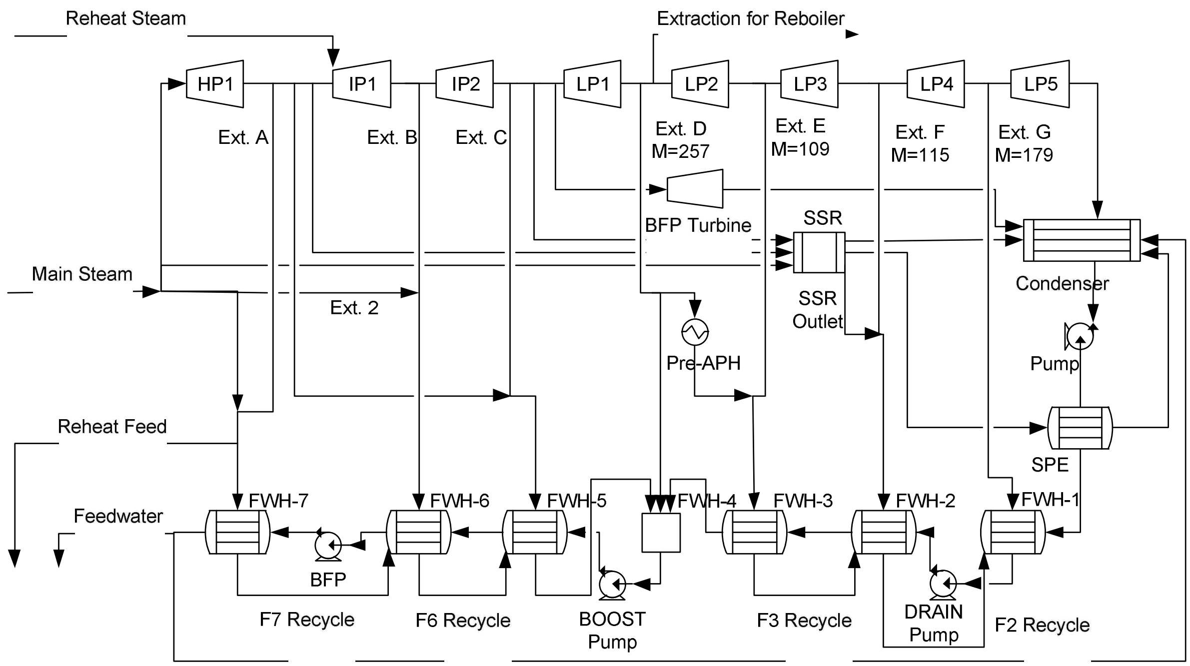


Figure 17. Steam turbine cycle showing mass flow rates of FWH steam extractions.

The process and results for integrating compressor cooler heat to FWH-1, FWH-2, and FWH-3 are presented. Then, the method and results are presented for the other compressors.

3.2.1 <u>RAMGEN Compressor to FWHs Results</u>

The hot cooling water from the Ramgen post-compressor cooler (PC2) was integrated to FWH-1, which has the lowest feedwater exit temperature. The process for the integration case to FWH-1 is illustrated in Figure 18. PC2 was selected because it had the highest CO_2 inlet temperatures and the largest maximum quantity of heat available. The F2 recycle stream and extraction G normally provide the thermal energy to FWH-1 (Figure 17).

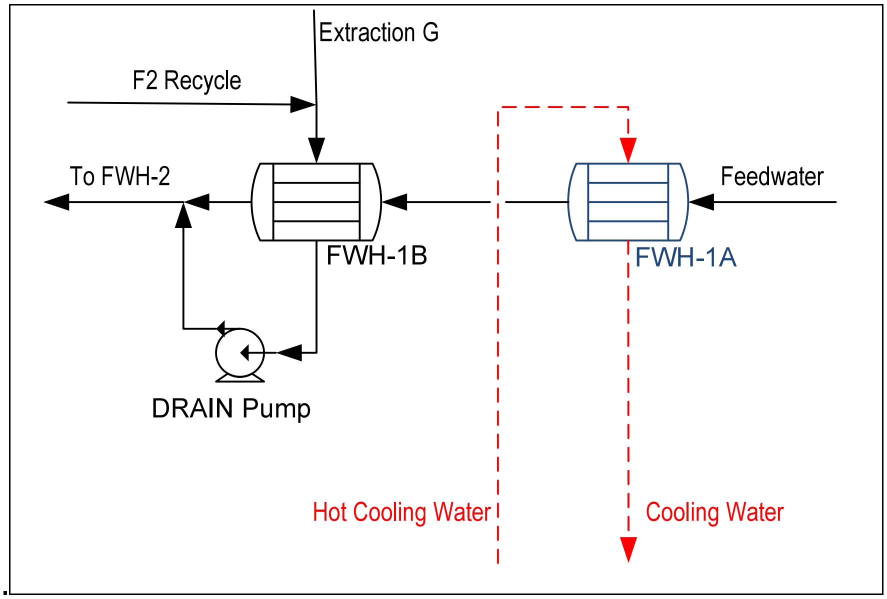


Figure 18. Compressor to FWH-1 Thermal Integration Process, Extraction G reduced.

In this case, the hot cooling water stream does not have sufficient thermal energy to eliminate extraction G. However, steam extraction G can be reduced. In order to use the available heat to offset the heat duty of FHW-1, the feedwater heater is divided into two heat exchangers. The first heat exchanger, FWH-1A, is used to transfer as much thermal energy as possible from the hot cooling water to the feedwater. This heat exchanger can be designed so that the hot cooling water exits the heat exchanger at 100°F, so that it is at the proper temperature to return to the compressor coolers for use as cooling water. The second heat exchanger, FWH-1B, uses the reduced steam extraction G to heat the feedwater to the designated exit temperature. Less extraction steam is required for FWH-1B than was originally required for FWH-1. Then, a pump is required for the cooling water after exiting FWH-1A to increase the pressure before the cooling water is returned to the compressor cooler.

As another option, the hot cooling water from the Ramgen post-compressor cooler (PC2) was integrated to FWH-2. The process for the integration case to FWH-2 is illustrated in Figure 19. As stated in the FWH-1 integration process, PC2 was selected because it had the highest CO_2 inlet temperatures and the largest maximum quantity of heat available. In addition, selecting the same thermal source as the other FWH integration cases makes it easier to compare the results.

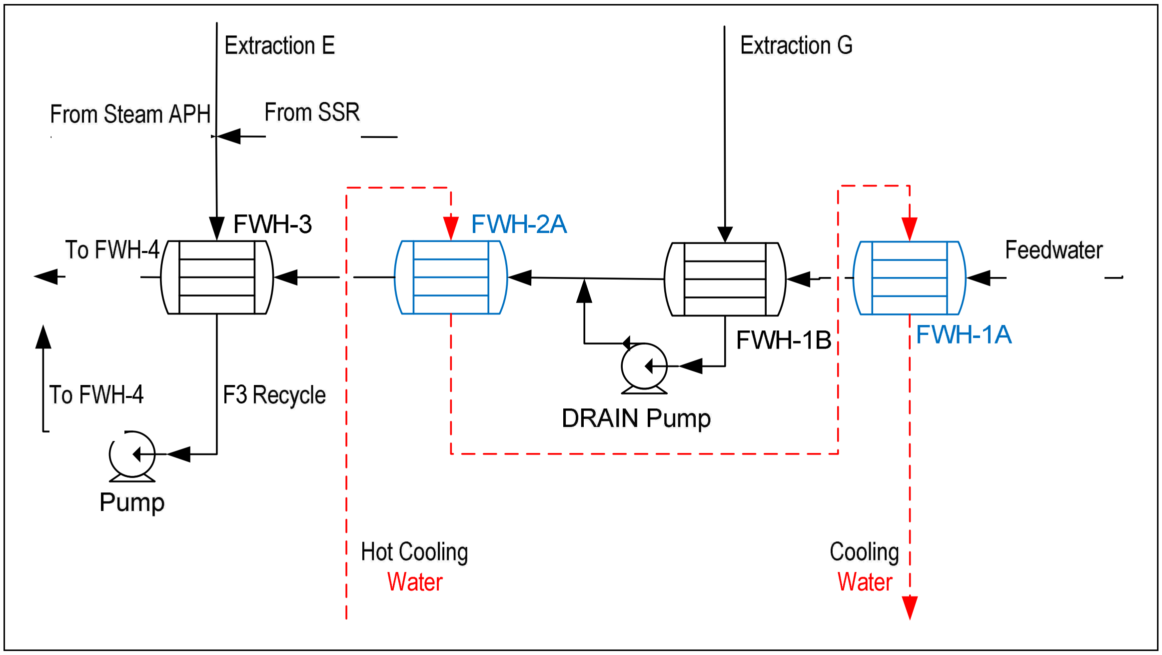


Figure 19. Compressor to FWH-2 Thermal Integration Process, Extraction F eliminated[2].

In this integration case, the hot cooling water stream has sufficient thermal energy to eliminate the need for the steam extraction normally required for FWH-2 (Extraction F). As discussed in Section 2.1, the steam extraction that normally passes through FWH-2 is sent to enter FWH-1, since there is still heat available in that stream (Figure 17). This cascading process was also repeated with the hot cooling water stream so that it passed through FWH-2 and then through FWH-1 in order to use the remaining heat available in the stream. In order to cascade the hot cooling water to FWH-1, two heat exchangers for FWH-1 are required, one for the hot cooling water and one for the steam extraction. This two heat exchanger design is similar to the design for integration to FWH-1, described above. Again, FWH-1A was designed to have the hot cooling water exit this heat exchanger at 100°F. The pressure drop of the hot cooling water increases since it passes through two heat exchangers, so the power requirement of the cooling water pump increases.

[2] The term "recycle" refers to the condensate after a turbine steam extraction has passed through a FWH, which is directed to another FWH to be used as a thermal source.

Since the integrated heat is enough to eliminate the 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 to FWH-1A to use the remaining heat available in that stream. A portion of the discharge from the SSR, along with the F3 recycle[3] stream, normally combine with the extraction for FWH-2 to provide thermal energy to FWH-2 (Figure 17). When the extraction for FWH-2 is eliminated, the F3 recycle stream is pumped and then re-directed to FWH-4. In addition, the SSR discharge stream is mixed with the extraction for FWH-3 and sent through that FWH instead.

The steam extraction for FWH-3 can be reduced because FWH-3 now has excess heat because the SSR discharge stream is sent through that FWH, described above. The steam extraction for FWH-2 can be eliminated because of the integration of hot cooling water. Also, the hot cooling water exiting FWH-2 is then sent to FWH-1A thus reducing the steam extraction required for FWH-1. Since the cooling water exits FWH-1A at 100°F, only a pump is required to elevate the pressure of the cooling water before it's returned to the compressor cooler.

As another option, the hot cooling water was integrated to FWH-3. This process is illustrated in Figure 20. As stated in the FWH-1 integration process, PC2 was selected because it had the highest CO_2 inlet temperatures and the largest quantity of heat available (Table 19).

[3]The term "recycle" refers to the condensate after a turbine steam extraction has passed through a FWH, which is directed to another FWH to be used as a thermal source.

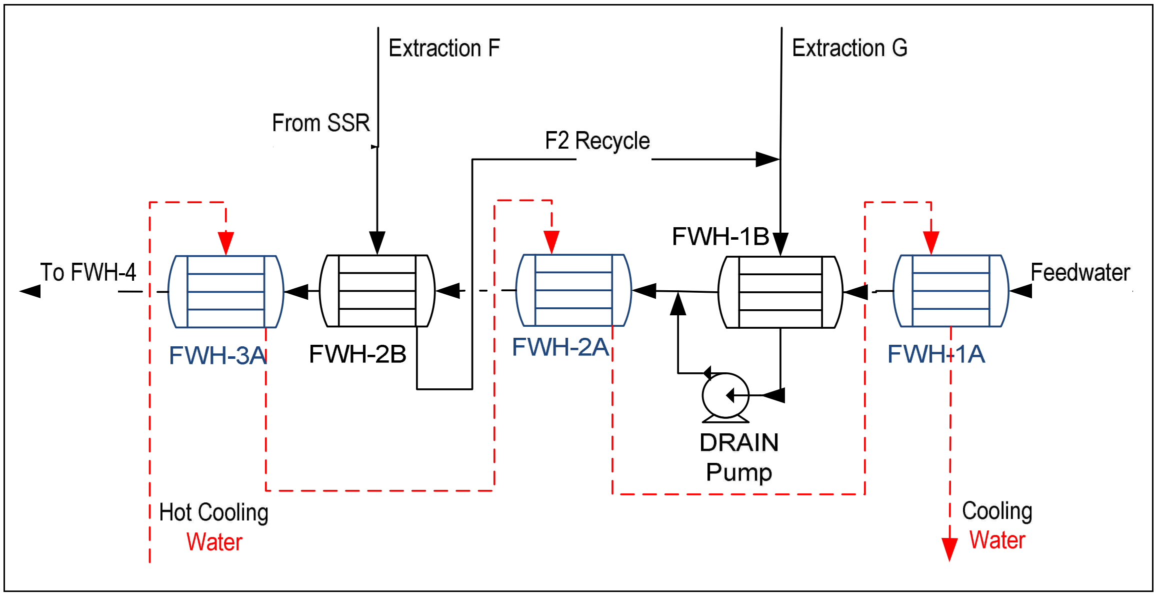


Figure 20. Compressor to FWH-3 Thermal Integration Process, Extraction E eliminated.

There was sufficient heat available in the hot cooling water to eliminate the need for the steam extraction normally required for FWH-3 (Extraction E). Since the steam extraction is eliminated, only one heat exchanger is used for the hot cooling water to heat the feedwater. The discharge from the Steam Air Pre-heater (Pre-APH) normally combines with extraction E to provide thermal energy to the feedwater in FWH-3 (Figure 17). When extraction E is eliminated, the Pre-APH discharge stream was re-directed to enter FWH-4.

Similar to the integration to FWH-2, the hot cooling water exiting FWH-3 is cascaded to enter FWH-2A and then FWH-1A to use any remaining heat available in the stream. Since the hot cooling water is used in three heat exchangers, the pressure drop is greater and the pump power required for the cooling water is also greater. The steam extractions for FWH-2 and FWH-1 can be reduced because the hot cooling water was used in FWH-2A and FWH-1A. The cooling water stream leaves FWH-1A at 100°F, so only a pump is required to elevate the pressure of the cooling water before it's returned to the compressor coolers.

The results for the Ramgen post-compressor cooler thermal integrations to FWH-1, FWH-2 and FWH-3 are shown in Table 36 and in Figure 58, Figure 59 and Figure 60 respectively (in Appendix F). In the thermal integration to FWH-1 (Figure 58), extraction G was reduced by

92.7% by weight. The actual mass flows are shown in Figure 58. This represented 65.5% of the total compressor cooler duties available, with a combined flow rate of cooling water through both coolers of 727,434 lb/hr. The net power increased by 3,805 kW, while the heat rate decreased by 134 Btu/kWh, which is 0.96% less than the base case.

In the thermal integration to FWH-2 (Figure 59), extraction E was reduced by 13.8%, extraction F was eliminated and extraction G was reduced by 22.3% by weight. The integrated heat represents 65.5% of the total compressor cooler duties available, with a combined flow rate of cooling water through both coolers of 727,434 lb/hr. An additional 6,170 kW of net power was produced and the heat rate decreased by 215 Btu/kWh, or 1.54%.

In the thermal integration to FWH-3 (Figure 60), extraction E was eliminated, extraction F was reduced by 16.2% and extraction G was reduced by 96.5% by weight. The thermal energy transferred from the hot cooling water to FWH-3 represents 65.5% of the total compressor cooler duties available, with a combined flow rate of cooling water through both coolers of 727,434 lb/hr. In other words, only 65.5% of the total thermal energy in the hot cooling water streams from both compressor coolers is transferred to FWH-3. The remaining heat (34.5%) can be used as a thermal source for another thermal sink. The reduction in extraction flow rates result in an additional 7,233 kW of net power production. The heat rate decreases by 252 Btu/kWh which is 1.80% below than the baseline case.

The FWH-3 integration case provides the greatest benefit to the plant compared to the integrations to FWH-2 and FWH-1. In the FWH-3 integration, additional steam flows through as many as three low pressure turbines. In the FWH-2 integration, additional steam flows through up to two of the turbines and, in the FWH-1 integration, additional steam flows through one turbine. Since the additional steam flows through more turbines in the FWH-3 case, the gross power is the highest for all of the cases (Table 36) which results in the highest net power and unit efficiency and the lowest heat rate. This is illustrated in Figure 17 (without thermal integration) and in Figure 60 (with thermal integration).

Table 36. Ramgen Post-Compressor Cooler to FWHs 3, 2, and 1 Thermal Integration Results.

	Unit	MEA HI: PC2 to FWH-3	MEA HI: PC2 to FWH-2	MEA HI: PC2 to FWH-1	MEA w/o Heat Integration
Reduction in Ext. E	[%]	100.0%	13.8%	0.0%	0.0%
Reduction in Ext. F	[%]	16.2%	100.0%	0.0%	0.0%
Reduction in Ext. G	[%]	96.5%	22.3%	92.7%	0.0%
Compressor Heat Used	[%]	65.5%	65.5%	65.5%	0.0%
FG Flow to Compressor	[lb/hr]	1,073,080	1,073,080	1,073,080	1,073,080
Gross Power	[kW]	493,627	492,546	490,086	486,222
Gross Power minus Gen. Losses	[kW]	486,222	485,158	482,735	478,929
Turbine Cycle Heat Rate	[Btu/kWh]	9,823	9,845	9,894	9,973
Fan Power	[kW]	18,405	18,405	18,405	18,405
Pulv. Power	[kW]	3,456	3,456	3,456	3,456
Pump Power	[kW]	2,532	2,530	2,472	2,471
Aux. Power	[kW]	15,000	15,000	15,000	15,000
Total Pss	[kW]	39,393	39,392	39,333	39,332
Compressor Power	[kW]	45,594	45,594	45,594	45,594
Net Power	[kW]	401,235	400,172	397,807	394,002
Net Unit Heat Rate	[Btu/kWh]	13,713	13,749	13,831	13,964
HR Improvement	[%]	-1.80%	-1.54%	-0.96%	0.00%
Unit Efficiency	[%]	24.88%	24.82%	24.67%	24.43%

3.2.2 Inline 4 Compressor to FWHs Results

The process for integrating the heat from the Inline 4 intercooler 2 (IC2) to FWH-1 is illustrated in Figure 18. The integration case to FWH-2 is illustrated in Figure 19 and the integration case to FWH-3 is illustrated in Figure 20. The processes are the same as described in the Ramgen cases (Section 3.2.1), except that the cooling water exits FWH-1A at 90°F in all of the Inline 4 integration cases. IC2 was selected for these thermal integration cases because it has the highest CO_2 inlet temperatures and the largest quantity of heat available (Table 22).

The results for thermally integrating the Inline 4 intercooler 2 (IC2) to FWH-1, FWH-2 and FWH-3 are shown in Table 37 and also in Figure 61, Figure 62 and Figure 63 respectively (in Appendix F). In the FWH-1 integration (Figure 61), extraction G was reduced by 86.6% by weight. The actual mass flows are shown in Figure 61. This represented 64.1% of the total compressor cooler duties available, with a combined flow rate of cooling water through both coolers of 714,627 lb/hr. The net power increases by 3,555 kW. The heat rate decreases by 124 Btu/kWh, which is 0.89% below the base case.

In the thermal integration to FWH-2 (Figure 62), extraction E was reduced by 13.8%, extraction F was eliminated and extraction G was reduced by 15.6% by weight. The integrated heat represents 64.1% of the total compressor cooler duties available, with a combined flow rate of cooling water through both coolers of 714,627 lb/hr. An additional 5,944 kW of net power is produced. The heat rate decreases by 206 Btu/kWh, which is 1.48%.

In the thermal integration to FWH-3 (Figure 63), extraction E was eliminated, extraction F was reduced by 10.1% and extraction G was reduced by 19.5% by weight. The integrated heat represents 64.1% of the total compressor cooler duties available, with a flow rate of cooling water through both coolers of 714,627 lb/hr. The net power increases by 7,098 kW. The heat rate decreases by 245 Btu/kWh, which is 1.76% less than the base case.

The Inline 4 IC2 to FWH-3 integration case provides the greatest benefit to the plant compared to the integrations to FWH-2 and FWH-1. This is due to the fact that additional steam flows through as many as three low pressure turbines and produces the highest gross power.

Table 37. Inline 4 Intercooler 2 (IC2) to Feedwater Heaters 1, 2, and 3 Thermal Integration Results.

	Unit	MEA HI: IC2 to FWH-3	MEA HI: IC2 to FWH-2	MEA HI: IC2 to FWH-1	MEA w/o heat integration
Reduction in Ext. E	[%]	100.0%	13.8%	0.0%	0.0%
Reduction in Ext. F	[%]	90.1%	100.0%	0.0%	0.0%
Reduction in Ext. G	[%]	19.5%	15.6%	86.6%	0.0%
Compressor Heat Used	[%]	64.1%	64.1%	64.1%	0.0%
FG Flow to Compressor	[lb/hr]	1,073,080	1,073,080	1,073,080	1,073,080
Gross Power	[kW]	493,440	492,267	489,830	486,222
Gross Power minus Gen. Losses	[kW]	486,038	484,883	482,482	478,929
Turbine Cycle Heat Rate	[Btu/kWh]	9,827	9,850	9,899	9,973
Fan Power	[kW]	18,405	18,405	18,405	18,405
Pulv. Power	[kW]	3,456	3,456	3,456	3,456
Pump Power	[kW]	2,483	2,482	2,475	2,471
Aux. Power	[kW]	15,000	15,000	15,000	15,000
Total Pss	[kW]	39,344	39,343	39,336	39,332
Compressor Power	[kW]	43,905	43,905	43,905	43,905
Net Power	[kW]	402,790	401,636	399,242	395,692
Net Unit Heat Rate	[Btu/kWh]	13,660	13,699	13,781	13,905
HR Improvement	[%]	-1.76%	-1.48%	-0.89%	0.00%
Unit Efficiency	[%]	24.98%	24.91%	24.76%	24.54%

3.2.3 Integrally Geared 1 – 149 Compressor to FWHs Results

The hot cooling water from the IG1 – 149 post-compressor cooler (PC7) was integrated to FWH-1. The process is the same as described in the Ramgen FWH-1 case (Section 3.2.1) except that FWH-1A is designed to have the hot cooling water exit the heat exchanger at 90°F. The process is illustrated in Figure 18. PC7 was selected because it had the highest CO_2 inlet temperatures and the largest maximum quantity of heat available.

As another option, the hot cooling water from the IG1 – 149 post-compressor cooler (PC7) was integrated to FWH-2. The process for the integration case to FWH-2 is illustrated in Figure 21. As stated in the FWH-1 integration process, PC7 was selected because it had the highest CO_2 inlet temperatures and the largest maximum quantity of heat available. In addition, selecting the same thermal source as the other FWH integration cases makes it easier to compare the results.

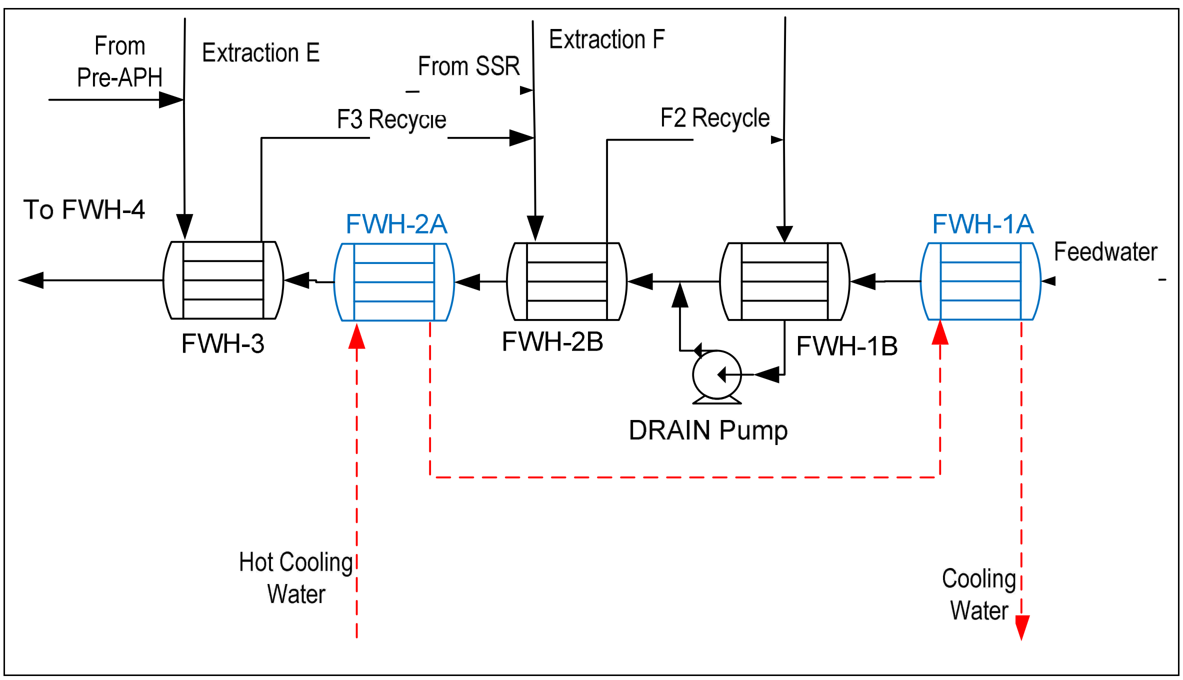


Figure 21. Compressor to FWH-2 Thermal Integration, Extraction F reduced.

In this integration case, the hot cooling water stream can only reduce the steam extraction normally required for FWH-2 (Extraction F). In order to use the available heat from the compressor to offset the heat duty of FHW-2, the feedwater heater is divided into two heat exchangers. The first heat exchanger, FWH-2B, is used to transfer thermal energy from the steam extraction to the feedwater. The second heat exchanger, FWH-2A, uses the hot cooling water from the compressor to heat the feedwater to the designated exit temperature. Less extraction steam is required for FWH-2B than was originally required for FWH-2 since part of the heat duty for FWH-2 was provided by the hot cooling water.

As discussed in Section 2.1, the steam extraction that normally passes through FWH-2 is sent to enter FWH-1, since there is still heat available in that stream, called F2 Recycle[4] (Figure 14). This cascading process was also repeated with the hot cooling water stream so that it passed through FWH-2A and then through FWH-1 in order to use the remaining heat available in the stream. The pressure drop of the hot cooling water increases since it passes through two heat exchangers, therefore the power requirement of the cooling water pump increases. In order to cascade the hot cooling water to FWH-1, two heat exchangers for FWH-1 are required, one for the hot cooling water and one for the steam extraction. This two heat exchanger design is similar to the design for integration to FWH-1, described in Section 3.2.1. However, FWH-1A was designed to have the hot cooling water exit this heat exchanger at 90°F. A pump is then required for the cooling water after it exits FWH-1A to increase the pressure before the cooling water is returned to the compressor cooler.

A portion of the discharge from the SSR, along with the FWH-3 recycle stream, normally combine with the extraction for FWH-2 to provide thermal energy to FWH-2 (Figure 14). These streams remain in the same location and provide thermal energy to the feedwater in FWH-2B.

As another option, the hot cooling water from the IG1 – 149 post-compressor cooler (PC7) was integrated to FWH-3. The process for this is illustrated in Figure 22. As stated in the IG1 – 149 FWH-1 integration process, PC7 was selected because it had the highest CO_2 inlet temperatures and the largest maximum quantity of heat available (Table 26).

[4] The term "recycle" refers to the condensate after a turbine steam extraction has passed through a FWH, which is directed to another FWH to be used as a thermal source.

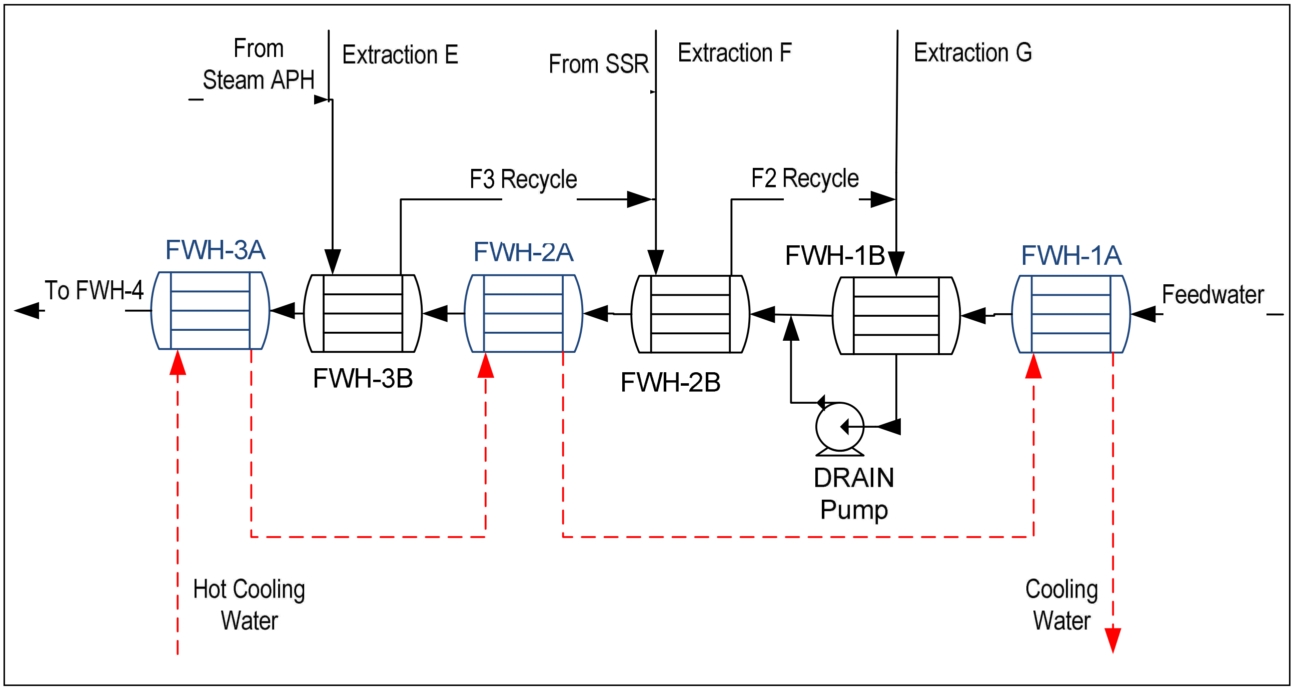


Figure 22. Compressor to FWH-3 Thermal Integration, Extraction E reduced.

In this integration case, the hot cooling water stream can only reduce the steam extraction normally required for FWH-3 (Extraction E). In order to use the available heat from the compressor to offset the heat duty of FHW-3, the feedwater heater is divided into two heat exchangers. The first heat exchanger, FWH-3B, is used to transfer thermal energy from the steam extraction to the feedwater. The second heat exchanger, FWH-3A, uses the hot cooling water from the compressor to heat the feedwater to the designated exit temperature. Less extraction steam is required for FWH-3B than was originally required for FWH-3 since part of the heat duty for FWH-3 was provided by the hot cooling water.

Similar to the integration to FWH-2, the hot cooling water exiting FWH-3A is cascaded, and enters FWH-2A and then FWH-1A to use any remaining heat available in the stream. Since the hot cooling water is used in three heat exchangers, the pressure drop is greater and the pump power required for the cooling water is also greater. The steam extractions for FWH-2 and FWH-1 can be reduced because the hot cooling water was used in FWH-2A and FWH-1A. The cooling water stream leaves FWH-1A at 90°F so it is at the proper temperature to be used as cooling water. A pump is then required to elevate the pressure of the cooling water before being returned to the compressor cooler. The discharge from the Steam Air Pre-heater (Pre-APH) normally combines with extraction E to provide thermal energy to the feedwater in FWH-3 (Figure 14). These streams remain in the same location and provide thermal energy to the feedwater in FWH-3B.

The results for thermally integrating the IG1-149 post-compressor cooler (PC7) to FWH-1, FWH-2 and FWH-3 are shown in Table 38 and also in Figure 64, Figure 65 and Figure 66, respectively (Appendix F). In the FWH-1 integration (Figure 64), extraction G was reduced by 63.7% by weight. This represented 45.7% of the total compressor cooler duties available, with a combined flow rate of cooling water through all of the coolers of 1,396,450 lb/hr. The net power increased by 2,613 kW. The heat rate decreases by 91 Btu/kWh, which is 0.65%.

In the FWH-2 integration (Figure 65), extraction F was reduced by 52.9% and extraction G was reduced by 32.4% by weight. The integrated heat represents 45.7% of the total compressor cooler duties available, with a combined flow rate of cooling water through all of the coolers of 1,396,450 lb/hr. An additional 3,702 kW of net power was produced. The heat rate decreases by 128 Btu/kWh, which is 0.93% below the base case.

In the FWH-3 integration (Figure 66), extraction E was reduced by 35.8%, extraction F was reduced by 14.4% and extraction G was reduced by 11.1% by weight. The integrated heat represents 45.7% of the total compressor cooler duties available, with a combined flow rate of cooling water through all of the coolers of 1,396,450 lb/hr. In other words, only 45.7% of the total thermal energy in the hot cooling water streams from all compressor coolers is transferred to FWH-3. The remaining heat (54.3%) can be used as a thermal source for another thermal sink. The change in steam flow rates result in an additional 3,182 kW in net power produced. The heat rate decreases by 110 Btu/kWh, which is 0.80% less than the base case.

The hot cooling water stream only reduces each feedwater heater extraction stream; it does not eliminate any of the extractions. The FWH-3 integration is not as beneficial as in the Ramgen & Inline 4 cases. For the thermal integration to FWH-3, the total heat transferred from the hot cooling water stream to FWH-3A and FWH-2A is 56.85 MBtu/hr. For the thermal integration to FWH-2, the hot cooling water transfers 60.17 MBtu/hr of thermal energy to FWH-2A. Thus, the thermal source can provide more thermal energy to FWH-2A alone than to FWH-3A and FWH-2A separately. As a result, the FWH-2 heat integration case provides the greatest benefit to the plant.

Table 38. Integrally Geared 1 – 149, Post-Compressor Cooler (PC7) to FWHs 1, 2, and 3 Thermal Integration Results.

	Unit	MEA HI: PC7 to FWH-3	MEA HI: PC7 to FWH-2	MEA HI: PC7 to FWH-1	MEA w/o heat integration
Reduction in Ext. E	[%]	35.8%	0.0%	0.0%	0.0%
Reduction in Ext. F	[%]	14.4%	52.9%	0.0%	0.0%
Reduction in Ext. G	[%]	11.1%	32.4%	63.7%	0.0%
Compressor Heat Used	[%]	45.7%	45.7%	45.7%	0.0%
FG Flow to Compressor	[lb/hr]	1,073,080	1,073,080	1,073,080	1,073,080
Gross Power	[kW]	489,474	489,990	488,875	486,222
Gross Power minus Gen. Losses	[kW]	482,132	482,640	481,542	478,929
Turbine Cycle Heat Rate	[Btu/kWh]	9,906	9,896	9,919	9,973
Fan Power	[kW]	18,405	18,405	18,405	18,405
Pulv. Power	[kW]	3,456	3,456	3,456	3,456
Pump Power	[kW]	2,517	2,508	2,498	2,496
Aux. Power	[kW]	15,000	15,000	15,000	15,000
Total Pss	[kW]	39,378	39,369	39,359	39,357
Compressor Power	[kW]	43,182	43,182	43,182	43,182
Net Power	[kW]	399,572	400,090	399,001	396,390
Net Unit Heat Rate	[Btu/kWh]	13,770	13,752	13,789	13,880
HR Improvement	[%]	-0.80%	-0.93%	-0.65%	0.00%
Unit Efficiency	[%]	24.78%	24.81%	24.74%	24.58%

3.2.4 Integrally Geared 1 – 110 Compressor to FWHs Results

The process for the integration case to FWH-1 is illustrated in Figure 18. The process for the integration case to FWH-2 is illustrated in Figure 21. The processes are the same as described in the IG1 – 149 cases (Section 3.2.3). The thermal integration of the hot cooling water from the IG1 – 110 compressor to FWH-3 was not analyzed because none of the hot cooling water temperatures are high enough to heat the feedwater entering FWH-3. IC4 was selected for both thermal integration cases because, among the three intercoolers that produce hot cooling water temperatures high enough to be a thermal source for both FWH-2 and FWH-1 (IC2, IC3 and IC4), it has the largest maximum quantity of heat available (Table 27).

The results for thermally integrating the IG1-110 intercooler 4 (IC4) to FWH-1 and FWH-2 are shown in Table 39 and also in Figure 67 and Figure 68, respectively (Appendix F). In the FWH-1 integration (Figure 67), extraction G is reduced by 63.7% by weight. This represented 43.9% of the total compressor cooler duties available, with a combined flow rate of cooling water through all of the coolers of 2,634,198 lb/hr. The net power increases by 1,488 kW. The heat rate decreases by 50 Btu/kWh, or 0.37%. The FWH-2 integration case provides the greatest benefit to the plant since the additional steam flows through two turbines.

In the FWH-2 integration (Figure 68), extraction F is reduced by 8.3% and extraction G is reduced by 27.4% by weight. The integrated heat represents 15.4% of the total compressor cooler duties available, with a combined flow rate of cooling water through all of the coolers of 2,634,198 lb/hr. In other words, only 15.4% of the total thermal energy in the hot cooling water streams from all compressor coolers is used in this integration case. The remaining heat (84.6%) can be used as a thermal source for another thermal sink(s). An additional 3,702 kW of net power was produced. The heat rate decreased by 128 Btu/kWh, or 0.93%.

Table 39. Integrally Geared 1 – 110 Intercooler 4 (IC4) to FWHs 1 and 2 Thermal Integration Results.

	Unit	MEA HI: IC4 to FWH-2	MEA HI: IC4 to FWH-1	MEA w/o heat integration
Reduction in Ext. F	[%]	8.3%	0.0%	0.0%
Reduction in Ext. G	[%]	27.4%	31.8%	0.0%
Compressor Heat Used	[%]	15.4%	15.4%	0.0%
FG Flow to Compressor	[lb/hr]	1,073,080	1,073,080	1,073,080
Gross Power	[kW]	487,743	487,548	486,222
Gross Power minus Gen. Losses	[kW]	480,426	480,235	478,929
Turbine Cycle Heat Rate	[Btu/kWh]	9,942	9,946	9,973
Fan Power	[kW]	18,405	18,405	18,405
Pulv. Power	[kW]	3,456	3,456	3,456
Pump Power	[kW]	2,548	2,543	2,538
Aux. Power	[kW]	15,000	15,000	15,000
Total Pss	[kW]	39,409	39,405	39,399
Compressor Power	[kW]	37,496	37,496	37,496
Net Power	[kW]	403,521	403,334	402,033
Net Unit Heat Rate	[Btu/kWh]	13,635	13,641	13,685
HR Improvement	[%]	-0.37%	-0.32%	0.00%
Unit Efficiency	[%]	25.02%	25.01%	24.93%

3.3 Compressor Coolers to Coal Dryer

The final heat integration case explores the use of hot cooling water from the compressor coolers to partially dry the coal before it enters the mills. There are numerous benefits to drying the coal. First, less coal is required to achieve the same steam throttle conditions since some of the thermal energy from the coal is needed to heat its own water content. Less water equals less additional coal flow. Second, when the coal flow is reduced, less combustion air is required and less flue gas is produced. This reduces the power load for the FD and ID fans, MEA system and compressor since they are processing a reduced flow. However an additional fan is required for the drying process which may increase the total fan power. Third, in a plant with an MEA carbon capture system, the gross power increases because the steam extraction for the capture system is reduced. Fourth, there are fewer emissions since the capture requirement is usually proportional to the total flow. In addition, dried coal is pulverized more easily in the mills, but this factor is not considered in this analysis. When taken together, less coal is required to run the plant, the gross power is higher and the compressor power is lower so the heat rate improvements are magnified.

Research and work on coal drying is in the beginning stages. The Energy Research Center has conducted theoretical modeling and limited field experiments to predict the quantity of heat required to dry coal to various final moisture contents. The first full-size coal dryer, a continuous flow fluidized bed dryer, was installed in 2010. A continuous flow fluidized bed dryer continuously passes coal through the heated, fluidized bed (Figure 23) where the heat transferred by the hot air and fluid removes a percentage of the moisture in the coal.

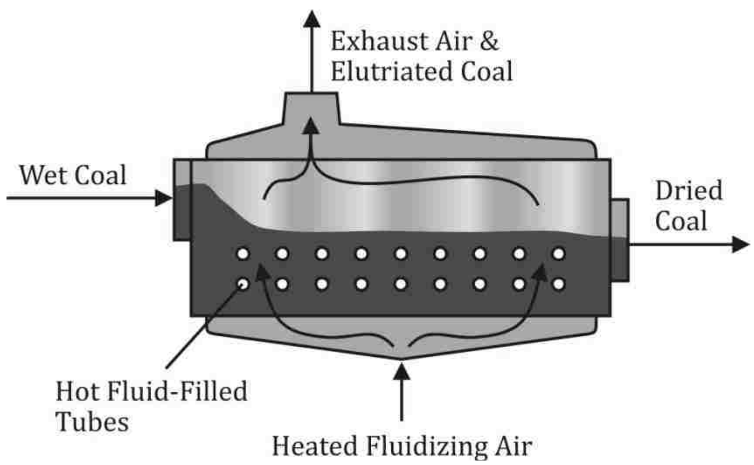


Figure 23. Illustration of Continuous Flow Fluidized Bed Dryer (Charles 2011).

Charles (2011) consolidated the results of the Energy Research Center's research into an Excel-based modeling program called the Coal Plant Program 6.0 (CPP). The layout of the coal dryer and air heater model is shown in Figure 24. The air is heated in a separate heater prior to the coal dryer bed and a fan is used to maintain the necessary flow of air through the bed.

Several factors affect the final moisture content of the coal including size of the fluidized bed, air temperature, air flow rate, water temperature, water flow rate and elutriation fraction. All of these variables can be adjusted to achieve the desired coal moisture content. The elutriation fraction is the portion of the coal that escapes the dryer bed with the air stream and thus is not dried properly. In the Coal Plant Program model, the elutriated coal is collected and mixed with the coal exiting the dryer. In calculations, it is assumed that the elutriated coal has the moisture content of the original feed coal. The dried coal, including the elutriated fraction, is then fed to the mills to be pulverized before going to the boiler.

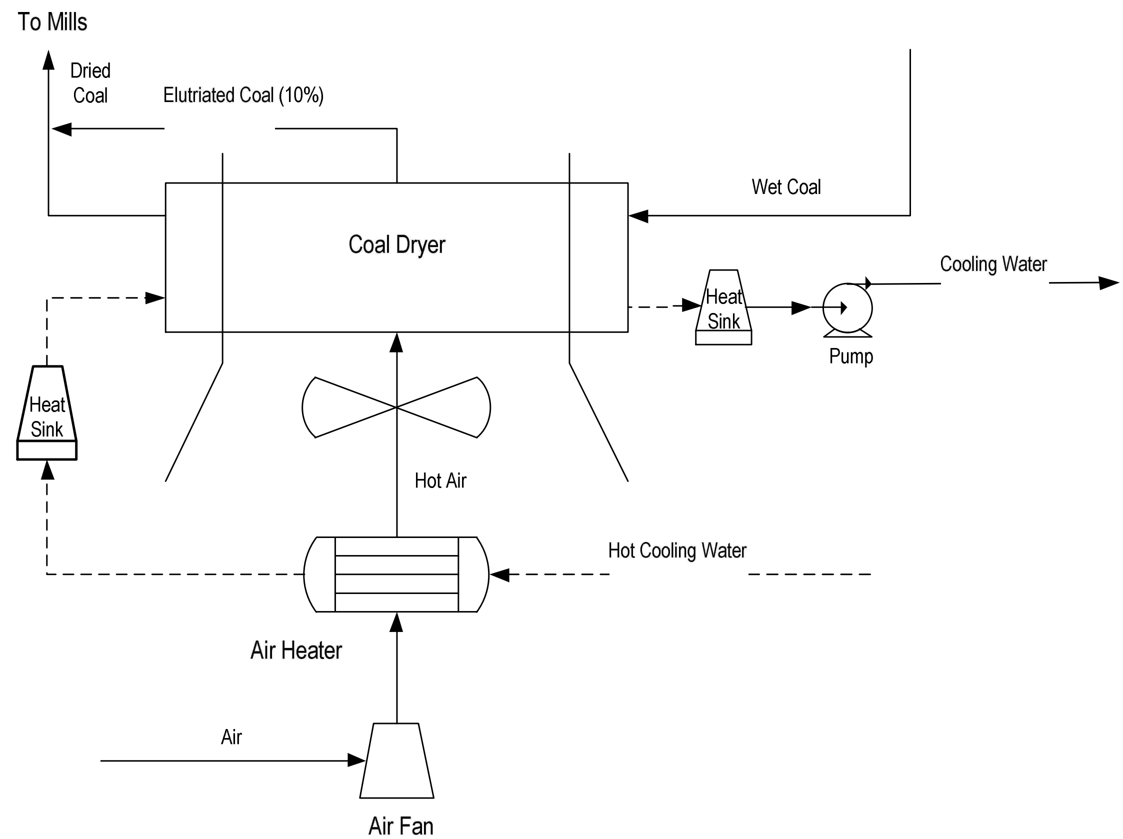


Figure 24. Diagram of Coal Dryer and Air Heater Heat Integration Case.

The combustion process is modeled as an equilibrium reaction in *Aspen Plus* (Aspen Technology 2000). However, in reality, the process does not reach equilibrium. The process is affected by the size of the coal particles, residence time in the boiler, air flow rate and amount of mixing between the coal and air, any of which can prevent the reaction from reaching equilibrium. Aspen Technology (2000) instructs users to specify the combustion process with "Identify possible products" with 14 separate products (H_2O, N_2, O_2, NO_2, NO, S, SO_2, SO_3, H_2, Cl_2, HCl, C, CO, and CO_2).

Typical measured values of flue gas carbon monoxide (CO) concentrations are 100 – 200 ppm, which can be neglected for combustion and boiler energy balance calculations. However, the drying process, when modeled with the method recommended by Aspen Technology, can produce very high CO concentrations (~1500ppm), which cannot be neglected. Thus, it is more accurate to specify the complete combustion of carbon to CO_2 and eliminate CO as a possible product. For the results described elsewhere in this report concerning the reboiler and FWH integrations, it is assumed that the higher CO concentration does not significantly affect the heat rate improvement results. For the coal drying cases, the specification for coal combustion products was changed from including CO to not including CO. Thus, comparisons between coal dryer integration cases and other integration cases must be made carefully, since the combustion process is slightly different.

The Coal Plant Program was used to calculate the heat duties and power required to dry coal using the compressor coolers as thermal sources. There are multiple factors that can affect the final coal moisture content and there is a range of possibilities for each factor. It is not possible, nor productive, to analyze each permutation of the various factors. Each set of results presented here represents one possible method of setting up the coal dryer.

In all of the following cases, the coal was dried as much as possible while assuming a lower limit of 15% moisture after the elutriated coal is mixed in. The cooling water flow rate and maximum cooling water temperature were entered into the Coal Plant Program. The actual water temperature and coal bed area were adjusted simultaneously until the lowest coal moisture is achieved between 15% and 28%. Then the CPP values were used as *Aspen Plus* model inputs, namely the inlet water temperatures for the air heater and coal dryer and the coal dryer heat duty.

The air heater is modeled as a heat exchanger with the air outlet temperature specified. The coal dryer bed has a specified inlet water temperature and dryer heat duty. To complete the process, a heat sink is placed before the coal dryer bed to set the water temperature entering the coal dryer. Another heat sink and a pump are placed after the dryer bed before the returning to the compressors as cooling water. In the Ramgen, Inline 4, and IG1-149 models, only one hot cooling water stream was used for the thermal integration. In the IG1-110 model, multiple hot cooling water streams were combined in order to achieve an appropriate hot cooling water flow rate and temperature.

3.3.1 RAMGEN Compressor to Coal Dryer Results

The case for thermally integrating the Ramgen post-compressor cooler (PC2) to the coal dryer is presented below. The CPP predicted performance of the drying process is compared to *Aspen Plus* results in Table 40. The hot cooling water from PC2 will provide heat to both the air heater and coal dryer bed. The coal is dried to 15.02% moisture using most of the available heat.

Table 40. Coal Plant Program and Aspen Plus Coal Dryer Results for Ramgen & Inline 4 Compressors.

	Unit	Coal Plant Program	Aspen Plus	Diff.
Coal Moisture	[%]	15.02%	15.02%	
Wet Coal Flow Into Dryer	[lb/hr]	638,103	630,384	-1.21%
Dry Coal Flow Out of Dryer	[lb/hr]	539,942	533,431	-1.21%
FG Flow	[lb/hr]	5,845,016	5,846,823	0.03%

The plant performances for no heat integration versus coal drying are presented in Table 41. In the coal drying case, the wet coal flow rate is reduced by 2.75%. Also, the steam extraction for the MEA system is reduced by 2.6% and the gross power increased by 3,948 kW. While the FD and ID fan powers are reduced, the addition of the air fan for the coal dryer increases the total fan power by 1,276 kW. The pump power increases since the hot cooling water experiences a pressure drop through each stage in the coal drying process. The pulverizer and compressor powers are reduced due to the change in flow rates through them.

The integrated heat represents 50.5% of the total compressor cooler duties available, with a combined flow rate of cooling water through both coolers of 1,257,659 lb/hr. In other words, only 50.5% of the total thermal energy in the hot cooling water streams from all compressor coolers is used in this integration case. The remaining heat (49.5%) can be used as a thermal source for another thermal sink(s). Overall, the heat rate decreased by 3.84% and the unit efficiency gained 0.98 percentage points.

Table 41. Ramgen Post-Compressor Cooler to Coal Dryer Thermal Integration Results.

	Unit	MEA HI: PC2 -> Coal Dryer	MEA w/o heat integration
Wet Coal Flow	[lb/hr]	630,384	648,178
Coal Flow Difference	[%]	-2.75%	0.00%
Coal HHV after Dryer	[Btu/lb]	9,957	8,426
Boiler Efficiency	[%]	89.92%	87.45%
Reduction in Reboiler Extraction	[%]	2.6%	0.0%
Compressor Heat Used	[%]	50.5%	0.0%
FG Flow to Compressor	[lb/hr]	1,047,160	1,077,250
Gross Power	[kW]	489,496	485,548
Gross Power minus Generator Losses	[kW]	482,154	478,265
Turbine Cycle Heat Rate	[Btu/kWh]	9,906	9,987
Fan Power	[kW]	19,616	18,340
Pulv. Power	[kW]	2,823	3,430
Pump Power	[kW]	2,493	2,470
Aux. Power	[kW]	15,000	15,000
Total Pss	[kW]	39,932	39,241
Compressor Power	[kW]	44,493	45,771
Net Power	[kW]	397,729	393,253
Net Unit Heat Rate	[Btu/kWh]	13,355	13,888
HR Improvement	[%]	-3.84%	0.00%
Unit Efficiency	[%]	25.55%	24.57%

3.3.2 Inline 4 Compressor to Coal Dryer Results

The case for thermally integrating the Inline 4 intercooler (IC2) to the coal dryer is presented below. The CPP predicted performance of the drying process is compared to *Aspen Plus* results in Table 40. The coal is dried to 15.02% moisture using most of the available heat. The same coal feedstock is used in each coal dryer integration case, so the heat required to dry the coal to specific moisture content is constant. Thus, the results in wet coal flow through station service power for both cases are practically identical. However, the results for compressor power, net power, net unit heat rate and unit efficiency reflect the compressor model selected.

The plant performances for no heat integration versus coal drying are presented in Table 42. In the coal drying case, the wet coal flow rate is reduced by 2.75%. Also, the steam extraction for the MEA system is reduced by 2.7% and the gross power increased by 3,940 kW.

While the FD and ID fan powers are reduced, the addition of the air fan for the coal dryer increases the total fan power by 607 kW. The pump power increases since the hot cooling water experiences a pressure drop through each stage in the coal drying process. The pulverizer and compressor powers are reduced due to the change in flow rates through them.

The integrated heat represents 53.5% of the total compressor cooler duties available, with a combined flow rate of cooling water through both coolers of 1,255,971 lb/hr. In other words, only 53.5% of the total thermal energy in the hot cooling water streams from all compressor coolers is used in this integration case. The remaining heat (46.5%) can be used as a thermal source for another thermal sink(s). Overall, the heat rate decreased by 3.84% and the unit efficiency gained 0.98 percentage points.

Table 42. Inline 4 Intercooler 2 to Coal Dryer Thermal Integration Results.

	Unit	MEA HI: IC2 to Coal Dryer	MEA w/o heat integration
Wet Coal Flow	[lb/hr]	630,384	648,177
Coal Flow Difference	[%]	-2.75%	0.00%
Coal HHV after Dryer	[Btu/lb]	9,957	8,426
Boiler Efficiency	[%]	89.92%	87.45%
Reduction in Reboiler Extraction	[%]	2.7%	0.0%
Compressor Heat Used	[%]	53.5%	0.0%
FG Flow to Compressor	[lb/hr]	1,047,080	1,077,240
Gross Power	[kW]	489,516	485,516
Gross Power minus Generator Losses	[kW]	482,173	478,233
Turbine Cycle Heat Rate	[Btu/kWh]	9,906	9,987
Fan Power	[kW]	19,616	18,340
Pulv. Power	[kW]	2,823	3,430
Pump Power	[kW]	2,493	2,470
Aux. Power	[kW]	15,000	15,000
Total Pss	[kW]	39,931	39,240
Compressor Power	[kW]	42,840	44,074
Net Power	[kW]	399,402	394,918
Unit HR	[Btu/kWh]	13,299	13,830
HR Improvement	[%]	-3.84%	0.00%
Unit Efficiency	[%]	25.66%	24.67%

3.3.3 Integrally Geared 1 – 149 Compressor to Coal Dryer Results

The case for thermally integrating the IG1-149 post-compressor cooler (PC7) to the coal dryer is presented below. The CPP predicted performance of the drying process is compared to *Aspen Plus* results in Table 43. The hot cooling water from PC7 will provide heat to both the air heater and coal dryer bed. The coal is dried to 17.49% moisture with the thermal energy transferred by the hot cooling water from PC7.

Table 43. Coal Plant Program and Aspen Plus Coal Dryer Results for the Integrally Geared 1 – 149 Compressor.

	Unit	Coal Plant Program	Aspen Plus	Diff.
Coal Moisture	[%]	17.49%	17.49%	
Wet Coal Flow Into Dryer	[lb/hr]	640,679	632,837	-1.22%
Dry Coal Flow Out of Dryer	[lb/hr]	558,341	551,537	-1.22%
FG Flow	[lb/hr]	5,891,392	5,889,948	-0.02%

The plant performances for no heat integration versus coal drying are presented in Table 44. In the coal drying case, the wet coal flow rate is reduced by 2.37%. Also, the steam extraction for the MEA system is reduced by 2.3% and the gross power increased by 3,448 kW. While the FD and ID fan powers are reduced, the addition of the air fan for the coal dryer increases the total fan power by 1,446 kW. The pump power increases since the hot cooling water experiences a pressure drop through each stage in the coal drying process. The pulverizer and compressor powers are reduced due to the change in flow rates through them.

The integrated heat represents 44.3% of the total compressor cooler duties available, with a combined flow rate of cooling water through both coolers of 1,666,607 lb/hr. In other words, only 44.3% of the total thermal energy in the hot cooling water streams from all compressor coolers is used in this integration case. The remaining heat (55.7%) can be used as a thermal source for another thermal sink(s). Overall, the heat rate decreased by 3.22% and the unit efficiency gained 0.82 percentage points.

Table 44. IG 1 – 149 Post-Compressor Cooler to Coal Dryer Thermal Integration Results.

	Unit	MEA HI: PC7 to Reboiler	MEA w/o heat integration
Wet Coal Flow	[lb/hr]	632,837	648,177
Coal Flow Difference	[%]	-2.37%	0.00%
Coal HHV after Dryer	[Btu/lb]	9,668	8,426
Boiler Efficiency	[%]	17.49%	28.09%
Reduction in Reboiler Extraction	[%]	2.3%	0.0%
Compressor Heat Used	[%]	44.3%	0.0%
FG Flow to Compressor	[lb/hr]	1,051,320	1,079,750
Gross Power	[kW]	488,956	485,508
Gross Power minus Generator Losses	[kW]	481,622	478,226
Turbine Cycle Heat Rate	[Btu/kWh]	9,917	9,987
Fan Power	[kW]	19,786	18,340
Pulv. Power	[kW]	2,919	3,430
Pump Power	[kW]	2,524	2,506
Aux. Power	[kW]	15,000	15,000
Total Pss	[kW]	40,229	39,276
Compressor Power	[kW]	42,306	43,352
Net Power	[kW]	399,087	395,597
Net Unit Heat Rate	[Btu/kWh]	13,361	13,806
HR Improvement	[%]	-3.22%	0.00%
Unit Efficiency	[%]	25.54%	24.71%

3.3.4 Integrally Geared 1 – 110 Compressor to Coal Dryer Results

The case for thermally integrating the IG1-110 intercoolers 2 through 6 (IC2, IC3, IC4, IC5 & IC6) to the coal dryer is presented below. There is a small amount of heat available in each intercooler, so the hot cooling water streams from 5 intercoolers were combined to achieve a high temperature and flow rate. The CPP predicted performance of the drying process is compared to *Aspen Plus* results in Table 45. The combined hot cooling water streams from IC2 through IC6 will provide heat to both the air heater and coal dryer bed. The coal is dried to 15.04% moisture with the available heat.

Table 45. Coal Plant Program and Aspen Plus Coal Dryer Results for the Integrally Geared 1 – 110 Compressor.

	Unit	Coal Plant Program	Aspen Plus	Diff.
Coal Moisture	[%]	15.04%	15.04%	
Wet Coal Flow Into Dryer	[lb/hr]	638,529	630,870	-1.20%
Dry Coal Flow Out of Dryer	[lb/hr]	540,478	533,967	-1.20%
FG Flow - ESP	[lb/hr]	5,849,164	5,851,331	0.04%

The plant performances for no heat integration versus coal drying are presented in Table 46. In the coal drying case, the wet coal flow rate is reduced by 2.67%. Also, the steam extraction for the MEA system is reduced by 2.6% and the gross power increased by 3,923 kW. While the FD and ID fan powers are reduced, the addition of the air fan for the coal dryer increases the total fan power by 1,291 kW. The pump power increases since the hot cooling water experiences a pressure drop through each stage in the coal drying process. The pulverizer and compressor powers are reduced due to the change in flow rates through them.

The integrated heat represents 55.4% of the total compressor cooler duties available, with a combined flow rate of cooling water through both coolers of 2,320,034 lb/hr. In other words, only 55.4% of the total thermal energy in the hot cooling water streams from all compressor coolers is used in this integration case. The remaining heat (44.6%) can be used as a thermal source for another thermal sink(s). Overall, the heat rate decreased by 3.67% and the unit efficiency gained 0.95 percentage points.

Table 46. Integrally Geared 1 – 110 Post-Compressor Cooler to Coal Dryer Thermal Integration Results.

	Unit	MEA HI: IC2-6 to Coal Dryer	MEA w/o heat integration
Wet Coal Flow	[lb/hr]	630,870	648,177
Coal Flow Difference	[%]	-2.67%	0.00%
Coal HHV after Dryer	[Btu/lb]	9,954	8,426
Boiler Efficiency	[%]	89.86%	87.45%
Reduction in Reboiler Extraction	[%]	2.6%	0.0%
Compressor Heat Used	[%]	55.4%	0.0%
FG Flow to Compressor	[lb/hr]	1,047,880	1,077,240
Gross Power	[kW]	489,428	485,505
Gross Power minus Generator Losses	[kW]	482,087	478,223
Turbine Cycle Heat Rate	[Btu/kWh]	9,907	9,987
Fan Power	[kW]	19,631	18,340
Pulv. Power	[kW]	2,826	3,430
Pump Power	[kW]	2,585	2,538
Aux. Power	[kW]	15,000	15,000
Total Pss	[kW]	40,042	39,308
Compressor Power	[kW]	36,615	37,641
Net Power	[kW]	405,430	401,274
Net Unit Heat Rate	[Btu/kWh]	13,111	13,611
HR Improvement	[%]	-3.67%	0.00%
Unit Efficiency	[%]	26.02%	25.07%

3.4 Stripper Condenser Heat Integration Cases

The next set of thermal integration cases used the stripper condenser as the thermal source. It is thermally integrated to the FWHs and to the coal dryer. In addition, a combination case was analyzed in which the integration of the condenser to the FWHs was combined with the integration of the compressor coolers to the reboiler.

The stripper condenser contains a large quantity of heat, greater than the heat in any of the compressor coolers. This is due to the large quantity of water vapor entering the condenser. The water releases a large amount of heat as it changes phase. The vapor that rises to the top of the stripper is approximately 50% water. This has also been predicted by other MEA Scrubber models reported in the literature (Table 47).

Table 47. Stripper Condenser Stream Composition Literature Comparison.

	Unit	Martin 2011	NETL 2002	Freguia 2002	Fashami 2007
		Model	Model	Model	Model
Gas Composition entering Condenser	CO_2 [mol %]	51%	54%	64%	55%
	H_2O [mol %]	49%	43%	36%	45%
Gas Composition exiting Condenser	CO_2 [mol %]	98%	91%	92%	95%
	H_2O [mol %]	2%	4%	8%	5%

The performance of the stripper condenser is the same regardless of the compressor selected for thermal integration cases to the FWHs. Thus, the boiler efficiency and throttle conditions remain the same. In addition, the change in net power and net unit heat rate for all of the compressor cases will be constant. Only the results for compressor power, net power, net unit heat rate and unit efficiency are unique to the compressor model selected.

However, in thermal integration cases of the compressor coolers to the coal dryer, the results can depend on the compressor selected. In thermal integration cases of the stripper condenser to the coal dryer, the results will not depend on the compressor selected. The heat available in the thermal source determines how much the coal can be dried. When dried coal is combusted in the boiler, less flue gas is produced. Thus, the power requirements for the MEA system and compressor are reduced since they are processing less flue gas. Yet, for the stripper condenser, the amount of drying will be constant since it processes the same amount of CO_2, regardless of the compressor that follows the MEA system. The details of the condenser to coal dryer integration case will be described further in section 3.6.3 along with the coal dryer results.

3.4.1 Stripper Condenser to FWHs

The next thermal integration case uses hot cooling water from the stripper condenser to offset the steam extractions for the FWHs. There is sufficient heat in the hot cooling water stream from the stripper condenser to replace FWH-1, FWH-2 and FWH-3 with a single heat exchanger and eliminate the usual extractions feeding them (Figure 25). The original extractions are shown in Figure 17. Since the discharge from the Steam APH and SSR originally entered FWH-3 and FWH-2, respectively, they are both re-directed to FWH-4. Since FWH-4 is receiving additional heat, Extraction D can be reduced as well. In addition, the Drain Pump was removed since there was no extraction that required pumping. After the hot cooling water passes through the single feedwater heater, it must go through a cooler and a pump before returning to the stripper condenser as cooling water.

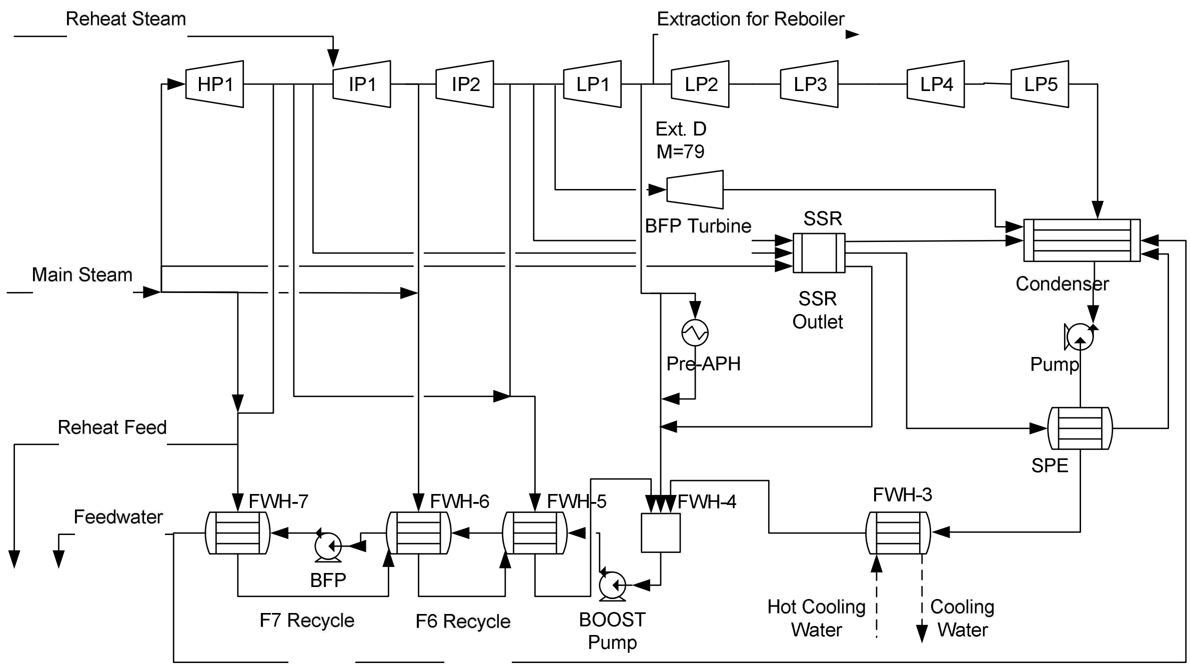


Figure 25. Stripper Condenser to FWH-3 Diagram.

For this case, the IG1 – 110 compressor was selected to demonstrate the results for this thermal integration case. The results for the condenser to FWH-3 thermal integration case are shown in Table 48 and in Figure 69 in Appendix F. The results for the Ramgen, Inline 4 and the IG1 – 149 compressors are shown in the next section. The gross power for the plant increases by 29,814 kW, due to the elimination of three steam extractions. The pump power increases to overcome the pressure drop through FWH-3A, but the drain pump power was eliminated.

The integrated heat represents 96.1% of the stripper condenser duty, with a flow rate of cooling water of 3,376,030 lb/hr. In other words, only 96.1% of the total thermal energy in the hot cooling water stream from the stripper condenser is used in this integration case. The remaining heat (3.9%) can be used as a thermal source for another thermal sink(s). Overall, the net unit heat rate decreases by 6.78%. The unit efficiency gained 1.81 percentage points.

Table 48. Stripper Condenser to FWH-3, with IG1-110, Thermal Integration Results.

	Unit	MEA HI: Condenser to FWH-3	MEA w/o heat integration
Reduction in Ext. D	[%]	69.3%	0%
Reduction in Ext. E	[%]	100.0%	0%
Reduction in Ext. F	[%]	100.0%	0%
Reduction in Ext. G	[%]	100.0%	0%
Other Model Changes:		Eliminated FWHs 1&2, & Drain Pump.	N/A
Condenser Heat Used	[%]	96.1%	0%
FG Flow to Compressor	[lb/hr]	1,073,080	1,073,080
Gross Power	[kW]	516,036	486,222
Gross Power minus Generator Losses	[kW]	508,295	478,929
Turbine Cycle Heat Rate	[Btu/kWh]	9,397	9,973
Fan Power	[kW]	18,405	18,405
Pulv. Power	[kW]	3,456	3,456
Pump Power	[kW]	2,681	2,538
Aux. Power	[kW]	15,000	15,000
Total Pss	[kW]	39,543	39,399
Compressor Power	[kW]	37,496	37,496
Net Power	[kW]	431,257	402,033
Net Unit Heat Rate	[Btu/kWh]	12,758	13,685
HR Improvement	[%]	-6.78%	0.00%
Unit Efficiency	[%]	26.74%	24.93%

3.4.2 Compressor & Condenser Combination Integration

In addition, a combination thermal integration case was analyzed. In this model, the compressor coolers are integrated with the stripper reboiler and the stripper condenser is integrated with FWH-3. The methods for the integrations were described previously. The results demonstrate the improvement in plant performance from the combination of thermal integration cases. The changes in plant performance in combination are the summation of the improvements from the separate integrations. The results for each compressor model are described below. Recall that the heat integration of IG1 – 110 to the reboiler is not feasible, so this compressor is not included here.

The Ramgen compressor results for the combination and individual thermal integration cases are compared to the base case in Table 49. The results for the combination case are illustrated in Figure 70 in Appendix F. For the integration, 53.5% of the compressor heat and 96.1% of the condenser heat was utilized. As a result three steam extractions were eliminated and two extractions were reduced. The drain pump and FWHs 1 and 2 were removed for the condenser integration cases. In the combination case, the gross power increased by 41,285 kW while the net power increased by 40,510 kW. The pump power increased to overcome the pressure drops through the integration-related heat exchangers. The net unit heat rate decreased by 1,302 Btu/kWh while the unit efficiency gained 2.52 percentage points.

The Inline 4 compressor results for the combination and individual thermal integration cases are compared to the base case in Table 50. The results for the combination case are illustrated in Figure 71 in Appendix F. For the integration, 49.7% of the compressor heat and 96.1% of the condenser heat was utilized. As a result three steam extractions were eliminated and two extractions were reduced. The drain pump and FWHs 1 and 2 were removed for cases including the condenser integration. In the combination case, the gross power increased by 39,876 kW while the net power increased by 39,122 kW. The pump power increased to overcome the pressure drops through the integration-related heat exchangers. The net unit heat rate decreased by 1,251 Btu/kWh while the unit efficiency gained 2.42 percentage points.

The IG1-149 compressor results for the combination and individual thermal integration cases are compared to the base case in Table 51. The results for the combination case are illustrated in Figure 72 in Appendix F. For the integration, 16.1% of the compressor heat and 96.1% of the condenser heat was utilized. As a result three steam extractions were eliminated and two extractions were reduced. The drain pump and FWHs 1 and 2 were removed for cases including the condenser integration. In the combination case, the gross power increased by 31,231 kW while the net power increased by 30,610 kW. The pump power increased to overcome the pressure drops through the integration-related heat exchangers. The net unit heat rate decreased by 995 Btu/kWh while the unit efficiency gained 1.90 percentage points.

Table 49. Ramgen Compressor to Reboiler and Condenser to FWH-3 Thermal Integration Results.

	Unit	MEA HI: Comp to Reboiler & Cond to FWH3	MEA HI: Cond to FWH3	MEA HI: Comp to Reboiler	MEA w/o heat integration
Reduction in Reboiler Ext.	[%]	7.7%	N/A	7.7%	0%
Reduction in Ext. D	[%]	69.3%	69.3%	N/A	0%
Reduction in Ext. E	[%]	100.0%	100.0%	N/A	0%
Reduction in Ext. F	[%]	100.0%	100.0%	N/A	0%
Reduction in Ext. G	[%]	100.0%	100.0%	N/A	0%
Other Model Changes:		Eliminated FWHs 1,2&3, & Drain Pump.	Eliminated FWHs 1,2&3, & Drain Pump.	N/A	N/A
Compressor Heat Used	[%]	53.5%	0%	53.5%	0%
Condenser Heat Used	[%]	96.1%	96.1%	0%	0%
FG Flow to Compressor	[lb/hr]	1,073,080	1,073,080	1,073,080	1,073,080
Gross Power	[kW]	527,507	516,036	497,694	486,222
Gross Power minus Generator Losses	[kW]	519,595	508,295	490,228	478,929
Turbine Cycle Heat Rate	[Btu/kWh]	9,192	9,397	9,743	9,973
Fan Power	[kW]	18,405	18,405	18,405	18,405
Pulv. Power	[kW]	3,456	3,456	3,456	3,456
Pump Power	[kW]	2,628	2,615	2,484	2,471
Aux. Power	[kW]	15,000	15,000	15,000	15,000
Total Pss	[kW]	39,489	39,476	39,346	39,332
Compressor Power	[kW]	45,594	45,594	45,594	45,594
Net Power	[kW]	434,512	423,225	405,289	394,002
Net Unit Heat Rate	[Btu/kWh]	12,662	13,000	13,575	13,964
HR Improvement	[%]	-9.32%	-6.90%	-2.78%	0.00%
Unit Efficiency	[%]	26.95%	26.25%	25.13%	24.43%

Table 50. Inline 4 Compressor to Reboiler and Condenser to FWH-3 Thermal Integration Results.

	Unit	MEA HI: Comp to Reboiler & Cond to FWH3	MEA HI: Cond to FWH3	MEA HI: Comp to Reboiler	MEA w/o heat integration
Reduction in Reboiler Ext.	[%]	6.7%	N/A	6.7%	0%
Reduction in Ext. D	[%]	69.3%	69.3%	N/A	0%
Reduction in Ext. E	[%]	100.0%	100.0%	N/A	0%
Reduction in Ext. F	[%]	100.0%	100.0%	N/A	0%
Reduction in Ext. G	[%]	100.0%	100.0%	N/A	0%
Other Model Changes:		Eliminated FWHs 1&2, & Drain Pump.	Eliminated FWHs 1&2, & Drain Pump.	N/A	N/A
Compressor Heat Used	[%]	49.7%	0%	49.7%	0%
Condenser Heat Used	[%]	96.1%	96.1%	0%	0%
FG Flow to Compressor	[lb/hr]	1,073,080	1,073,080	1,073,080	1,073,080
Gross Power	[kW]	526,098	516,036	496,284	486,222
Gross Power minus Generator Losses	[kW]	518,206	508,295	488,840	478,929
Turbine Cycle Heat Rate	[Btu/kWh]	9,217	9,397	9,771	9,973
Fan Power	[kW]	18,405	18,405	18,405	18,405
Pulv. Power	[kW]	3,456	3,456	3,456	3,456
Pump Power	[kW]	2,627	2,614	2,484	2,471
Aux. Power	[kW]	15,000	15,000	15,000	15,000
Total Pss	[kW]	39,488	39,475	39,345	39,332
Compressor Power	[kW]	43,905	43,905	43,905	43,905
Net Power	[kW]	434,814	424,915	405,590	395,692
Net Unit Heat Rate	[Btu/kWh]	12,654	12,948	13,565	13,905
HR Improvement	[%]	-9.00%	-6.88%	-2.44%	0.00%
Unit Efficiency	[%]	26.96%	26.35%	25.15%	24.54%

Table 51. Integrally Geared 1 – 149 Compressor to Reboiler and Condenser to FWH-3 Thermal Integration Results.

	Unit	MEA HI: Comp to Reboiler & Cond to FWH3	MEA HI: Cond to FWH3	MEA HI: Comp to Reboiler	MEA w/o heat integration
Reduction in Reboiler Ext.	[%]	1.0%	N/A	1.0%	0%
Reduction in Ext. D	[%]	69.3%	69.3%	N/A	0%
Reduction in Ext. E	[%]	100.0%	100.0%	N/A	0%
Reduction in Ext. F	[%]	100.0%	100.0%	N/A	0%
Reduction in Ext. G	[%]	100.0%	100.0%	N/A	0%
Other Model Changes:		Eliminated FWHs 1&2, & Drain Pump.	Eliminated FWHs 1&2, & Drain Pump.	N/A	N/A
Compressor Heat Used	[%]	16.1%	0%	16.1%	0%
Condenser Heat Used	[%]	96.1%	96.1%	0%	0%
FG Flow to Compressor	[lb/hr]	1,073,080	1,073,080	1,073,080	1,073,080
Gross Power	[kW]	517,558	516,036	487,746	486,222
Gross Power minus Generator Losses	[kW]	509,795	508,295	480,430	478,929
Turbine Cycle Heat Rate	[Btu/kWh]	9,369	9,397	9,942	9,973
Fan Power	[kW]	18,405	18,405	18,405	18,405
Pulv. Power	[kW]	3,456	3,456	3,456	3,456
Pump Power	[kW]	2,651	2,640	2,507	2,497
Aux. Power	[kW]	15,000	15,000	15,000	15,000
Total Pss	[kW]	39,512	39,501	39,368	39,358
Compressor Power	[kW]	43,182	43,182	43,182	43,182
Net Power	[kW]	427,101	425,612	397,880	396,388
Net Unit Heat Rate	[Btu/kWh]	12,882	12,927	13,828	13,880
HR Improvement	[%]	-7.19%	-6.87%	-0.37%	0.00%
Unit Efficiency	[%]	26.49%	26.39%	24.67%	24.58%

3.4.3 Stripper Condenser to Coal Dryer Results

The combustion process is modeled as an equilibrium reaction in *Aspen Plus* (Aspen Technology 2000). However, in reality, the process does not reach equilibrium. The process is affected by the size of the coal particles, residence time in the boiler, air flow rate and amount of mixing between the coal and air, any of which can prevent the reaction from reaching equilibrium. Aspen Technology (2000) instructs users to specify the combustion process with "Identify possible products" with 14 separate products (H_2O, N_2, O_2, NO_2, NO, S, SO_2, SO_3, H_2, Cl_2, HCl, C, CO, and CO_2).

Typical measured values of flue gas carbon monoxide (CO) concentrations are 100 – 200 ppm, which can be neglected for combustion and boiler energy balance calculations. However, the drying process, when modeled with the method recommended by Aspen Technology, can produce very high CO concentrations (~1500ppm), which cannot be neglected. Thus, it is more accurate to specify the complete combustion of carbon to CO_2 and eliminate CO as a possible product. For the results described elsewhere in this report concerning the reboiler and FWH integrations, it is assumed that the higher CO concentration does not significantly affect the heat rate improvement results. For the coal drying cases, the specification for coal combustion products was changed from including CO to not including CO. Thus, comparisons between coal dryer integration cases and other integration cases must be made carefully, since the combustion process is slightly different.

The case for thermally integrating the Stripper Condenser to the coal dryer is presented below. The history, changes in plant performance and the overall benefits of coal drying are reviewed in Section 3.5. The drying process is illustrated in Figure 24. The performance of the drying process predicted by the Coal Plant Program is compared to *Aspen Plus* results in Table 52. The hot cooling water from the condenser will provide heat to both the air heater and coal dryer bed. The coal is dried to 15.03% moisture using the available heat.

There is a high quantity of heat available in the stripper condenser. There is too much to use for coal drying alone. In this thermal integration case, the cooling water flow rate in the stripper condenser was set to $3.375*10^6$ lb/hr to achieve a high discharge temperature. Then the flow was split so only $1.0*10^6$ lb/hr traveled through the air heater and coal dryer. The hot cooling water temperature was high enough to dry the coal to 15.03%.

Table 52. Coal Plant Program and Aspen Plus Coal Dryer Results for the Stripper Condenser to Coal Drying.

	Unit	Coal Plant Program	Aspen Plus	Diff.
Coal Moisture	[%]	15.03%	15.03%	
Wet Coal Flow Into Dryer	[lb/hr]	638,058	630,334	-1.21%
Dry Coal Flow Out of Dryer	[lb/hr]	539,996	533,451	-1.21%
FG Flow - ESP	[lb/hr]	5,844,728	5,846,455	0.03%

The plant performances for no heat integration versus coal drying are presented in Table 53. If the condenser is used as the thermal source for the coal dryer, the amount of drying will be constant, regardless of the compressor selected. Results for this thermal integration case are presented using a Ramgen compressor. In the coal drying case, the wet coal flow rate is reduced by 2.75%. Also, the steam extraction for the MEA system is reduced by 2.7% and the gross power increased by 3,993 kW. While the FD and ID fan powers are reduced, the addition of the air fan for the coal dryer increases the total fan power by 1,275 kW. The pump power increases since the hot cooling water experiences a pressure drop through each stage in the coal drying process. The pulverizer and compressor powers are reduced due to the change in flow rates through them. Overall, the heat rate decreased by 3.85% and the unit efficiency gained 0.98 percentage points.

Table 53. Condenser to Coal Dryer Thermal Integration Results.

	Unit	MEA HI: Condenser to Coal Dryer	MEA w/o heat integration
Wet Coal Flow	[lb/hr]	630,334	648,178
Coal Flow Difference	[%]	-2.75%	0.00%
Coal HHV after Dryer	[Btu/lb]	9,956	8,426
Boiler Efficiency	[%]	89.93%	87.45%
Reboiler LPS Ext.	[%]	97.3%	100.0%
Condenser Heat Used	[%]	26.5%	0.0%
FG Flow to Compressor	[lb/hr]	1,047,030	1,077,250
Gross Power	[kW]	489,541	485,548
Gross Power minus Generator Losses	[kW]	482,198	478,265
Turbine Cycle Heat Rate	[Btu/kWh]	9,905	9,987
Fan Power	[kW]	19,615	18,340
Pulv. Power	[kW]	2,823	3,430
Pump Power	[kW]	2,518	2,470
Aux. Power	[kW]	15,000	15,000
Total Pss	[kW]	39,956	39,241
Compressor Power	[kW]	44,487	45,771
Net Power	[kW]	397,755	393,253
Net Unit Heat Rate	[Btu/kWh]	13,353	13,888
HR Improvement	[%]	-3.85%	0.00%
Unit Efficiency	[%]	25.55%	24.57%

4.0 Conclusions

The energy penalty of the MEA capture system is large. The net power of the plant is reduced by approximately 33% to run the MEA system and compressor. Technologies designed to reduce this energy penalty will be sought by plant owners. Several thermal integration cases were evaluated using the CO_2 compressor and the MEA stripper condenser as thermal sources. In the real world, there are myriad compressor configurations and methods for thermal integration. Thus, it must be reiterated that the thermal integration cases presented here are not the only possibilities, so these conclusions may not always apply.

The four CO_2 compressors evaluated for this study have different power requirements. At the beginning of this study, it was thought that thermal integration would result in the compressors with the largest power requirement having better performance due to higher CO_2 stage discharge temperatures and larger quantities of waste heat than the compressors with the smallest power requirement. However, unexpectedly, the benefits of a small compressor power outweighed the gains from thermal integration. In addition, no single compressor option had the best performance in all cases. The results are highly case specific. Plant optimization will require careful analysis of several thermal integration scenarios along with space and cost considerations. The following are some general guidelines drawn from these results that will help evaluate possible thermal integration cases.

1. It is better to consider the heat rate value, rather than the percent heat rate improvement, when evaluating plant performance. The main reason is that each base case has a different heat rate, depending on the compressor used. The compressors with a high power requirement have the highest CO_2 discharge temperatures and therefore the best opportunities for thermal integration. As a result, the percent heat rate improvement will be relatively large. The compressors with a low power requirement tend to have lower CO_2 discharge temperatures and fewer opportunities for thermal integration. As a result, the percent heat rate improvement will be relatively small. In addition, the percent heat rate improvement

is calculated by dividing the heat rate from the thermal integration case by the base case heat rate for that particular compressor. Thus, the percent improvements are not comparable between compressor models.

2. The greatest reduction in energy penalty is obtained from selecting the compressor with the lowest power requirement. In this study, a Ramgen, an inline and two integrally geared compressors, which differed in isentropic efficiencies and pressure ratios, were analyzed. The Ramgen compressor requires the highest power at 45.6 MW and has a base case heat rate of 13,964 Btu/kWh. The Inline 4 compressor requires 43.9 MW of power and it has a base case heat rate of 13,905 Btu/kWh. For the Integrally Geared compressors, the IG 1 – 149 requires 43.2 MW and it has a base case heat rate of 13,880 Btu/kWh, while the IG 1 – 110 model requires 37.5 MW and has a base case heat rate of 13,685 Btu/kWh. The best base case (i.e. without thermal integration) is for the IG 1 – 110 compressor because it has the lowest power requirement.

3. Finally, select the best method to recycle thermal waste energy in the plant. The differences in the compressor configurations result in a variety of compressor stage CO_2 discharge temperatures. The thermal integration method depends first on the CO_2 and hot cooling water temperatures available. Since heat transfer is dependent on a temperature difference, the most important factor for thermal integration is the CO_2 discharge temperature. The other factor is the quantity of energy available in the thermal source. Given the same quantity of thermal energy, it is possible for two different temperature thermal sources to provide the same heat rate improvement. However, if the temperature of a thermal source is not greater than the temperature of the thermal sink, the heat integration is not possible, regardless of quantity.

Now, the results from this study will be explained using these guidelines. The Ramgen, Inline 4 and IG 1 – 149 compressor coolers have discharge temperatures from at least one heat exchanger that can be used as a thermal source for the reboiler, the first three FWHs, and the coal dryer. The thermal energy from the IG 1 – 110 compressor coolers can be used as a thermal source for the first two FWHs and coal dryer. None of the IG 1 – 110 stage discharge temperatures is higher than the reboiler operating temperature or the design feedwater discharge temperature from FWH-3. This precludes the IG 1 – 110 compressor from being a thermal source for the reboiler or for FWH-3.

Three compressors can be thermally integrated to the stripper reboiler. The results are shown in Figure 26, Figure 27, and Figure 28. The best performing reboiler integration was Inline 4. Although the heat rates for the Ramgen and the Inline 4 reboiler integration cases are similar (Figure 26), the net power for the Inline 4 reboiler case is higher than the Ramgen because less power goes to the compressor itself (Figure 28). The IG 1 – 149 compressor has a lower quantity of heat available in the post-compressor cooler so the benefits of thermal integration to the reboiler are smaller.

All four compressors can be thermally integrated to the FWHs. The results are shown in Figure 26, Figure 27, and Figure 28. The best heat rate improvement for Ramgen and Inline 4 was obtained with thermal integration to FWH-3. The best heat rate improvement for the IG 1 compressors was obtained with thermal integration to FWH-2. The best heat rate overall was in the IG 1 – 110 IC4 to FWH-2 case, for which the heat rate is 13,635 Btu/kWh (Figure 26). The other compressors exhibited large gains in net power. However, only the Inline 4 case, the second best FWH integration, performed better than the IG 1 – 110 base case.

Again, all four compressors can be thermally integrated to the coal dryer. The results are shown in Figure 29, Figure 30, and Figure 31. Comparison of coal dryer cases to the reboiler or FWH cases must be performed carefully since the base case model is different for the coal dryer integration. The thermal energy from Ramgen, Inline 4 and IG 1 – 110 was able to dry the coal to approximately 15%, while the energy from IG 1 – 149 was able to dry the coal to 17.5%. The IG 1 – 110 compressor to coal dryer heat integration case had the best performance. Again, Ramgen has the largest heat rate improvement, but still has the highest heat rate.

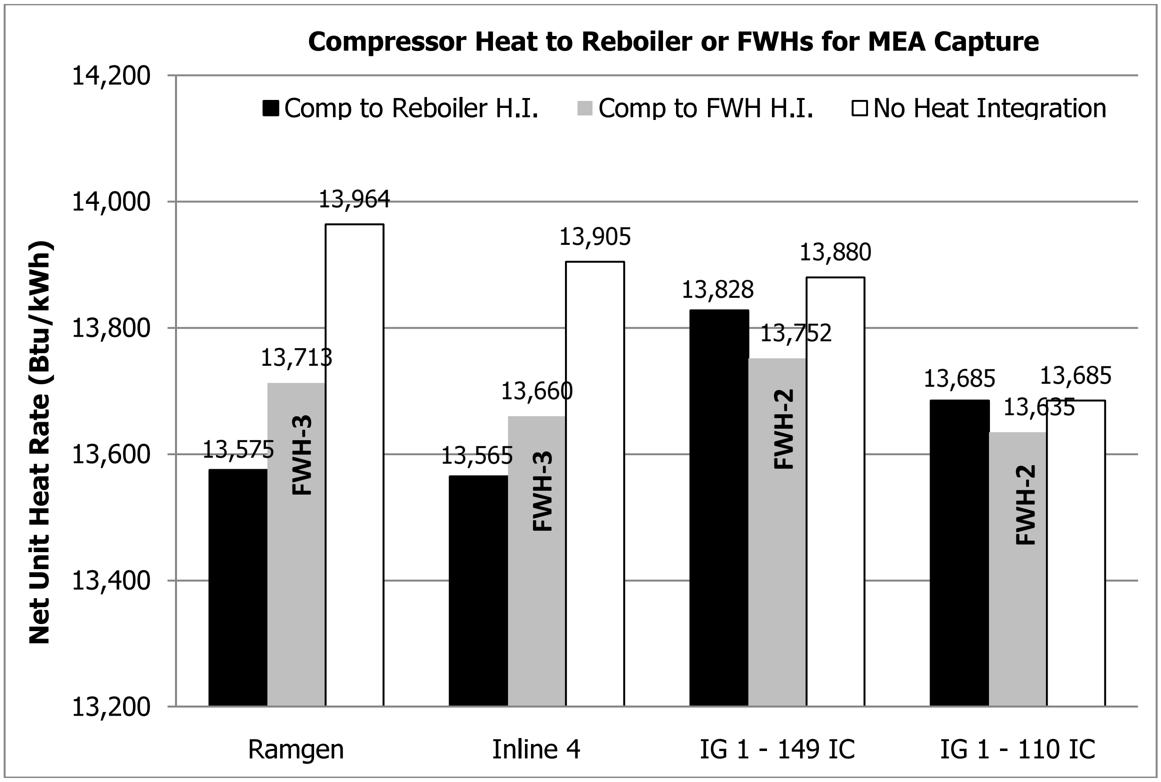


Figure 26. Compressor to Reboiler or FWH Thermal Integration Results for Net Unit Heat Rate.

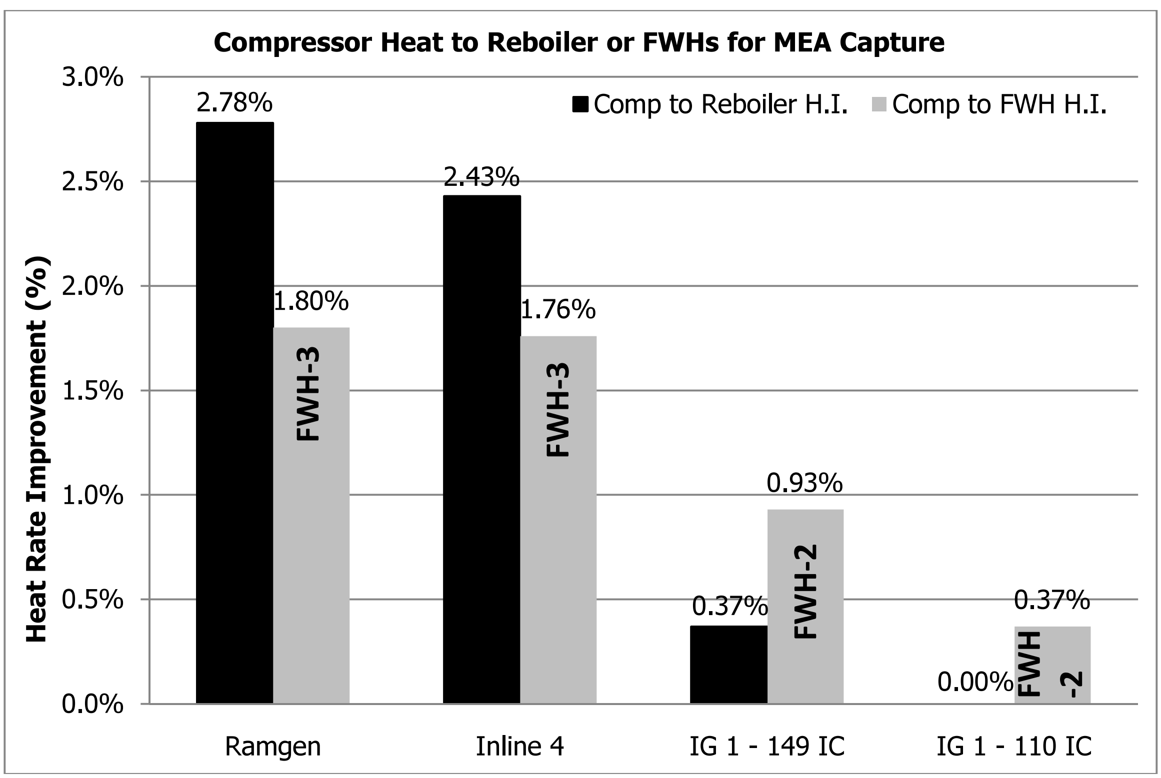


Figure 27. Compressor to Reboiler or FWH Thermal Integration Results for Heat Rate Improvement.

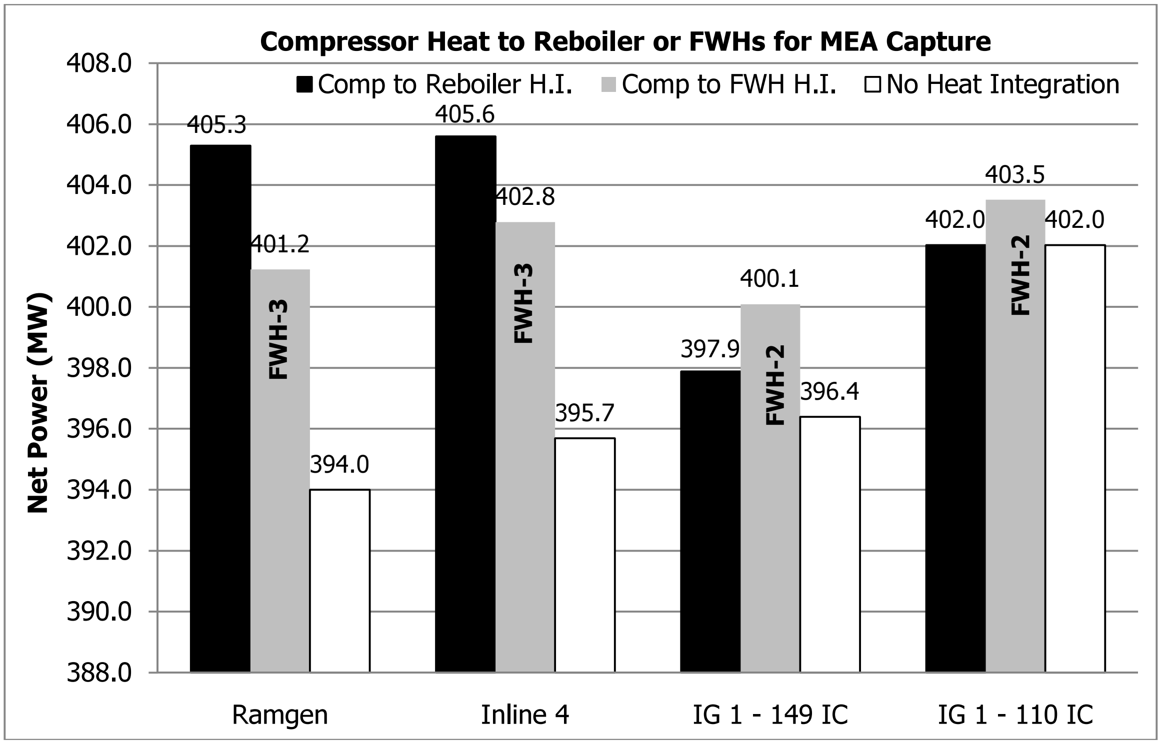


Figure 28. Compressor to Reboiler or FWH Thermal Integration Results for Net Power.

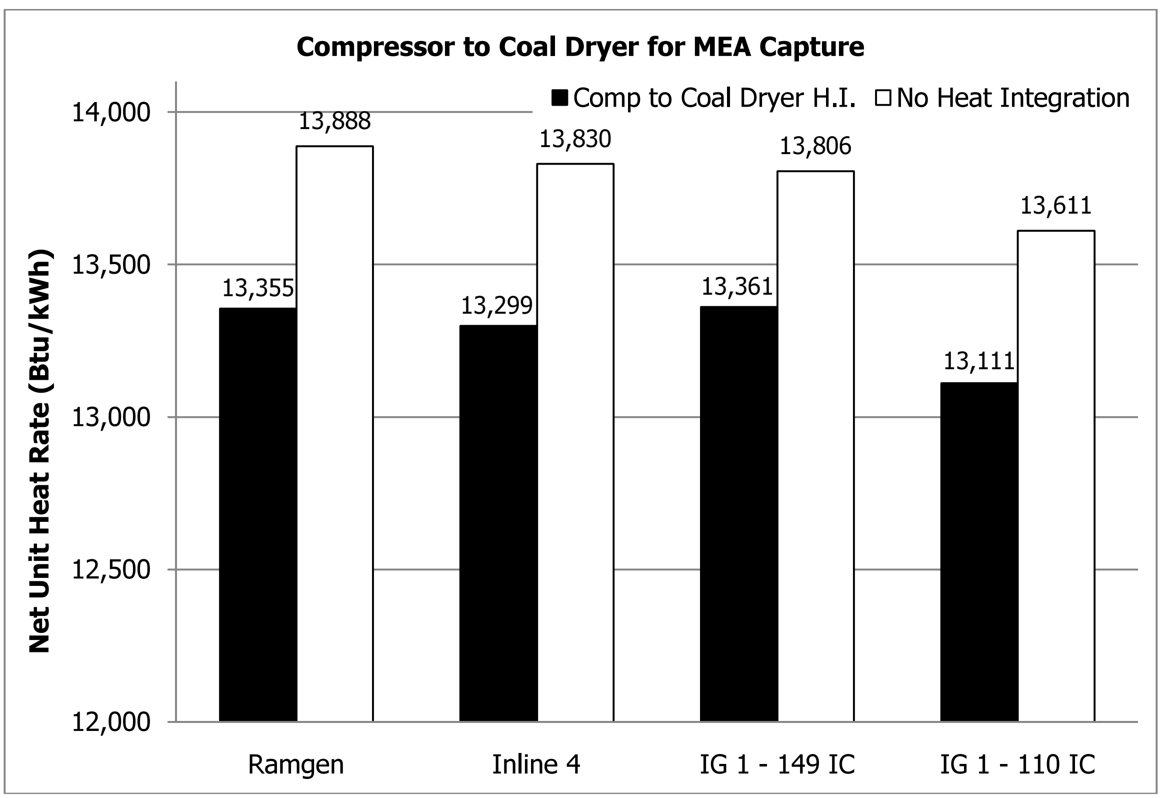


Figure 29. Compressors and Condenser to Coal Dryer Thermal Integration Results for Net Unit Heat Rate.

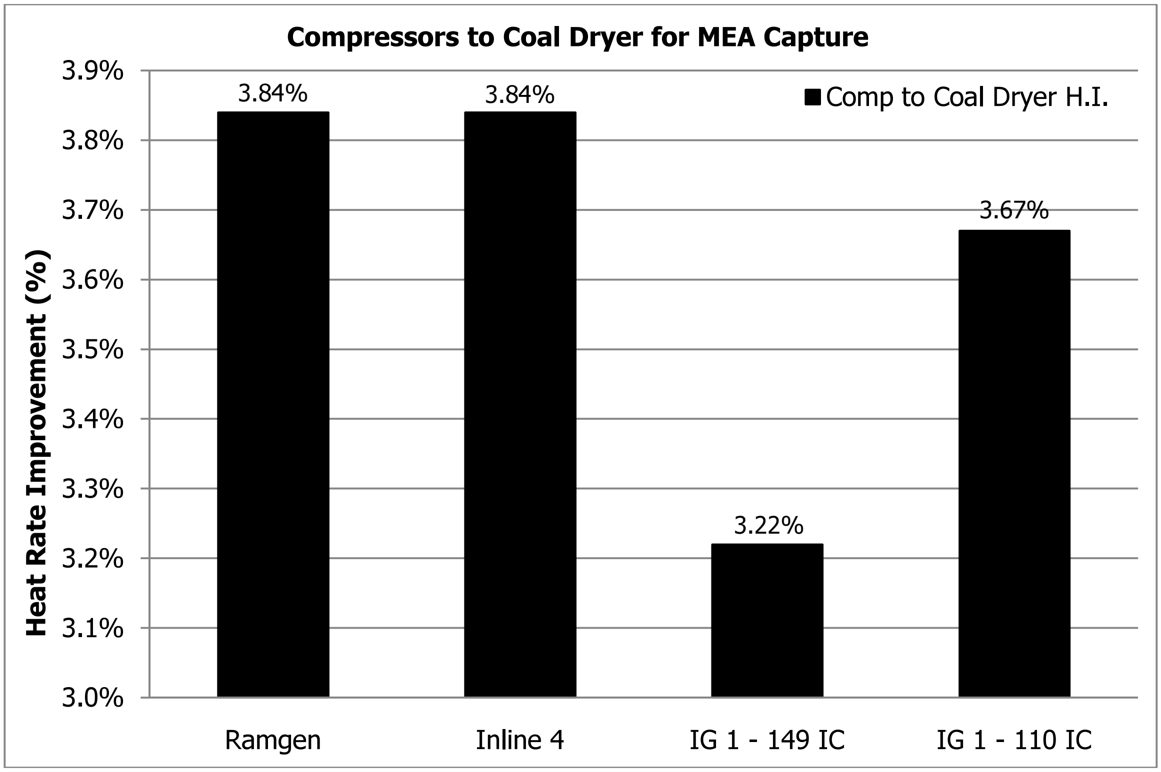


Figure 30. Compressors and Condenser to Coal Dryer Thermal Integration Results for Heat Rate Improvement.

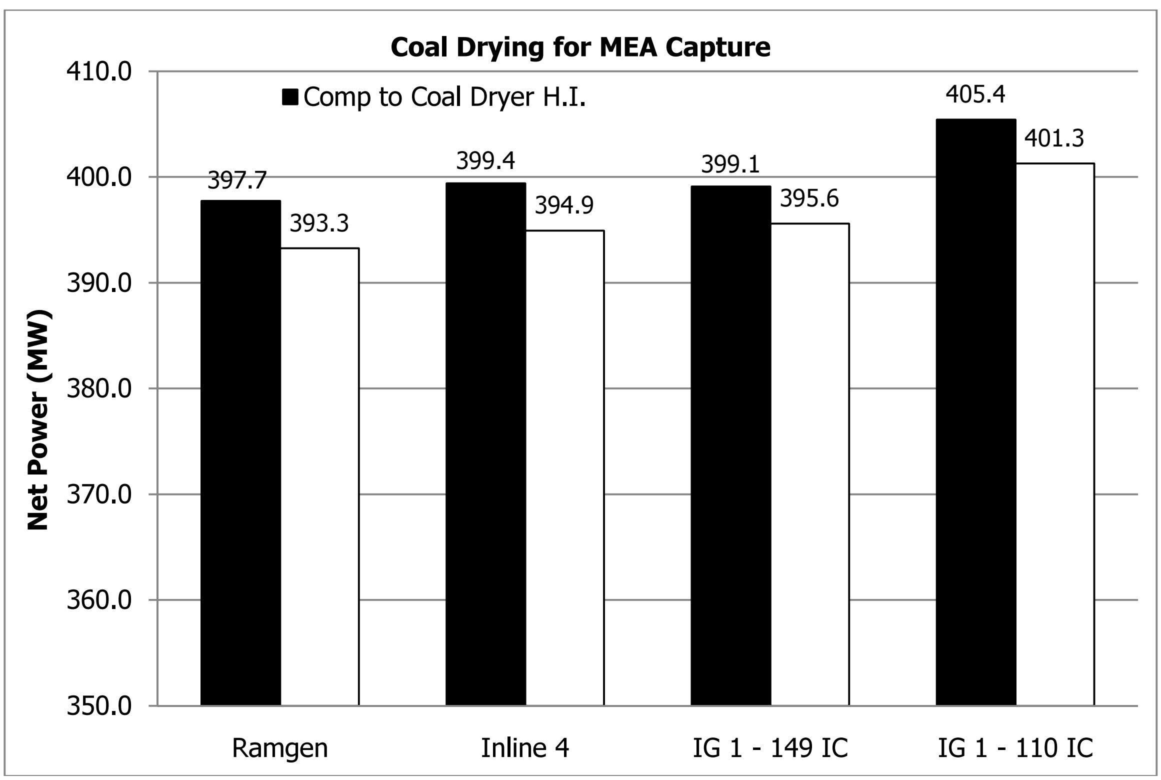


Figure 31. Compressors and Condenser to Coal Dryer Thermal Integration Results for Net Power.

Another way to interpret the results is compressor-specific. For many plants, the compressor is already installed or the plant site limits the compressor options. In all cases, coal drying provides the highest percent heat rate improvement and the lowest heat rate for a given compressor. If a Ramgen or Inline 4 compressor has been selected already, the next best thermal sink for heat rate improvement is the reboiler. If an IG 1 – 149 compressor has been selected already, the next best thermal sink for heat rate improvement is FWH-2. If an IG 1 – 110 compressor has been selected already, the best thermal sink for heat rate improvement is the coal dryer.

The stripper condenser has a larger quantity of thermal energy than the compressor coolers. Two individual cases (FWH-3 and the coal dryer) and one combination thermal integration case (condenser to FWH-3 and compressor to reboiler) were evaluated. The FWH-3 and combination results are shown in Figure 32, Figure 33, and Figure 34. The coal drying results are shown in Figure 35, Figure 36, and Figure 37. For the individual cases, the best heat rate was obtained with the thermal integration to FWH-3. The condenser to coal drying was only performed with the Ramgen compressor model since the amount of coal drying is not dependent on the compressor selected. However, this thermal integration case did not reduce the heat rate as much as the integration to FWH-3. The combination thermal integration case, with the condenser to FWH-3 and the compressor to reboiler, is the best thermal integration case. This is because the results for net power, unit efficiency, and net unit heat rate are the sum of the respective results from the individual cases for each compressor.

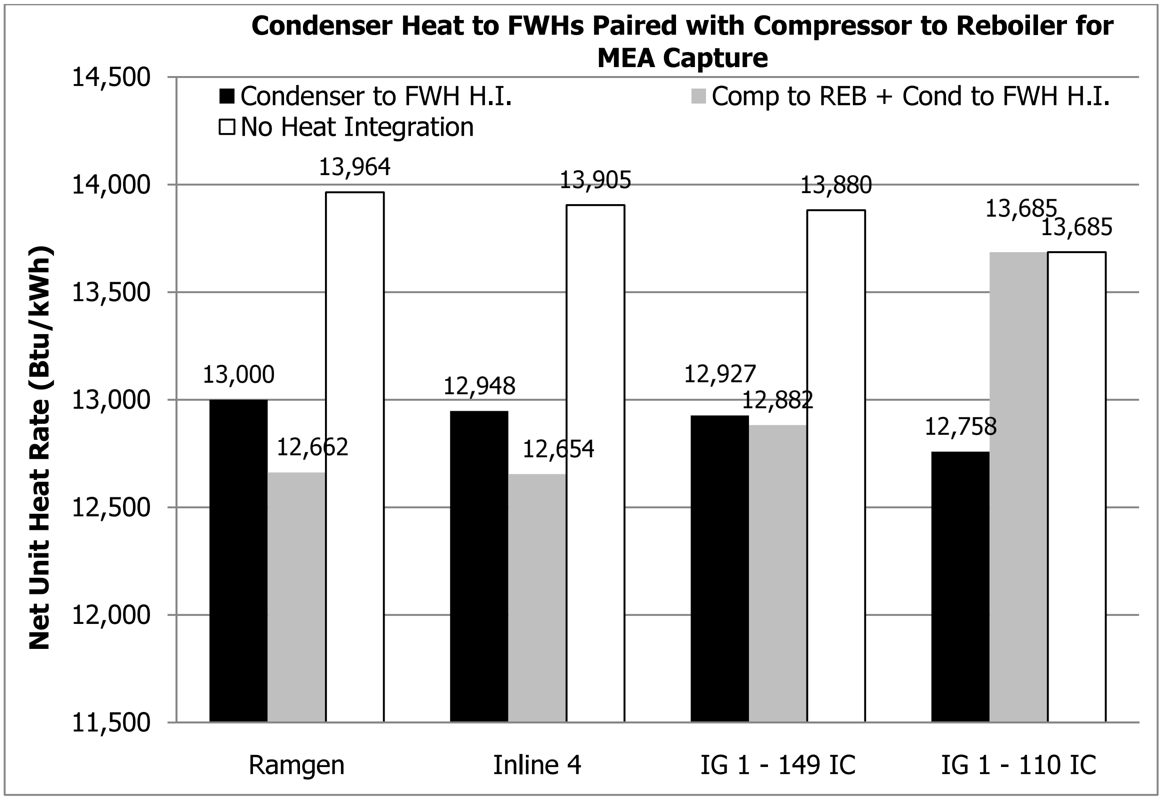


Figure 32. Condenser Thermal Integration Results for Net Unit Heat Rate.

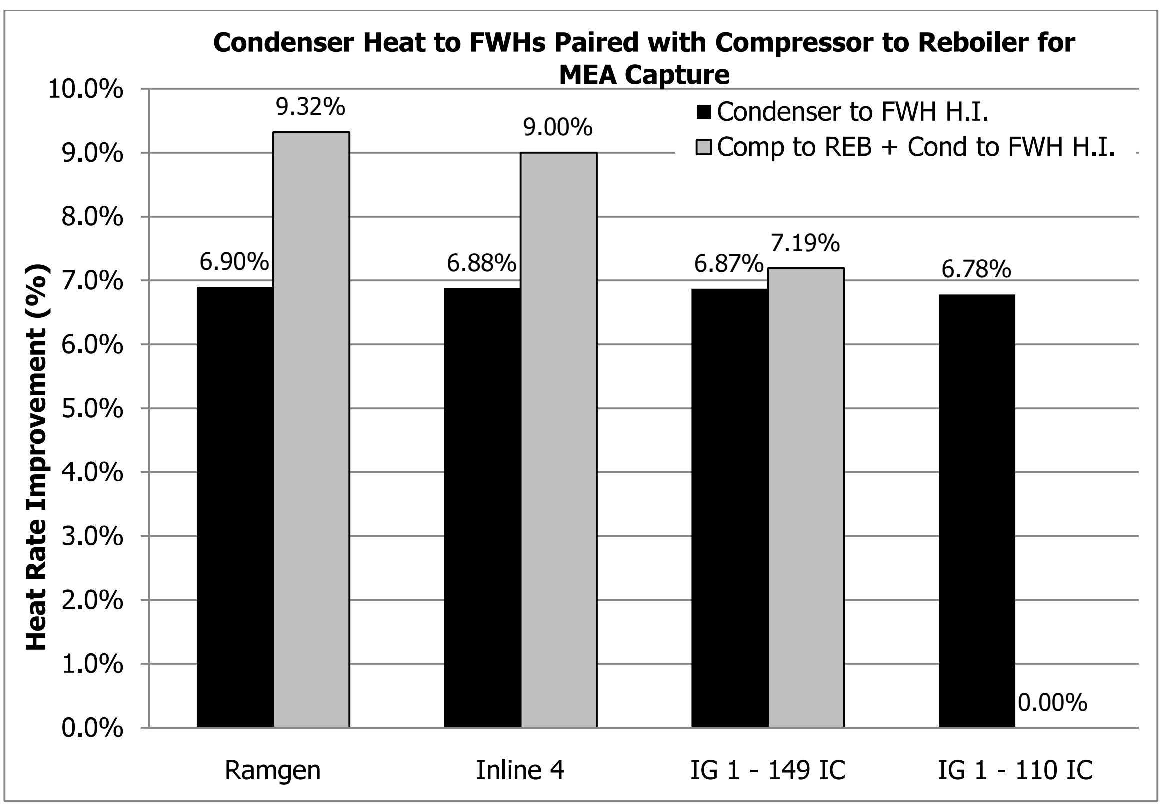


Figure 33. Condenser Thermal Integration Results for Heat Rate Improvement.

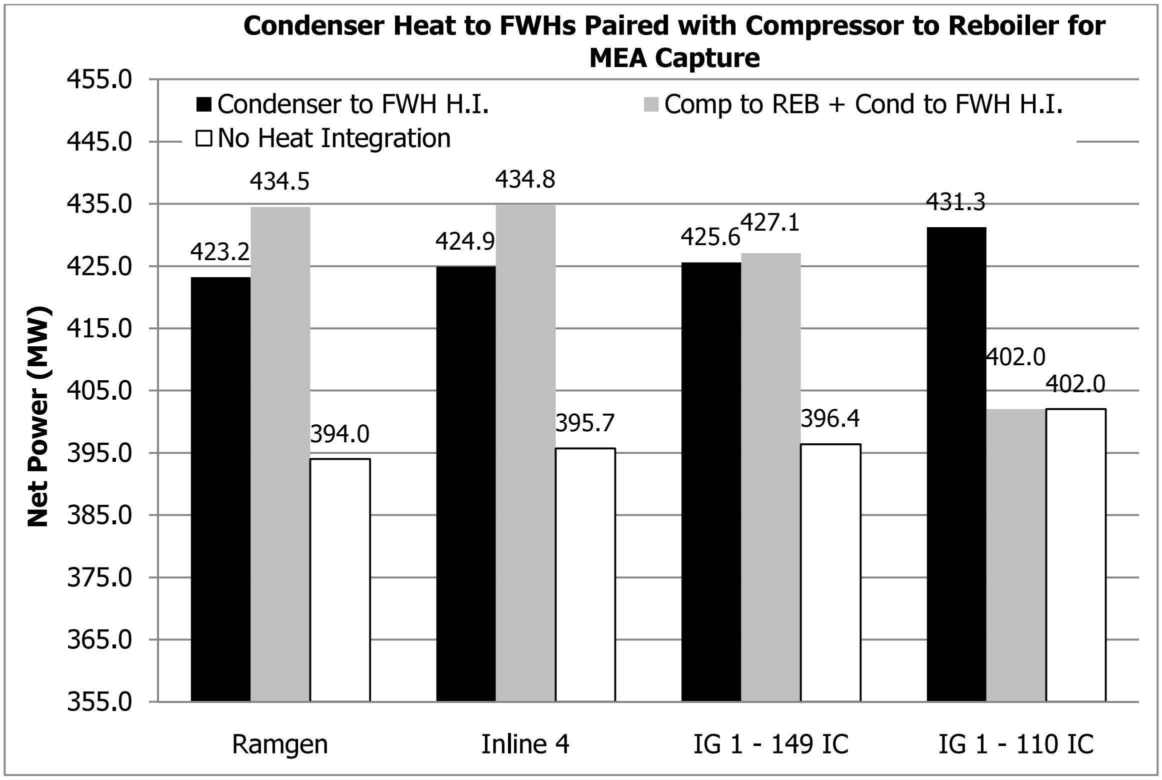


Figure 34. Condenser Thermal Integration Results for Net Power.

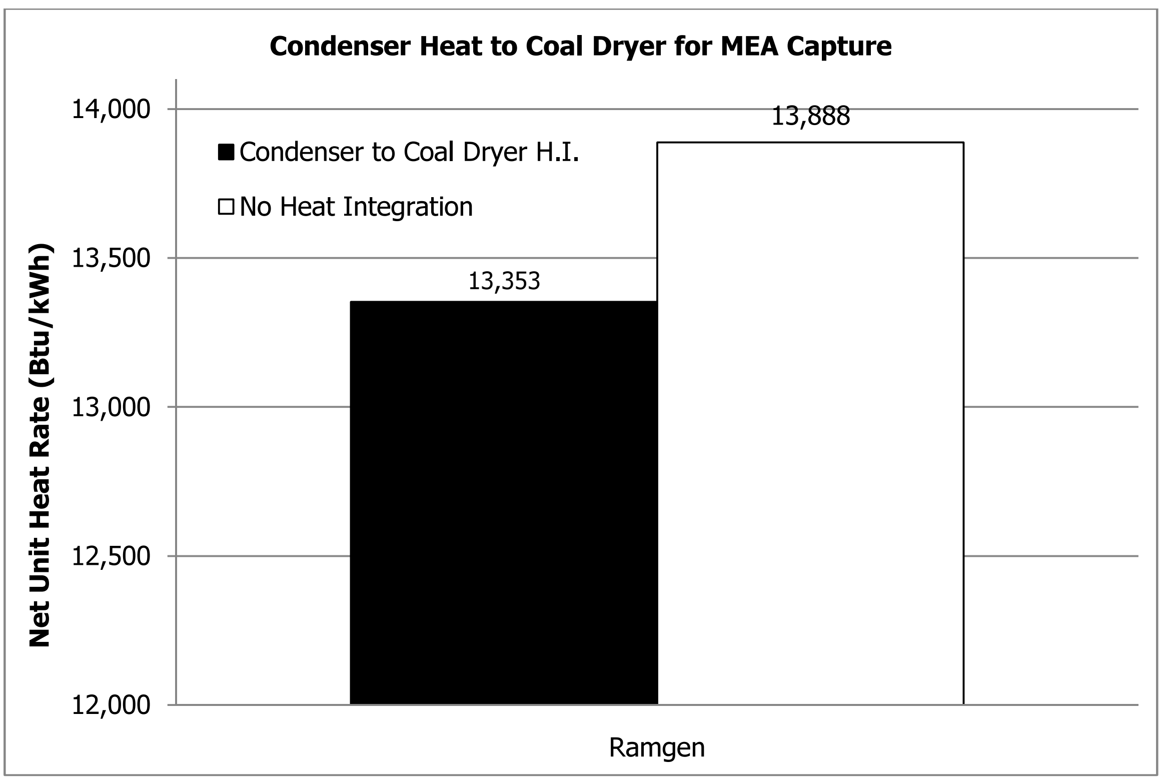


Figure 35. Condenser to Coal Dryer Thermal Integration Results for Net Unit Heat Rate.

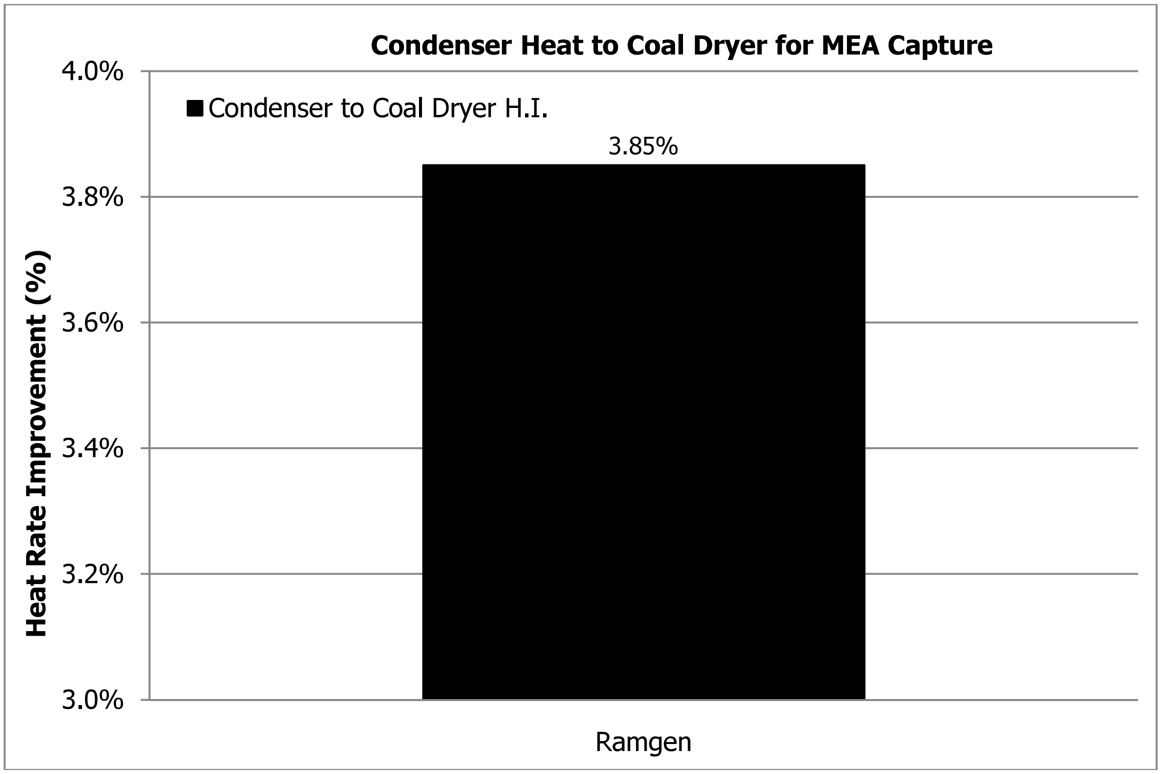


Figure 36. Condenser to Coal Dryer Thermal Integration Results for Heat Rate Improvement.

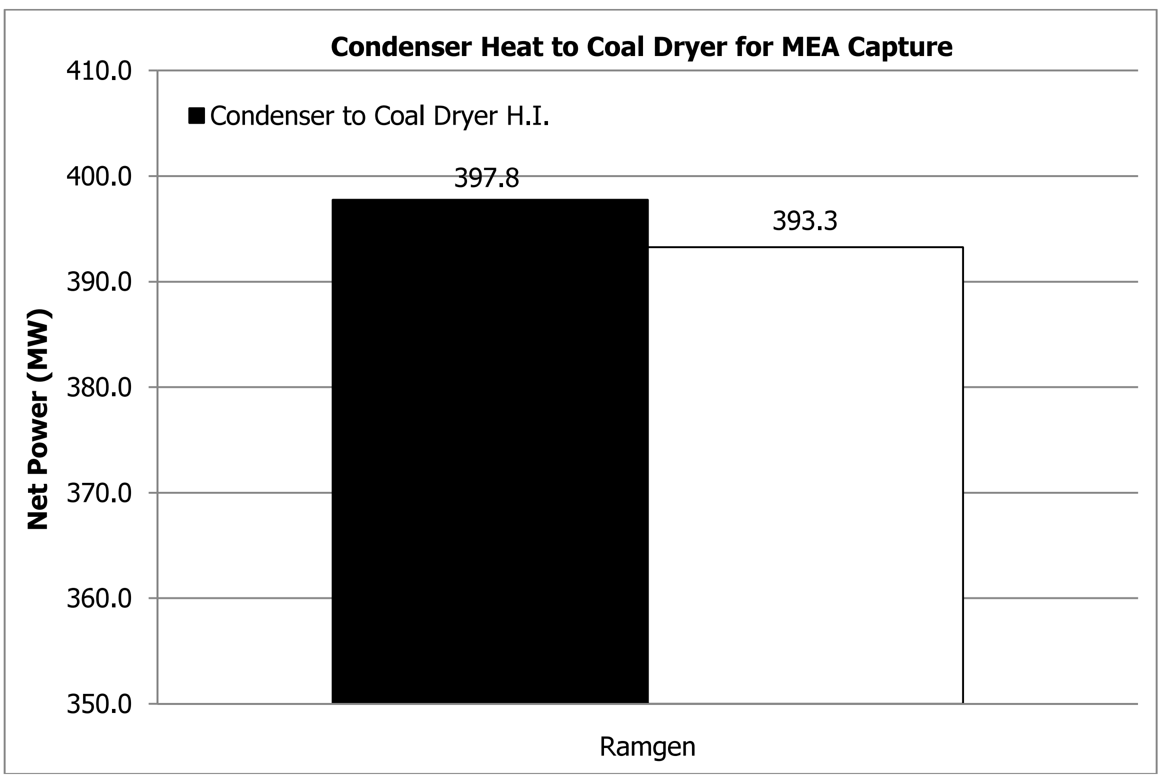


Figure 37. Condenser to Coal Dryer Thermal Integration Results for Net Power.

References

Alie, C. (2004). *CO_2 Capture with MEA: Integrating the Absorption Process and Steam Cycle of an Existing Coal-Fired Power Plant*. Waterloo, Ontario: University of Waterloo.

Aspen Technology, Inc. (2001). *Aspen Plus 11.1 User Guide*. Cambridge, MA.

Aspen Technology, Inc. (2000). *Aspen Plus Version 10.2, Getting Started Modeling Processes with Solids*. Cambridge, MA.

Cengel, Y. A. & Boles, M. A. (2008). *Thermodynamics: An Engineering Approach (6^{th} Ed.)*. New York: McGraw Hill.

Charles, J. (2011). *An Examination of Heat Rate Improvements Due to Waste Heat Integration in an Oxycombustion Pulverized Coal Power Plant*. Bethlehem, PA: Energy Research Center, Lehigh University.

Charles, J. (2010). *Review of CO_2 Compressors*. Bethlehem, PA: Energy Research Center, Lehigh University.

Drbal, L. P. (1996). *Power Plant Engineering*. New York: Springer Publishing.

Fashami, S. Z. (2007). *CO_2 Removal with 7M Monoethanolamine (MEA) – Aspen Plus Model for Base Case*. Austin, TX: Department of Chemical Engineering, University of Texas.

Fisher, K. S., Beitler, C. et al. (2005). *Integrating MEA Regeneration with CO_2 Compression and Peaking to Reduce CO_2 Capture Costs*. Buda, TX: Trimeric Corporation.

Freguia, S. (2002). *Modeling of CO_2 Removal from Flue Gases with Monoethanolamine*. Austin, TX: University of Texas.

Incropera, F. A. (2007). *Fundamentals of Heat and Mass Transfer (6^{th} Ed.)*. Hoboken, NJ: John Wiley & Sons.

Knudsen, J. N. Jensen, J.N. et al. (2009). Experience with CO_2 Capture from Coal Flue Gas in Pilot-Scale: Testing of Different Amine Solvents. *Energy Procedia*, 1, 783-790.

Levy, E. K. (2010). *Thermal Integration of CO_2 Compression Processes with Coal-Fired Power Plants Equipped with Carbon Capture*. Bethlehem, PA: Energy Research Center, Lehigh University.

National Energy Technology Laboratory (NETL). (2008). *Pulverized Coal Oxycombustion Power Plants Vol. 1: Bituminous Coal to Electricity*. <www.netl.doe.gov/energy-analyses/pubs/PC Oxyfuel Combustion Revised Report 2008.pdf>. Accessed September 2008.

National Energy Technology Laboratory (NETL). (2002). *Aspen Plus Simulation of CO_2 Recovery Process*. Pittsburgh, PA.

National Institute of Standards and Technology's Reference Thermodynamic Properties (REFPROP). Gaithersburg, Maryland.

Sarunac, N. (1993). *HEATRT User's Guide*. Bethlehem, PA: Energy Research Center, Lehigh University.

Szatkowski, A. E. (2009). *Modeling CO_2 Capture from a Supercritical Power Plant with ASPEN Plus*. Bethlehem, PA: Energy Research Center, Lehigh University.

Appendix A – Turbine Kit Diagram

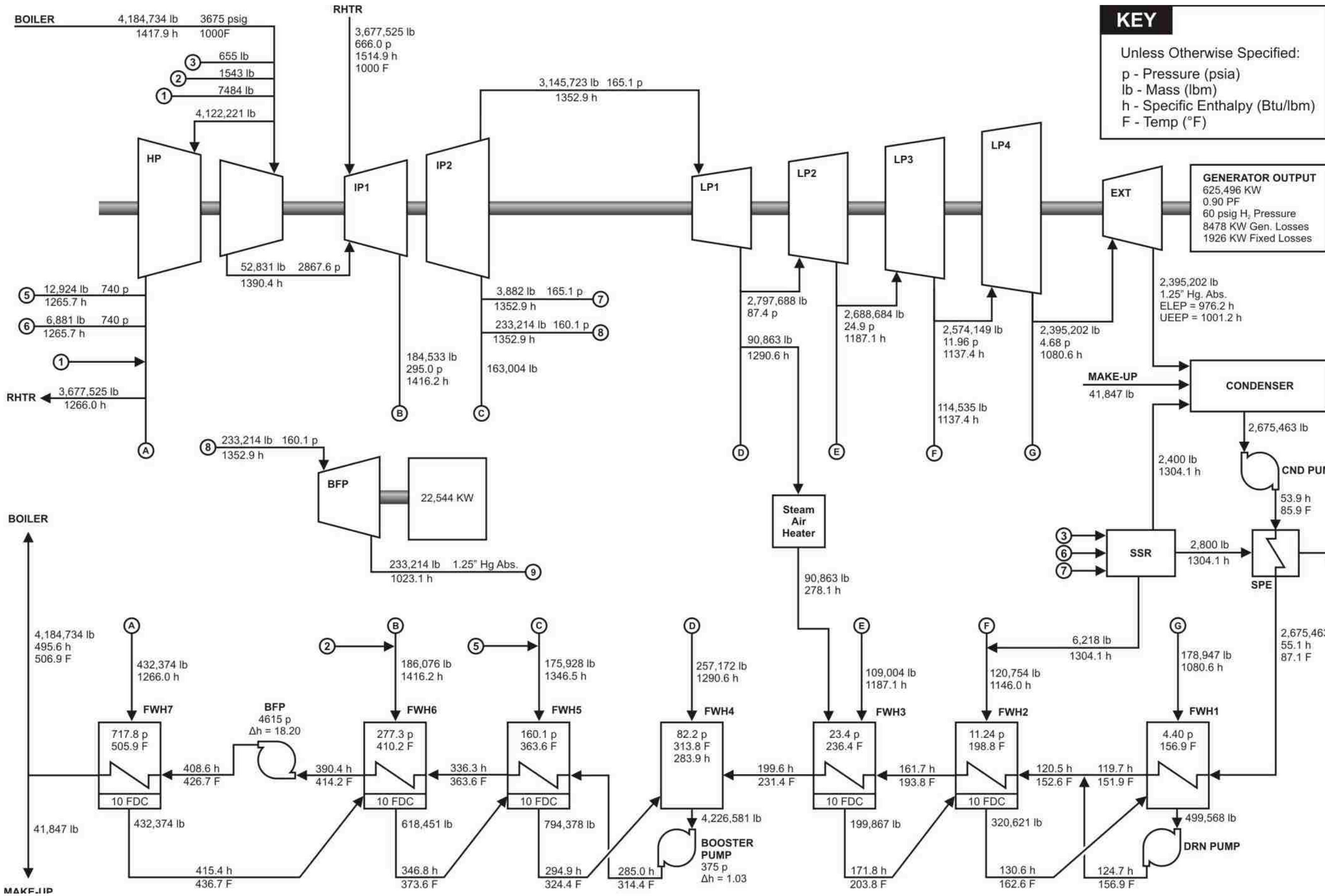


Figure 38. Supercritical Turbine Kit (redrawn).

Appendix B – Aspen Plus Model Diagrams

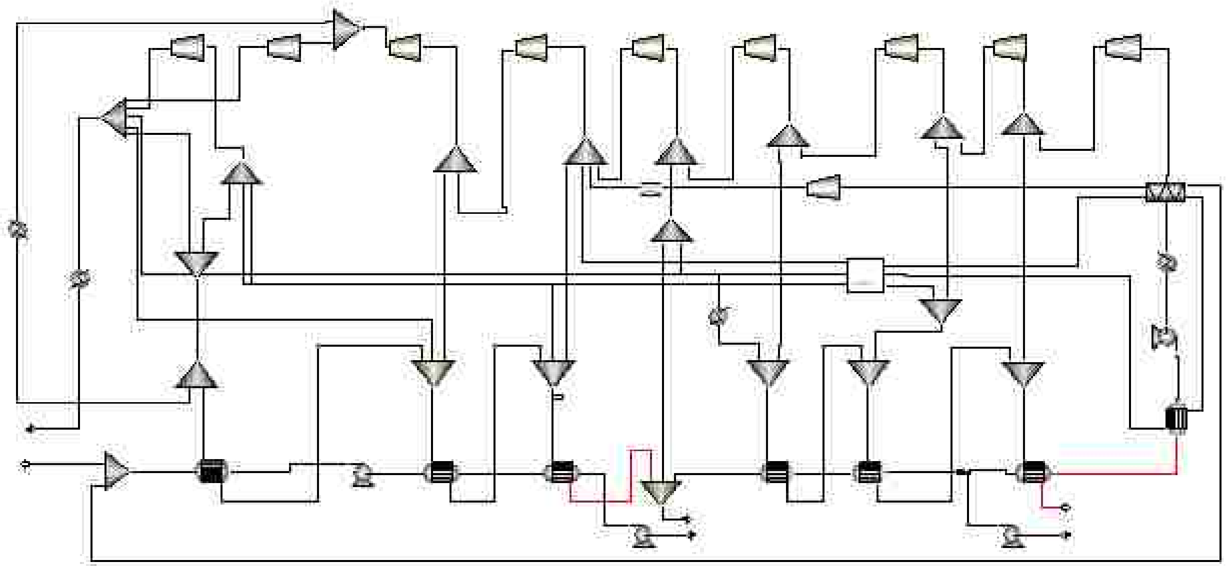


Figure 39. Aspen Plus Model for Supercritical Steam Cycle.

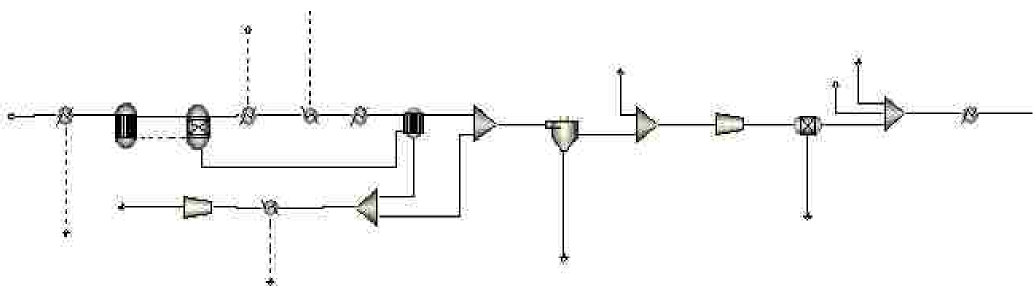


Figure 40. Aspen Plus Model of Boiler and Pollution Control Equipment.

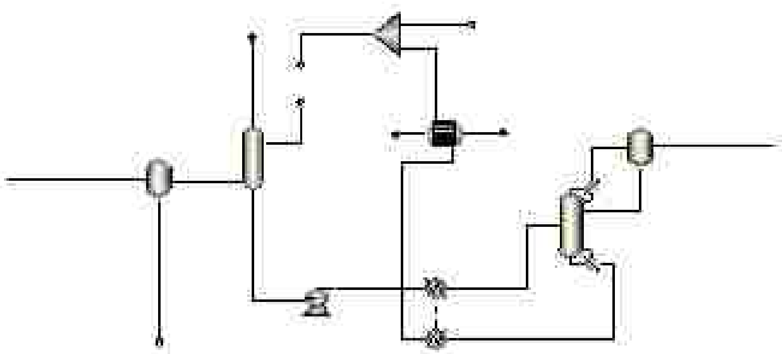


Figure 41. Aspen Plus Model of MEA System.

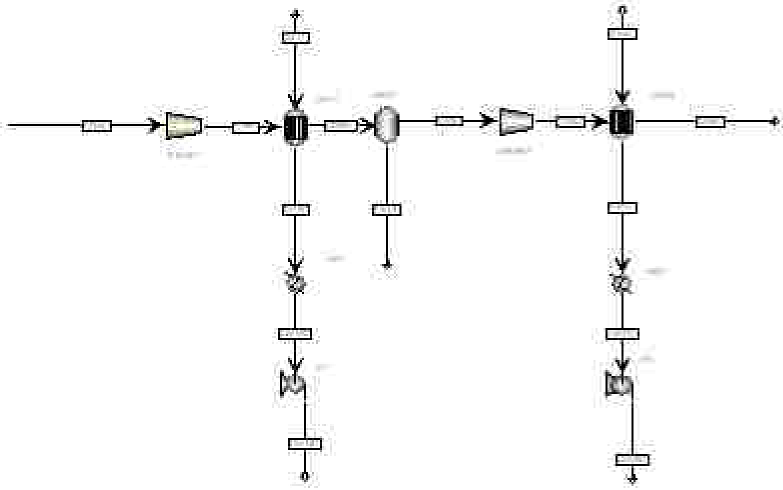


Figure 42. Aspen Plus Model of Ramgen Compressor.

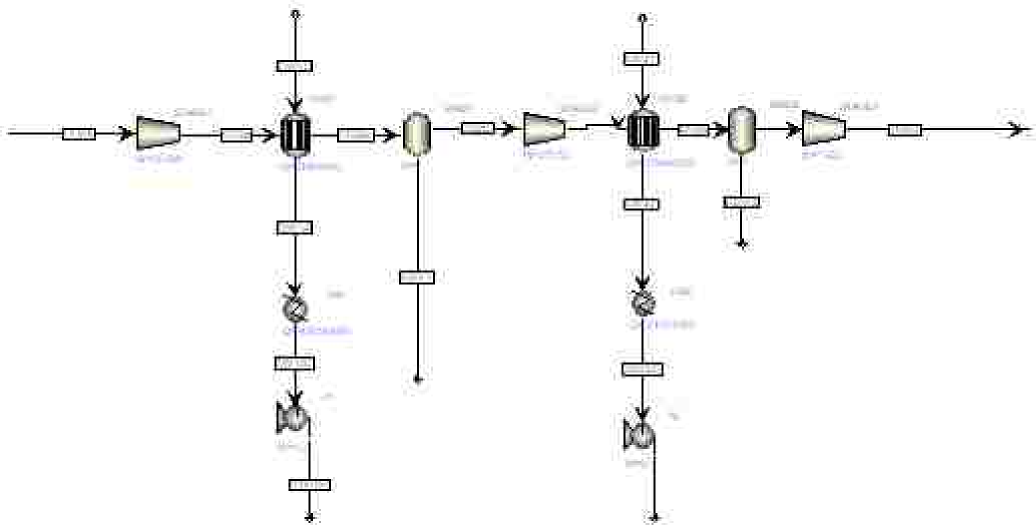


Figure 43. Aspen Plus Model of Inline 4 Compressor.

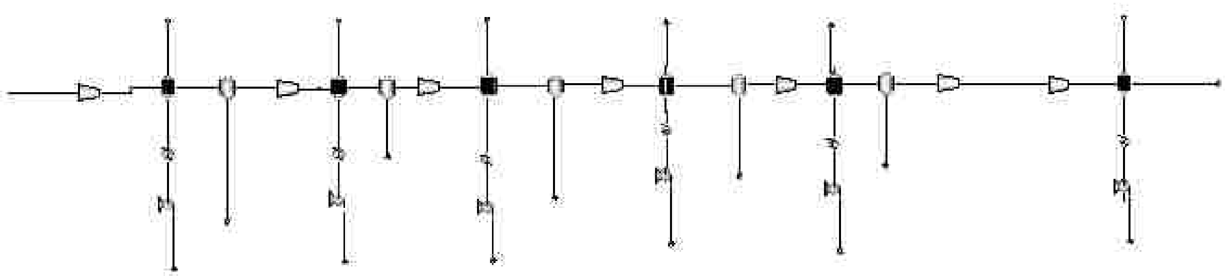


Figure 44. Aspen Plus Model of IG 1 - 149 Compressor.

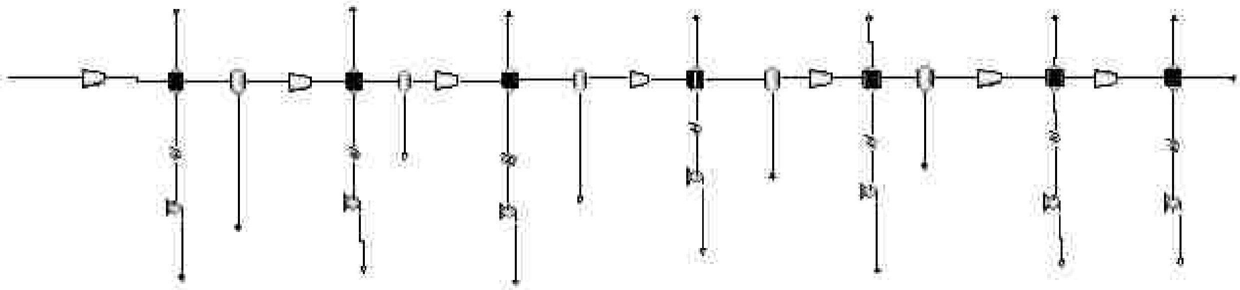


Figure 45. Aspen Plus Model of IG 1 - 110 Compressor.

Appendix C –*Aspen Plus* Settings

Aspen Plus has a variety of property calculation methods and iterative numerical solution methods available. Each of the property methods is intended for a different application. Changing the property method for a given *Aspen Plus* block and set of streams may produce different results. In this analysis, the steam cycle uses STEAMNBS, the boiler uses PR-BM, the MEA cycle uses ELECNRTL with the KEMEA insert and the pollution control equipment and the compressors use RK-SOAVE. STEAMNBS references the ASME Code steam tables and is useful for calculations involving water only. PR-BM uses the Peng-Robinson equation of state with the Boston-Mathias alpha function and it is useful for balancing hydrocarbon separation reactions such as coal comubustion (Aspen Technology 2001). ELECNRTL is used for reactions involving polar ions and electrolyte components. This includes acidic gas absorption into amine solutions, which is the method used by the CO_2 capture system (Aspen Technology 2001). The KEMEA insert changes the kinetic reaction characteristics slightly and produces more accurate results (Alie 2004). RK-SOAVE uses the Redlich-Kwong-Soave equation of state. It is used for hydrocarbon processing applications including the flue gas in the pollution control equipment and compressors.

In order to run a variety of input conditions in Aspen Plus, the software allows users to create design spec and calculator blocks. Design spec blocks allow the user to vary one model variable until a second variable reaches a desired value. Blocks were created for variables in each system and are described below.

In the boiler, the coal flow rate into the boiler is varied in a design block until the temperature of the flue gas exiting the boiler reaches 600°F. This means that *Aspen Plus* iteratively adjusts the coal flow rate until the amount of coal is reconciled with the flue gas exit temperature, the heat added by the mills, the boiler radiation losses, input energy from the combustion air and the throttle and reheat steam flow rates and temperature. This runs simultaneously with another design spec that determines the amount of combustion air required by the boiler. The DS-FGT design spec sets the flue gas temperature exiting the APH at 300°F. The APH-LEAK design spec sets the amount of air leaking through the APH at 6% of the flue gas flow rate.

Table 54. Coal Mass Flow Design Block.

Design Spec:	COALBURN
Define:	
Flowsheet variable	Definition
FGTEMP	Stream-Var Stream=PROD3 Substream=MIXED Variable=TEMP Units=F
Spec:	Tolerance:
FGTEMP = 600	0.001
Vary:	
Mass-Flow Stream=COAL Substream=NCPSD Component=COAL Units=lb/hr	
Limits:	
Lower:	100
Upper:	1.8 E6

Table 55. Combustion Air Mass Flow Design Block.

Design Spec:	O2
Define:	
Flowsheet variable	Definition
O2LEV	Mole-Frac Stream=PROD2 Substream=MIXED Component=O2
Spec:	Tolerance:
O2LEV = 0.035	0.0001
Vary:	
Stream-Var Stream=AIR Substream=MIXED Variable=MASS-FLOW Units=lb/hr	
Limits:	
Lower:	1.0 E4
Upper:	1.7 E7

Table 56. Flue Gas Temperature Design Block.

Design Spec:	DS-FGT
Define:	
Flowsheet variable	Definition
FGT	Stream-Var Stream=PROD4 Substream=MIXED Variable=TEMP Units=F
Spec:	Tolerance:
FGT = 300	0.1
Vary:	
Block-Var Block=APH Variable=T-HOT Sentence=PARAM Units=F	
Limits:	
Lower:	300
Upper:	400

Table 57. APH Air Leakage Rate Design Block.

Design Spec:	APH-LEAK
Define:	
Flowsheet variable	Definition
APHLK	Stream-Var Stream=LEAK Substream=MIXED Variable=MASS-FLOW Units=lb/hr
FGAS	Stream-Var Stream=PROD4 Substream=MIXED Variable=MASS-FLOW Units=lb/hr
Spec:	Tolerance:
APHLK = 0.06*FGAS	0.01
Vary:	
Block-Var Block=SPLT-APH Variable=FLOW/FRAC Sentence=FLOW/FRAC ID1=LEAK	
Limits:	
Lower:	0.02
Upper:	0.2

For the pollution control equipment, the ESP-LEAK design spec sets the amount of air leaking into the ESP at 5% of the flue gas flow rate. The FGD-AIR design spec sets the air leaking into the FGD at approximately 1.07% of the flue gas flow rate. The FGD-H2O design spec sets the water added to the flue gas stream in the FGD at approximately 8.02% of the flue gas flow rate. The relationships in the FGD design blocks were calculated based on stream data from the air-fired reference case in NETL (2008).

Table 58. ESP Air Leakage Rate Design Block.

Design Spec:	ESP-LEAK
Define:	
Flowsheet variable	Definition
PROD5	Stream-Var Stream=PROD5 Substream=MIXED Variable=MASS-FLOW Units=lb/hr
AIRLK	Stream-Var Stream=AIR-LEAK Substream=MIXED Variable=MASS-FLOW Units=lb/hr
Spec:	Tolerance:
AIRLK = PROD5*0.05	0.01
Vary:	
Stream-Var Stream=AIR-LEAK Substream=MIXED Variable=MASS-FLOW Units=lb/hr	
Limits:	
Lower:	1.0 E3
Upper:	1.0 E7

Table 59. FGD Air Leakage Design Block.

Design Spec:	FGD-AIR
Define:	
Flowsheet variable	Definition
FGDAIR	Stream-Var Stream=AIR-FGD Substream=MIXED Variable=MOLE-FLOW Units=lbmol/hr
FGAS3	Stream-Var Stream=FGAS3 Substream=MIXED Variable=MOLE-FLOW Units=lbmol/hr
Spec:	Tolerance:
FGDAIR = FGAS3*0.01068908	0.01
Vary:	
Stream-Var Stream=AIR-FGD Substream=MIXED Variable=MASS-FLOW Units=lb/hr	
Limits:	
Lower:	100
Upper:	1.0 E7

Table 60. FGD Additional Water Design Spec.

Design Spec:	FGD-H2O
Define:	
Flowsheet variable	Definition
FGDH2O	Stream-Var Stream=H2O-FGD Substream=MIXED Variable=MOLE-FLOW Units=lbmol/hr
FGAS3	Stream-Var Stream=FGAS3 Substream=MIXED Variable=MOLE-FLOW Units=lbmol/hr
Spec:	Tolerance:
FGDH2O = FGAS3*0.080150025	0.01
Vary:	
Stream-Var Stream=H2O-FGD Substream=MIXED Variable=MASS-FLOW Units=lb/hr	
Limits:	
Lower:	100
Upper:	1.0 E7

The performance of the MEA system is controlled by four separate design spec blocks. First, AMINEFLR sets the flow rate of the MEA and water mixture at four times the flue gas flow rate entering the absorber. CO2CAP varies the CO_2 content of the MEA solution entering the absorber in order to achieve a 90% CO_2 capture rate. Different capture rates can be achieved by changing the spec in this block. CO2LD varies the duty of the reboiler to ensure the CO_2 content of the lean amine and the initial MEA solution is equal. LPS-FLR calculates the required mass flow rate of the reboiler steam extraction to deliver enough heat to the reboiler.

Table 61. Amine Flow Rate Design Block.

Design Spec:	AMINEFLR
Define:	
Flowsheet variable	Definition
AMINEF	Stream-Var Stream=AMINE Substream=MIXED Variable=MASS-FLOW Units=lb/hr
FGFLR	Stream-Var Stream=FG-COLD Substream=MIXED Variable=MASS-FLOW Units=lb/hr
Spec:	Tolerance:
AMINEF = 4*FGFLR	50
Vary:	
Stream-Var Stream=AMINE Substream=MIXED Variable=MASS-FLOW Units=lb/hr	
Limits:	
Lower:	1000
Upper:	1.0 E8

Table 62. CO_2 Capture Rate Design Block.

Design Spec:	CO2CAP
Define:	
Flowsheet variable	Definition
CO2LOD	Mole-Flow Stream=AMINE Substream=MIXED Component=CO2 Units=lbmol/hr
CO2IN	Mole-Flow Stream=FG-COLD Substream=MIXED Component=CO2 Units=lbmol/hr
CO2OUT	Mole-Flow Stream=CO2-DRY Substream=MIXED Component=CO2 Units=lbmol/hr
Spec:	Tolerance:
CO2OUT/CO2IN = 0.9	1
Vary:	
Mole-Flow Stream=AMINE Substream=MIXED Component=CO2 Units=lbmol/hr	
Limits:	
Lower:	10000
Upper:	15000

Table 63. CO_2 Content in Lean Amine Stream Design Block.

Design Spec:	CO2LD
Define:	
Flowsheet variable	Definition
CO2IN	Mole-Flow Stream=AMINE Substream=MIXED Component=CO2 Units=lbmol/hr
CO2RTN	Mole-Flow Stream=HOTLEAN Substream=MIXED Component=CO2 Units=lbmol/hr
QREB	Block-Var Block=STRIPPER Variable=REB-DUTY Sentence=RESULTS Units=Btu/hr
Spec:	Tolerance:
CO2RTN = CO2IN	2
Vary:	
Block-Var Block=STRIPPER Variable=REB-DUTY Sentence=COL-SPECS Units=Btu/hr	
Limits:	
Lower:	1.5 E9
Upper:	2.0 E9

Table 64. Reboiler Steam Extraction Flow Rate Design Block.

Design Spec:	LPS-FLR
Define:	
Flowsheet variable	Definition
LPSREB	Block-Var Block=LPS-REB Variable=QCALC Sentence=PARAM Units=Btu/hr
QREB	Block-Var Block=STRIPPER Variable=REB-DUTY Sentence=RESULTS Units=Btu/hr
LPSFLO	Stream-Var Stream=LPS-CO2 Substream=MIXED Variable=MASS-FLOW Units=lb/hr
LPSS5	Block-Var Block=S-5 Variable=FLOW/FRAC Sentence=FLOW/FRAC ID1=LPS-CO2
Spec:	Tolerance:
LPSREB = -QREB	5
Vary:	
Block-Var Block=S-5 Variable=FLOW/FRAC Sentence=FLOW/FRAC ID1=LPS-CO2	
Limits:	
Lower:	1468745
Upper:	1958327

For the thermal integration cases, the cooling water exiting heat exchangers is desired at a high temperature. The heat duty of the heat exchanger is determined by the change in enthalpy of the CO_2 flow rate alone. When the heat duty is fixed, a high exit temperature is produced when the cooling water flow rate is low. However, in a counter-flow heat exchanger, the exit temperature of the cooling water cannot be greater than the CO_2 inlet temperature. For example, CW1FR varies the cooling water flow rate until the temperature difference between the CO_2 inlet temperature and cooling water outlet temperature is 10°F, conservatively. The design block was repeated for each cooling water heat exchanger used in the thermal integration cases.

Table 65. Cooling Water Flow Rate Design Block.

Design Spec:	CW1FR
Define:	
Flowsheet variable	Definition
FCW11	Stream-Var Stream=CW11 Substream=MIXED Variable=MASS-FLOW Units=lb/hr
FCO2B	Stream-Var Stream=CO2B Substream=MIXED Variable=MASS-FLOW Units=lb/hr
TCO2B	Stream-Var Stream=CO2B Substream=MIXED Variable=TEMP Units=F
TCW12	Stream-Var Stream=CW12 Substream=MIXED Variable=TEMP Units=F
Spec:	Tolerance:
TCO2B-TCW12 = 10	0.2
Vary:	
Stream-Var Stream=CW11 Substream=MIXED Variable=MASS-FLOW Units=lb/hr	
Limits:	
Lower:	10000
Upper:	50000

Aspen Plus also has calculator blocks that allow users to write Fortran programming statements using specified variables. The calculator ensures that user-defined relationships are maintained. COMBUST calculates the composition of the coal entering the boiler. QMILL determines the required mill power and QRAD calculates the amount of heat lost in the boiler due to radiation.

Table 66. Mill Power Calculator.

Calculator:	Q-MILL
Define:	
Variable name	Definition
QMILL	Heat-Duty Stream=Q-MILL Units=Btu/hr
MCOAL	Mass-Flow Stream=COAL-IN Substream=NCPSD Component=COAL Units=lb/hr
Calculate:	
QMILL=MCOAL*18.06	
Sequence:	
Before Unit Operation PULV	

Table 67. Boiler Radiation Losses Calculator.

Calculator:	Q-RAD
Define:	
Variable name	Definition
MCOAL	Stream-Var Stream=COAL-IN Substream=NCPSD Variable=MASS-FLOW Units=lb/hr
QRAD	Block-Var Block=RAD Variable=DUTY Sentence=PARAM Units=Btu/hr
Calculate:	
QRAD=-(MCOAL*8426*0.008)	
Sequence:	
Before Unit Operation RAD	

Table 68. Coal Composition Normalization Calculator.

Calculator:	COMBUST
Define:	
Variable name	Definition
ULT	Compattr-Vec Stream=COAL Substream=NCPSD Component=COAL Attribute=ULTANAL
WATER	Compattr-Var Stream=COAL Substream=NCPSD Component=COAL Attribute=PROXANAL Element=1
H2O	Block-Var Block=DECOMP Variable=MASS-YIELD Sentence=MASS-YIELD ID1=H2O ID2=MIXED
ASH	Block-Var Block=DECOMP Variable=MASS-YIELD Sentence=MASS-YIELD ID1=ASH ID2=NCPSD
CARB	Block-Var Block=DECOMP Variable=MASS-YIELD Sentence=MASS-YIELD ID1=C ID2=CIPSD
H2	Block-Var Block=DECOMP Variable=MASS-YIELD Sentence=MASS-YIELD ID1=H2 ID2=MIXED
N2	Block-Var Block=DECOMP Variable=MASS-YIELD Sentence=MASS-YIELD ID1=N2 ID2=MIXED
CL2	Block-Var Block=DECOMP Variable=MASS-YIELD Sentence=MASS-YIELD ID1=CL2 ID2=MIXED
SULF	Block-Var Block=DECOMP Variable=MASS-YIELD Sentence=MASS-YIELD ID1=S ID2=MIXED
O2	Block-Var Block=DECOMP Variable=MASS-YIELD Sentence=MASS-YIELD ID1=O2 ID2=MIXED
Calculate:	
FACT=(100-WATER)/100	
H2O=WATER/100	
ASH=ULT(1)/100*FACT	
CARB=ULT(2)/100*FACT	
H2=ULT(3)/100*FACT	
N2=ULT(4)/100*FACT	
CL2=ULT(5)/100*FACT	
SULF=ULT(6)/100*FACT	
O2=ULT(7)/100*FACT	
Sequence:	
Before Unit Operation DECOMP	

Appendix D – REFPROP vs Aspen Plus CO_2 Properties

Table 69. Pure CO_2 *Aspen Plus* and REFPROP Entropy Values.

Entropy Values [Btu/lb-R]									
Temp.	P = 200 psia			P = 600 psia			P = 1000 psia		
[F]	Aspen	REFPROP	Diff.	Aspen	REFPROP	Diff.	Aspen	REFPROP	Diff.
100	0.5386	0.5386	0.00%	0.4721	0.4720	0.04%	0.4205	0.4196	0.22%
125	0.5484	0.5485	-0.02%	0.4846	0.4850	-0.08%	0.4417	0.4422	-0.12%
150	0.5579	0.5581	-0.03%	0.4961	0.4967	-0.13%	0.4576	0.4586	-0.23%
175	0.5671	0.5672	-0.03%	0.5068	0.5076	-0.15%	0.4711	0.4724	-0.28%
200	0.5759	0.5761	-0.02%	0.5169	0.5177	-0.15%	0.4832	0.4846	-0.28%
225	0.5845	0.5846	-0.01%	0.5266	0.5273	-0.14%	0.4944	0.4957	-0.27%
250	0.5929	0.5929	0.00%	0.5358	0.5365	-0.13%	0.5048	0.5060	-0.25%
275	0.6011	0.6010	0.01%	0.5447	0.5453	-0.11%	0.5146	0.5158	-0.22%
300	0.6090	0.6089	0.02%	0.5533	0.5538	-0.09%	0.5239	0.5250	-0.20%
325	0.6167	0.6166	0.03%	0.5616	0.5620	-0.07%	0.5329	0.5338	-0.17%
350	0.6243	0.6240	0.04%	0.5696	0.5699	-0.05%	0.5415	0.5422	-0.14%
375	0.6316	0.6314	0.04%	0.5774	0.5777	-0.04%	0.5497	0.5504	-0.12%
400	0.6389	0.6385	0.05%	0.5850	0.5852	-0.02%	0.5577	0.5582	-0.09%
425	0.6459	0.6456	0.05%	0.5924	0.5925	-0.01%	0.5655	0.5659	-0.07%
450	0.6528	0.6524	0.06%	0.5996	0.5996	0.00%	0.5730	0.5733	-0.05%
475	0.6596	0.6592	0.06%	0.6066	0.6066	0.02%	0.5803	0.5805	-0.03%
500	0.6662	0.6658	0.06%	0.6135	0.6134	0.02%	0.5874	0.5875	-0.02%
Temp.	P = 400 psia			P = 800 psia			P = 1200 psia		
[F]	Aspen	REFPROP	Diff.	Aspen	REFPROP	Diff.	Aspen	REFPROP	Diff.
100	0.4996	0.4996	0.01%	0.4475	0.4471	0.09%	0.3745	0.3712	0.90%
125	0.5105	0.5107	-0.05%	0.4627	0.4632	-0.10%	0.4193	0.4197	-0.10%
150	0.5208	0.5212	-0.08%	0.4759	0.4767	-0.18%	0.4398	0.4410	-0.27%
175	0.5306	0.5311	-0.08%	0.4878	0.4888	-0.21%	0.4556	0.4572	-0.33%
200	0.5401	0.5405	-0.08%	0.4988	0.4999	-0.22%	0.4691	0.4708	-0.34%
225	0.5492	0.5496	-0.07%	0.5091	0.5102	-0.21%	0.4812	0.4828	-0.33%
250	0.5580	0.5583	-0.06%	0.5189	0.5199	-0.19%	0.4924	0.4939	-0.31%
275	0.5664	0.5667	-0.05%	0.5282	0.5291	-0.17%	0.5027	0.5041	-0.28%
300	0.5747	0.5749	-0.03%	0.5372	0.5380	-0.15%	0.5125	0.5138	-0.24%
325	0.5827	0.5828	-0.02%	0.5458	0.5465	-0.12%	0.5218	0.5229	-0.21%
350	0.5905	0.5905	-0.01%	0.5541	0.5546	-0.10%	0.5307	0.5316	-0.18%
375	0.5981	0.5980	0.00%	0.5621	0.5626	-0.08%	0.5392	0.5400	-0.15%
400	0.6055	0.6054	0.02%	0.5699	0.5702	-0.06%	0.5474	0.5481	-0.12%
425	0.6127	0.6125	0.03%	0.5775	0.5777	-0.04%	0.5553	0.5559	-0.10%
450	0.6197	0.6195	0.03%	0.5848	0.5850	-0.02%	0.5630	0.5634	-0.07%
475	0.6266	0.6264	0.04%	0.5920	0.5921	-0.01%	0.5705	0.5708	-0.05%
500	0.6334	0.6331	0.04%	0.5990	0.5990	0.00%	0.5777	0.5779	-0.04%

Entropy Values [Btu/lb-R]									
Temp.	P = 1400 psia			P = 1800 psia			P = 2200 psia		
[F]	Aspen	REFPROP	Diff.	Aspen	REFPROP	Diff.	Aspen	REFPROP	Diff.
100	0.3284	0.3189	2.98%	0.3055	0.3016	1.28%	0.2937	0.2929	0.27%
125	0.3928	0.3922	0.17%	0.3460	0.3405	1.62%	0.3253	0.3223	0.92%
150	0.4218	0.4229	-0.26%	0.3854	0.3844	0.27%	0.3581	0.3554	0.77%
175	0.4408	0.4424	-0.36%	0.4122	0.4132	-0.25%	0.3871	0.3866	0.12%
200	0.4560	0.4578	-0.39%	0.4316	0.4333	-0.39%	0.4098	0.4108	-0.23%
225	0.4692	0.4710	-0.38%	0.4474	0.4493	-0.42%	0.4281	0.4296	-0.37%
250	0.4811	0.4828	-0.35%	0.4611	0.4630	-0.41%	0.4434	0.4453	-0.41%
275	0.4921	0.4937	-0.32%	0.4733	0.4751	-0.39%	0.4570	0.4588	-0.40%
300	0.5023	0.5038	-0.29%	0.4845	0.4862	-0.35%	0.4692	0.4710	-0.38%
325	0.5120	0.5133	-0.25%	0.4949	0.4965	-0.31%	0.4804	0.4820	-0.34%
350	0.5211	0.5223	-0.22%	0.5047	0.5061	-0.27%	0.4908	0.4923	-0.31%
375	0.5299	0.5309	-0.18%	0.5140	0.5153	-0.23%	0.5006	0.5020	-0.27%
400	0.5384	0.5392	-0.15%	0.5229	0.5240	-0.20%	0.5099	0.5111	-0.23%
425	0.5465	0.5471	-0.12%	0.5314	0.5323	-0.16%	0.5188	0.5198	-0.19%
450	0.5543	0.5549	-0.10%	0.5396	0.5403	-0.13%	0.5273	0.5281	-0.16%
475	0.5619	0.5623	-0.07%	0.5475	0.5480	-0.10%	0.5354	0.5361	-0.13%
500	0.5693	0.5696	-0.05%	0.5551	0.5555	-0.08%	0.5433	0.5438	-0.10%
Temp.	P = 1600 psia			P = 2000 psia					
[F]	Aspen	REFPROP	Diff.	Aspen	REFPROP	Diff.			
100	0.3142	0.3081	2.00%	0.2989	0.2968	0.72%			
125	0.3650	0.3606	1.23%	0.3339	0.3295	1.32%			
150	0.4033	0.4038	-0.11%	0.3701	0.3677	0.65%			
175	0.4263	0.4278	-0.35%	0.3989	0.3992	-0.08%			
200	0.4435	0.4453	-0.41%	0.4203	0.4217	-0.33%			
225	0.4580	0.4599	-0.41%	0.4374	0.4392	-0.41%			
250	0.4707	0.4726	-0.39%	0.4520	0.4539	-0.42%			
275	0.4823	0.4841	-0.36%	0.4649	0.4667	-0.40%			
300	0.4930	0.4946	-0.32%	0.4766	0.4783	-0.37%			
325	0.5031	0.5045	-0.28%	0.4874	0.4890	-0.33%			
350	0.5126	0.5139	-0.25%	0.4975	0.4990	-0.29%			
375	0.5216	0.5227	-0.21%	0.5071	0.5084	-0.25%			
400	0.5303	0.5312	-0.18%	0.5162	0.5173	-0.22%			
425	0.5386	0.5394	-0.14%	0.5248	0.5258	-0.18%			
450	0.5466	0.5472	-0.12%	0.5332	0.5340	-0.15%			
475	0.5543	0.5548	-0.09%	0.5412	0.5418	-0.12%			
500	0.5618	0.5622	-0.07%	0.5489	0.5494	-0.09%			

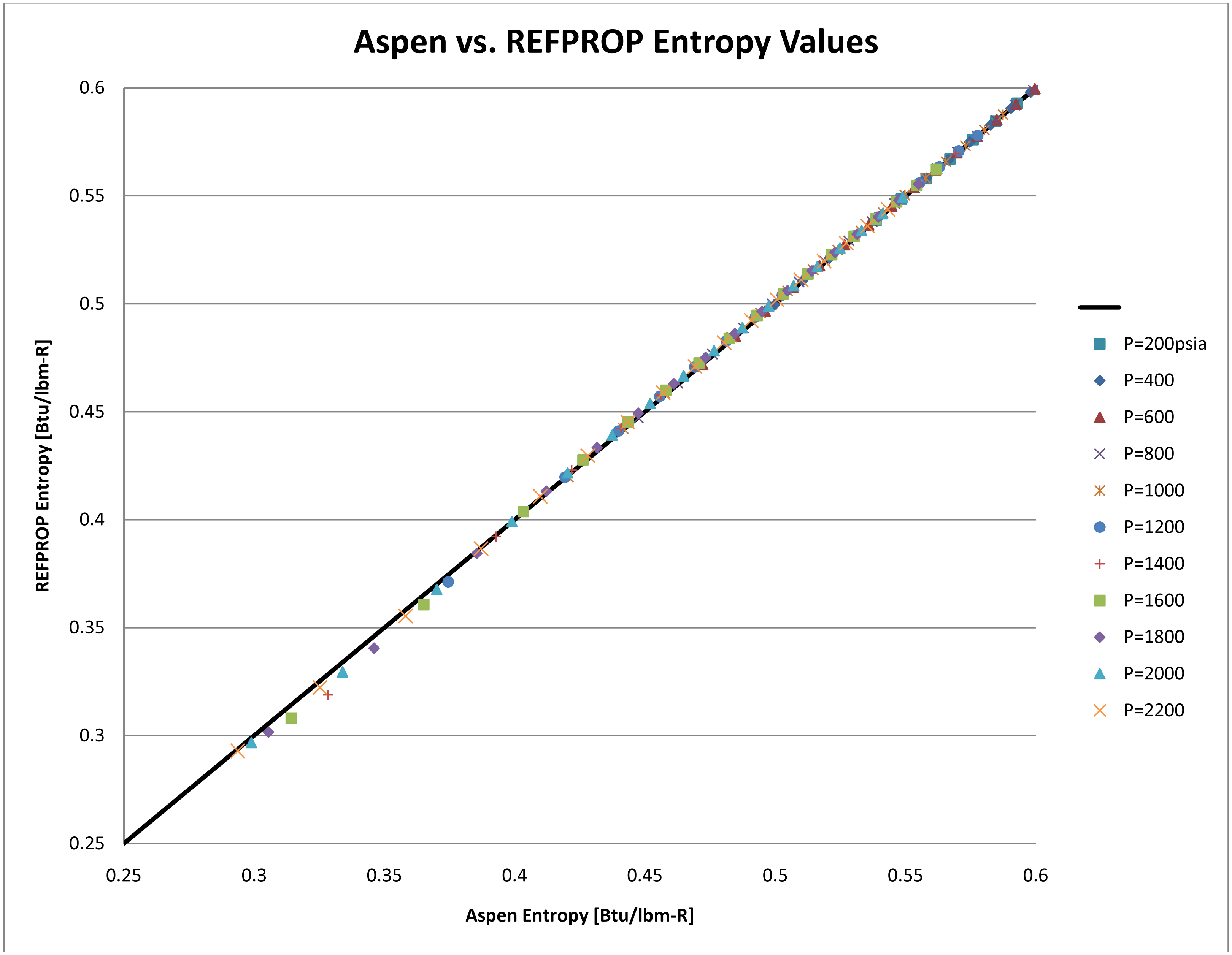


Figure 46. Pure CO_2 *Aspen Plus* versus REFPROP Entropy Values.

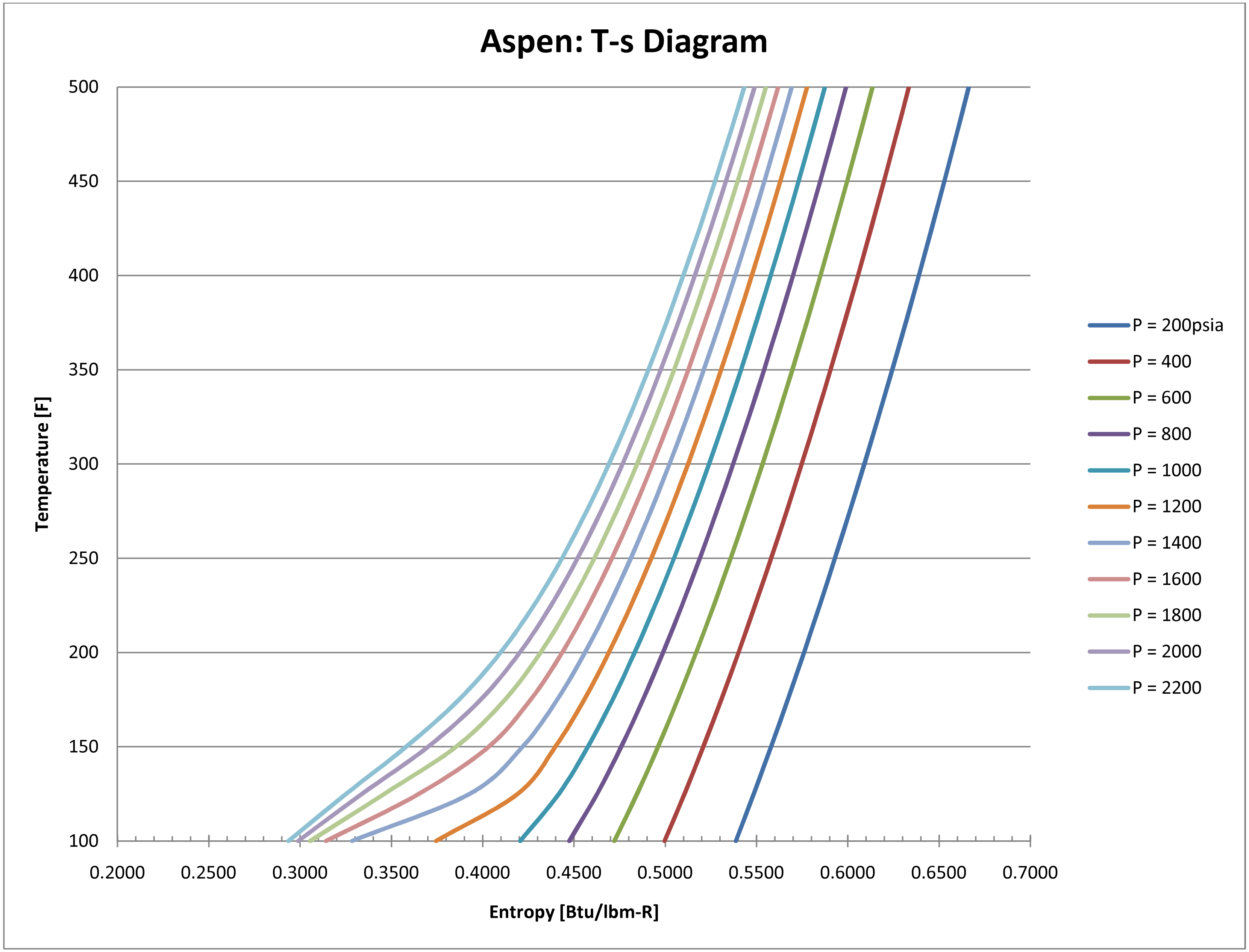


Figure 47. ***Aspen Plus*** **Entropy versus Temperature.**

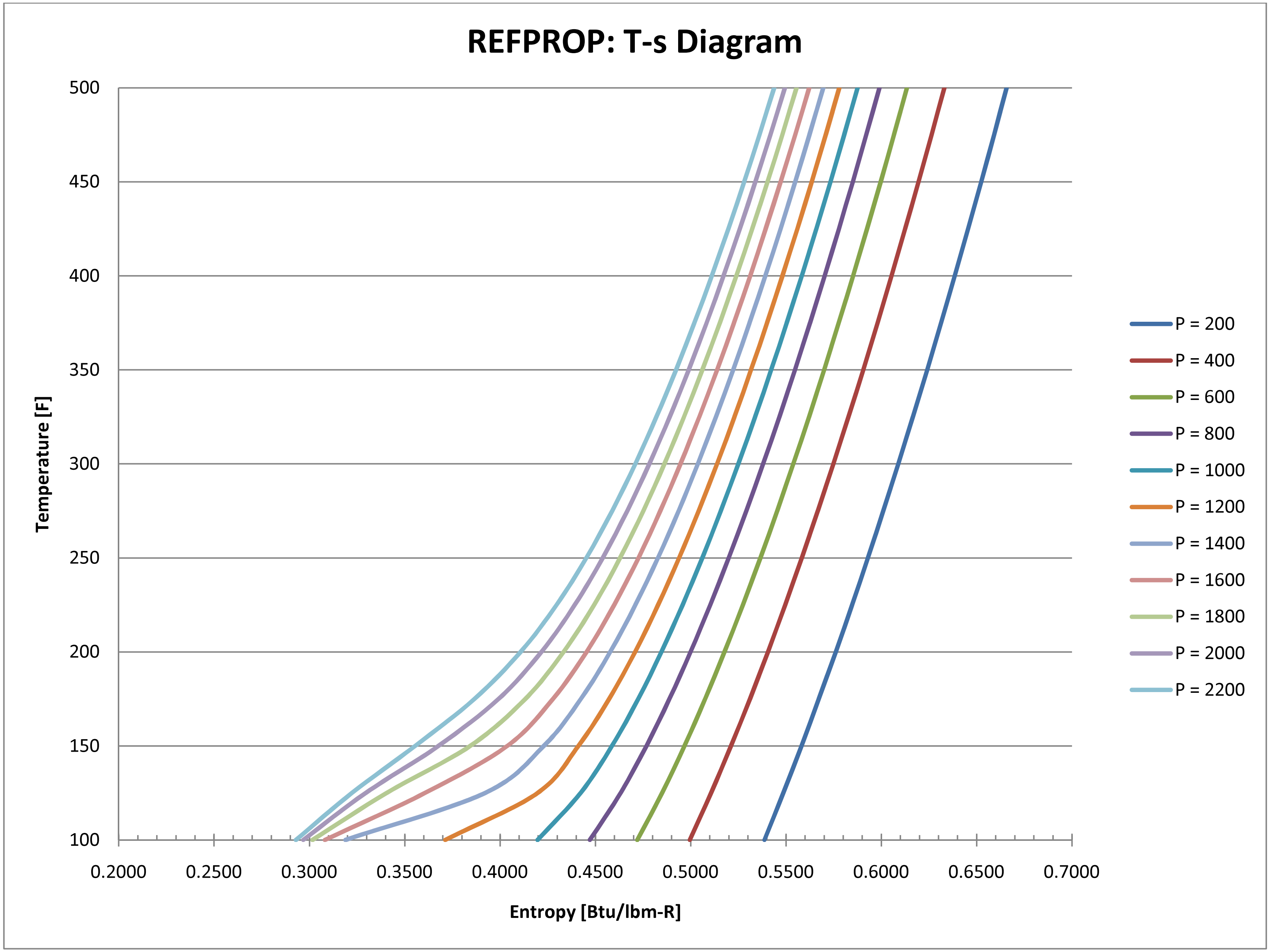


Figure 48. REFPROP Entropy versus Temperature.

Table 70. Pure CO_2 *Aspen Plus* and REFPROP Enthalpy Values.

Enthalpy Values [Btu/lb]									
Temp.	P = 200 psia			P = 600 psia			P = 1000 psia		
[F]	Aspen	REFPROP	Diff.	Aspen	REFPROP	Diff.	Aspen	REFPROP	Diff.
100	217.4	217.4	-0.01%	204.7	204.6	0.05%	185.0	184.4	0.32%
125	223.0	223.1	-0.04%	211.8	212.0	-0.10%	197.1	197.3	-0.12%
150	228.7	228.8	-0.05%	218.7	219.0	-0.17%	206.6	207.1	-0.27%
175	234.4	234.5	-0.05%	225.3	225.8	-0.20%	215.0	215.7	-0.32%
200	240.1	240.2	-0.04%	231.9	232.4	-0.20%	222.8	223.6	-0.33%
225	245.9	246.0	-0.03%	238.4	238.8	-0.18%	230.3	231.1	-0.32%
250	251.7	251.8	-0.01%	244.8	245.2	-0.16%	237.6	238.3	-0.28%
275	257.6	257.6	0.01%	251.3	251.6	-0.13%	244.7	245.3	-0.24%
300	263.5	263.5	0.03%	257.7	257.9	-0.10%	251.7	252.2	-0.20%
325	269.5	269.4	0.04%	264.1	264.3	-0.07%	258.6	259.0	-0.16%
350	275.5	275.4	0.06%	270.5	270.6	-0.04%	265.4	265.7	-0.11%
375	281.6	281.4	0.07%	276.9	276.9	-0.01%	272.2	272.4	-0.07%
400	287.7	287.5	0.08%	283.3	283.3	0.02%	279.0	279.1	-0.03%
425	293.8	293.6	0.09%	289.8	289.6	0.04%	285.7	285.7	0.01%
450	300.0	299.7	0.10%	296.2	296.0	0.06%	292.5	292.4	0.04%
475	306.3	306.0	0.10%	302.7	302.5	0.08%	299.2	299.0	0.07%
500	312.5	312.2	0.11%	309.2	308.9	0.09%	306.0	305.7	0.10%
Temp.	P = 400 psia			P = 800 psia			P = 1200 psia		
[F]	Aspen	REFPROP	Diff.	Aspen	REFPROP	Diff.	Aspen	REFPROP	Diff.
100	211.5	211.5	0.01%	196.4	196.1	0.13%	161.7	159.6	1.26%
125	217.7	217.9	-0.07%	205.1	205.3	-0.12%	187.1	187.2	-0.05%
150	223.9	224.1	-0.11%	213.0	213.4	-0.22%	199.4	200.0	-0.29%
175	230.0	230.3	-0.12%	220.4	220.9	-0.26%	209.2	210.0	-0.37%
200	236.1	236.4	-0.12%	227.5	228.1	-0.27%	218.0	218.8	-0.38%
225	242.2	242.5	-0.11%	234.4	235.0	-0.26%	226.1	226.9	-0.36%
250	248.3	248.6	-0.09%	241.3	241.8	-0.23%	233.8	234.6	-0.33%
275	254.5	254.6	-0.06%	248.0	248.5	-0.19%	241.3	242.0	-0.28%
300	260.6	260.7	-0.04%	254.7	255.1	-0.16%	248.6	249.2	-0.23%
325	266.8	266.8	-0.01%	261.3	261.6	-0.12%	255.8	256.3	-0.18%
350	273.0	273.0	0.01%	267.9	268.1	-0.08%	262.9	263.2	-0.13%
375	279.2	279.2	0.03%	274.5	274.7	-0.04%	269.9	270.1	-0.08%
400	285.5	285.4	0.05%	281.1	281.2	-0.01%	276.8	276.9	-0.04%
425	291.8	291.6	0.06%	287.7	287.7	0.02%	283.8	283.8	0.00%
450	298.1	297.9	0.08%	294.3	294.2	0.05%	290.6	290.5	0.04%
475	304.5	304.2	0.09%	300.9	300.7	0.07%	297.5	297.3	0.07%
500	310.9	310.6	0.10%	307.6	307.3	0.09%	304.4	304.1	0.11%

Enthalpy Values [Btu/lb]									
Temp.	P = 1400 psia			P = 1800 psia			P = 2200 psia		
[F]	Aspen	REFPROP	Diff.	Aspen	REFPROP	Diff.	Aspen	REFPROP	Diff.
100	137.1	131.5	4.31%	126.3	123.5	2.30%	121.5	120.1	1.13%
125	174.0	173.4	0.34%	149.5	145.8	2.59%	139.6	137.0	1.92%
150	191.2	191.7	-0.23%	173.0	172.0	0.63%	159.2	156.7	1.57%
175	203.0	203.8	-0.37%	189.7	189.9	-0.10%	177.2	176.1	0.60%
200	212.9	213.7	-0.40%	202.3	202.9	-0.30%	191.9	191.8	0.07%
225	221.7	222.6	-0.39%	212.9	213.6	-0.35%	204.2	204.4	-0.14%
250	230.0	230.9	-0.35%	222.4	223.1	-0.34%	214.9	215.3	-0.21%
275	237.9	238.7	-0.31%	231.2	231.9	-0.30%	224.6	225.1	-0.21%
300	245.6	246.2	-0.25%	239.6	240.2	-0.25%	233.8	234.2	-0.18%
325	253.1	253.6	-0.20%	247.6	248.1	-0.19%	242.4	242.7	-0.14%
350	260.4	260.7	-0.14%	255.5	255.8	-0.14%	250.7	250.9	-0.08%
375	267.6	267.8	-0.09%	263.1	263.3	-0.08%	258.8	258.9	-0.03%
400	274.7	274.8	-0.04%	270.6	270.7	-0.02%	266.7	266.6	0.03%
425	281.8	281.8	0.01%	278.0	277.9	0.03%	274.4	274.2	0.08%
450	288.8	288.7	0.05%	285.3	285.1	0.08%	282.0	281.6	0.13%
475	295.9	295.6	0.08%	292.6	292.3	0.12%	289.5	289.0	0.18%
500	302.8	302.5	0.12%	299.8	299.4	0.16%	297.0	296.3	0.22%
Temp.	P = 1600 psia			P = 2000 psia					
[F]	Aspen	REFPROP	Diff.	Aspen	REFPROP	Diff.			
100	130.3	126.3	3.16%	123.6	121.6	1.66%			
125	159.4	156.4	1.89%	143.6	140.3	2.33%			
150	182.2	182.1	0.03%	165.2	163.1	1.28%			
175	196.4	197.0	-0.30%	183.1	182.7	0.23%			
200	207.6	208.4	-0.38%	197.0	197.2	-0.14%			
225	217.3	218.2	-0.39%	208.4	209.0	-0.26%			
250	226.2	227.0	-0.36%	218.6	219.2	-0.29%			
275	234.6	235.3	-0.31%	227.9	228.5	-0.27%			
300	242.6	243.2	-0.26%	236.6	237.2	-0.22%			
325	250.3	250.8	-0.20%	245.0	245.4	-0.17%			
350	257.9	258.3	-0.15%	253.1	253.4	-0.12%			
375	265.3	265.6	-0.09%	260.9	261.1	-0.06%			
400	272.6	272.7	-0.03%	268.6	268.6	0.00%			
425	279.9	279.9	0.02%	276.2	276.0	0.06%			
450	287.1	286.9	0.06%	283.7	283.4	0.10%			
475	294.2	293.9	0.10%	291.1	290.6	0.15%			
500	301.3	300.9	0.14%	298.4	297.8	0.19%			

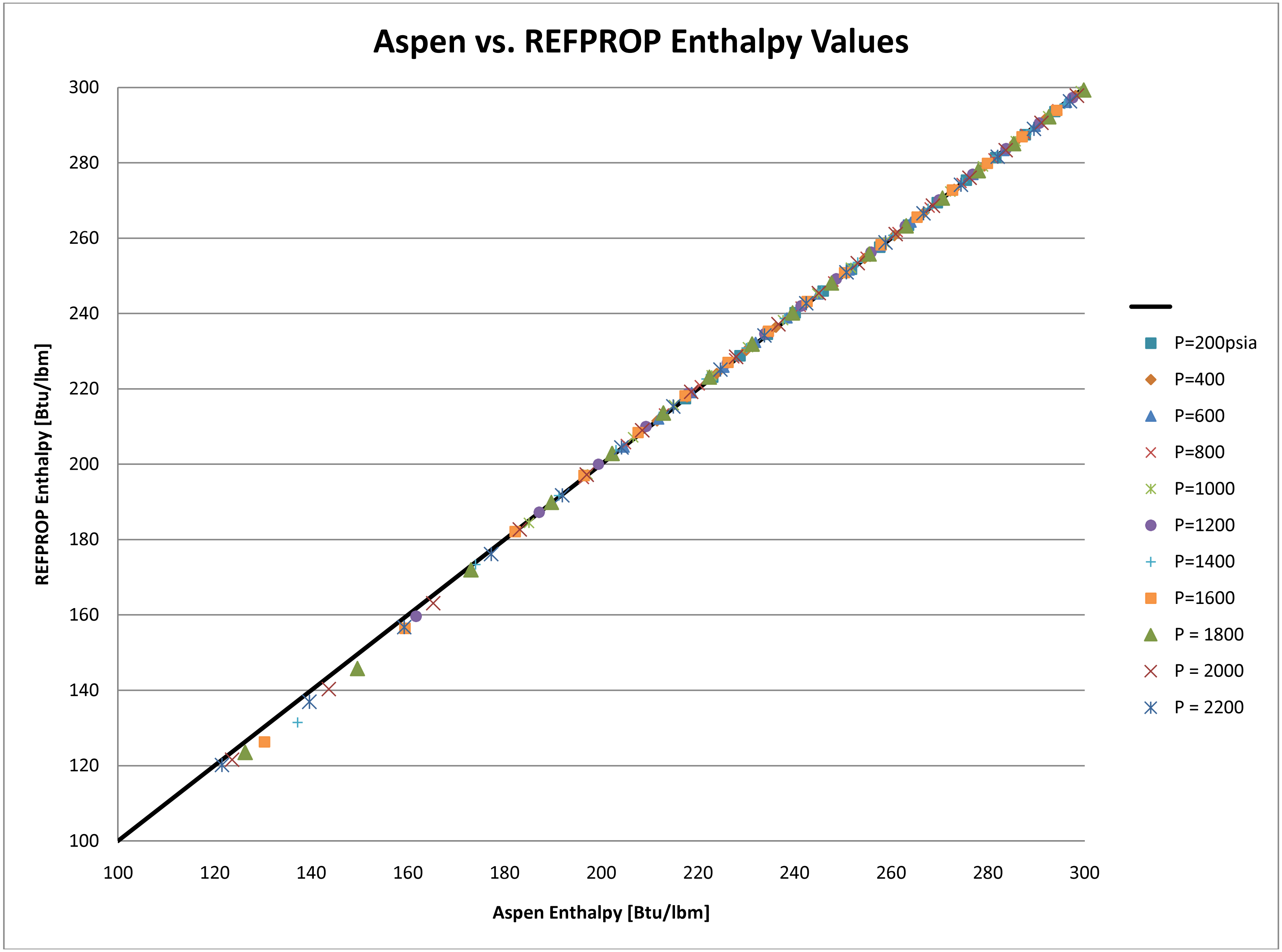


Figure 49. Pure CO_2 *Aspen Plus* versus REFPROP Enthalpy Values.

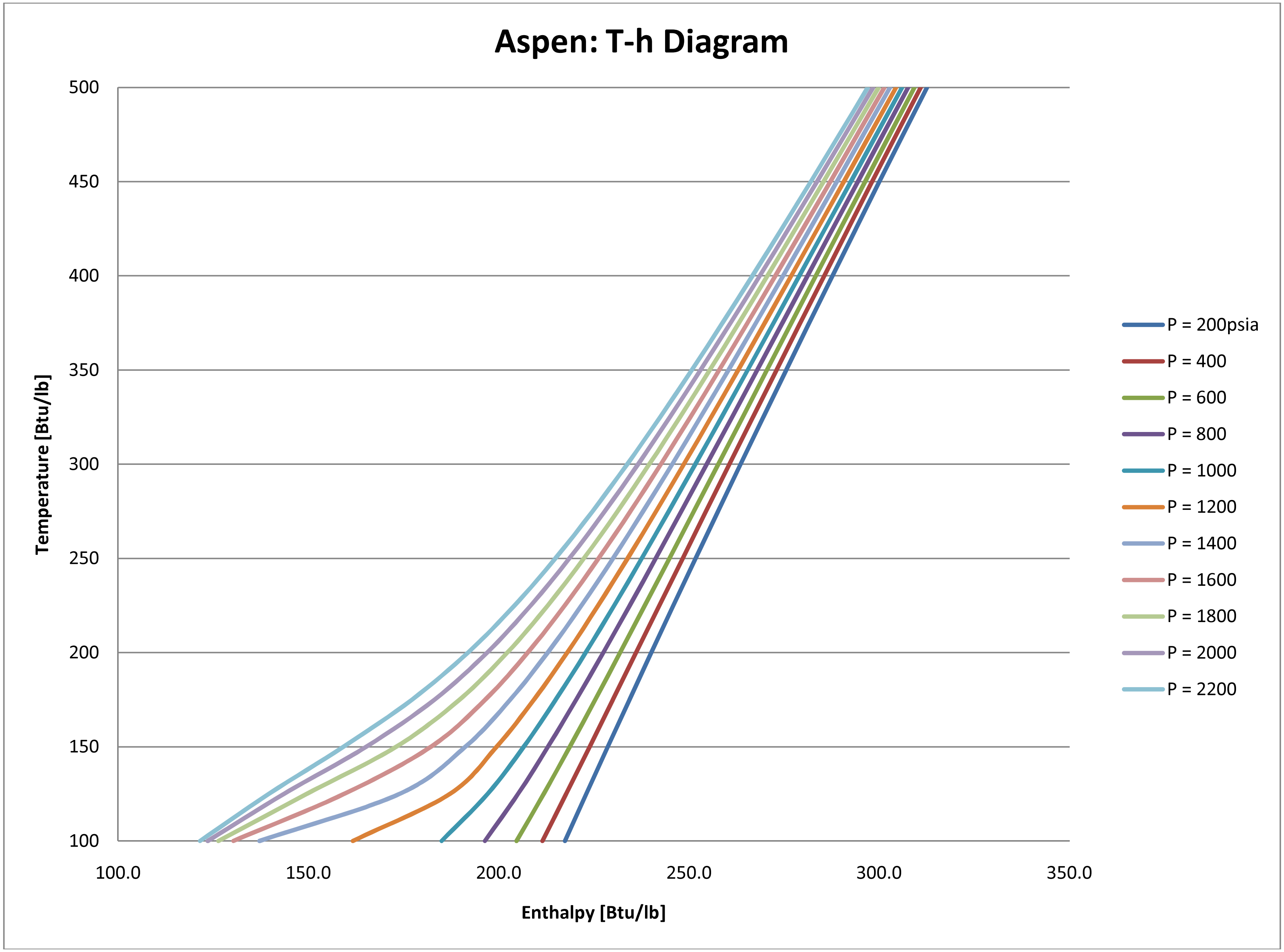


Figure 50. ***Aspen Plus*** **Enthalpy versus Temperature.**

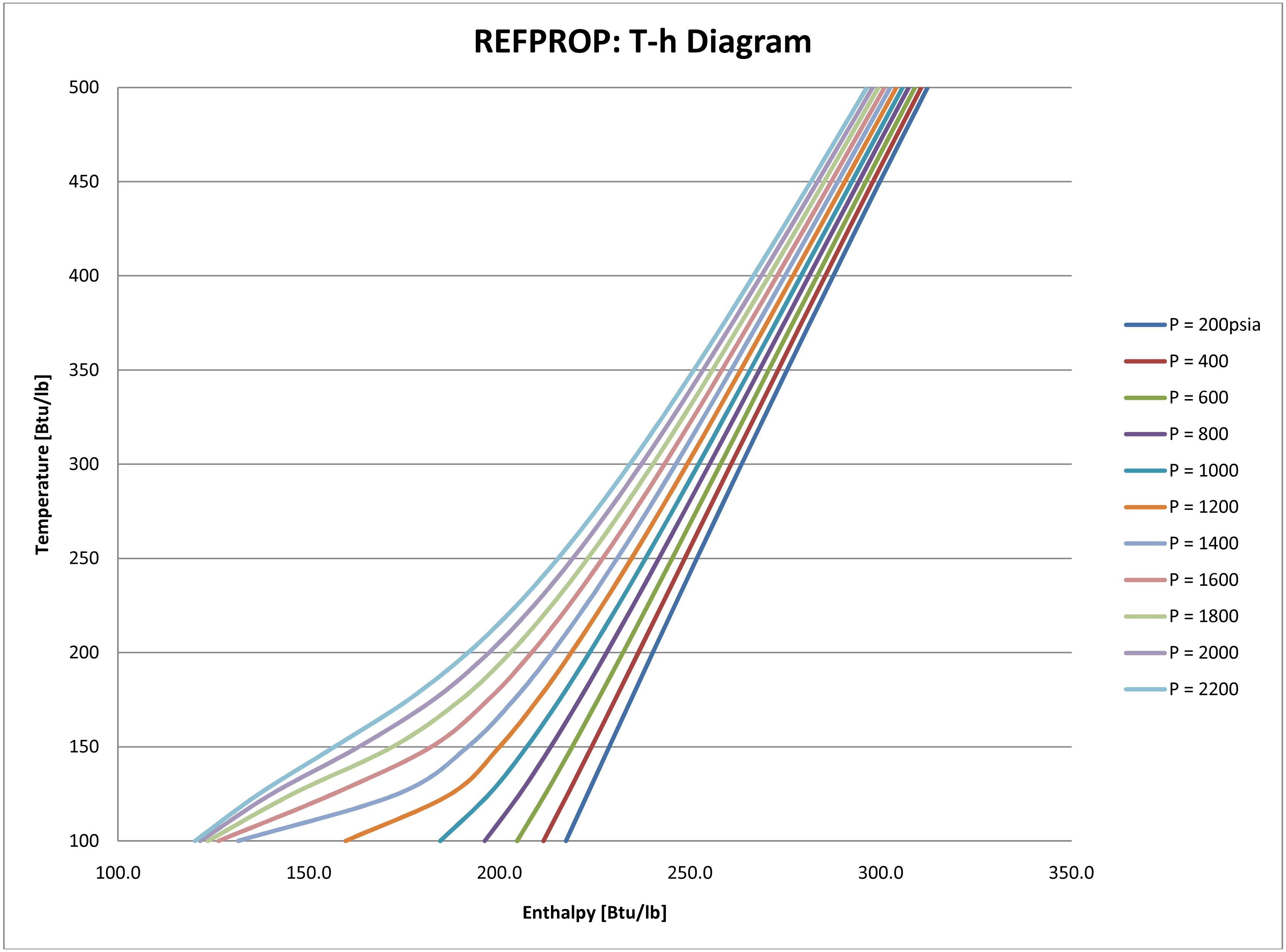


Figure 51. REFPROP Enthalpy versus Temperature.

Table 71. Pure CO_2 *Aspen Plus* and REFPROP Specific Heat Values.

	Specific Heat		
Temp.	[Btu/lb-R]		
[F]	Aspen	REFPROP	Diff.
100	0.2056	0.2048	0.39%
125	0.2093	0.2082	0.53%
150	0.2128	0.2114	0.64%
175	0.2160	0.2145	0.71%
200	0.2192	0.2175	0.75%
225	0.2221	0.2205	0.74%
250	0.2249	0.2233	0.72%
275	0.2276	0.2261	0.67%
300	0.2301	0.2288	0.60%
325	0.2326	0.2314	0.53%
350	0.2349	0.2339	0.44%
375	0.2371	0.2363	0.35%
400	0.2393	0.2387	0.26%
425	0.2414	0.2410	0.16%
450	0.2434	0.2433	0.07%
475	0.2454	0.2454	-0.02%
500	0.2473	0.2476	-0.10%

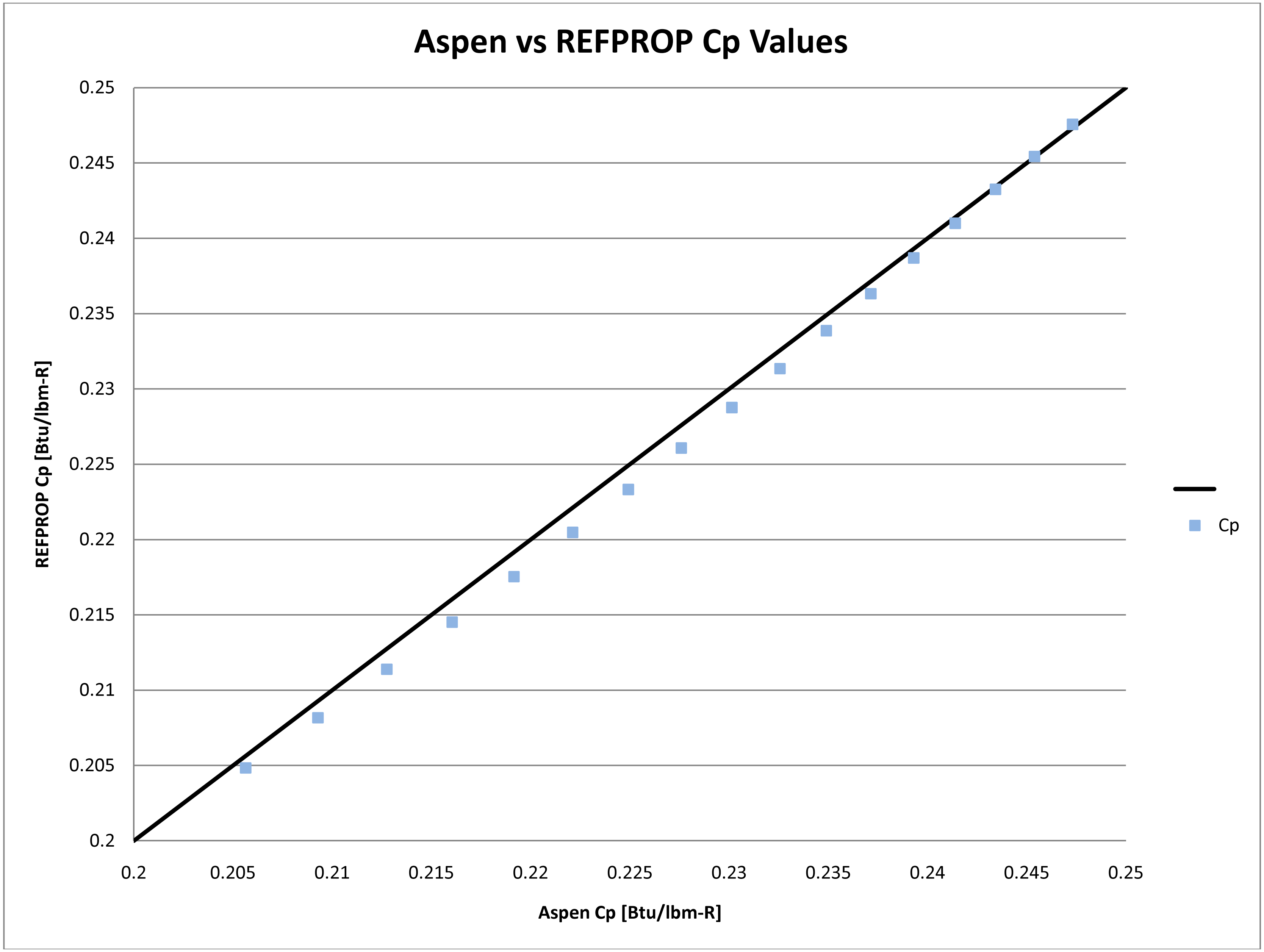


Figure 52. Pure CO_2 *Aspen Plus* versus REFPROP Specific Heat Values.

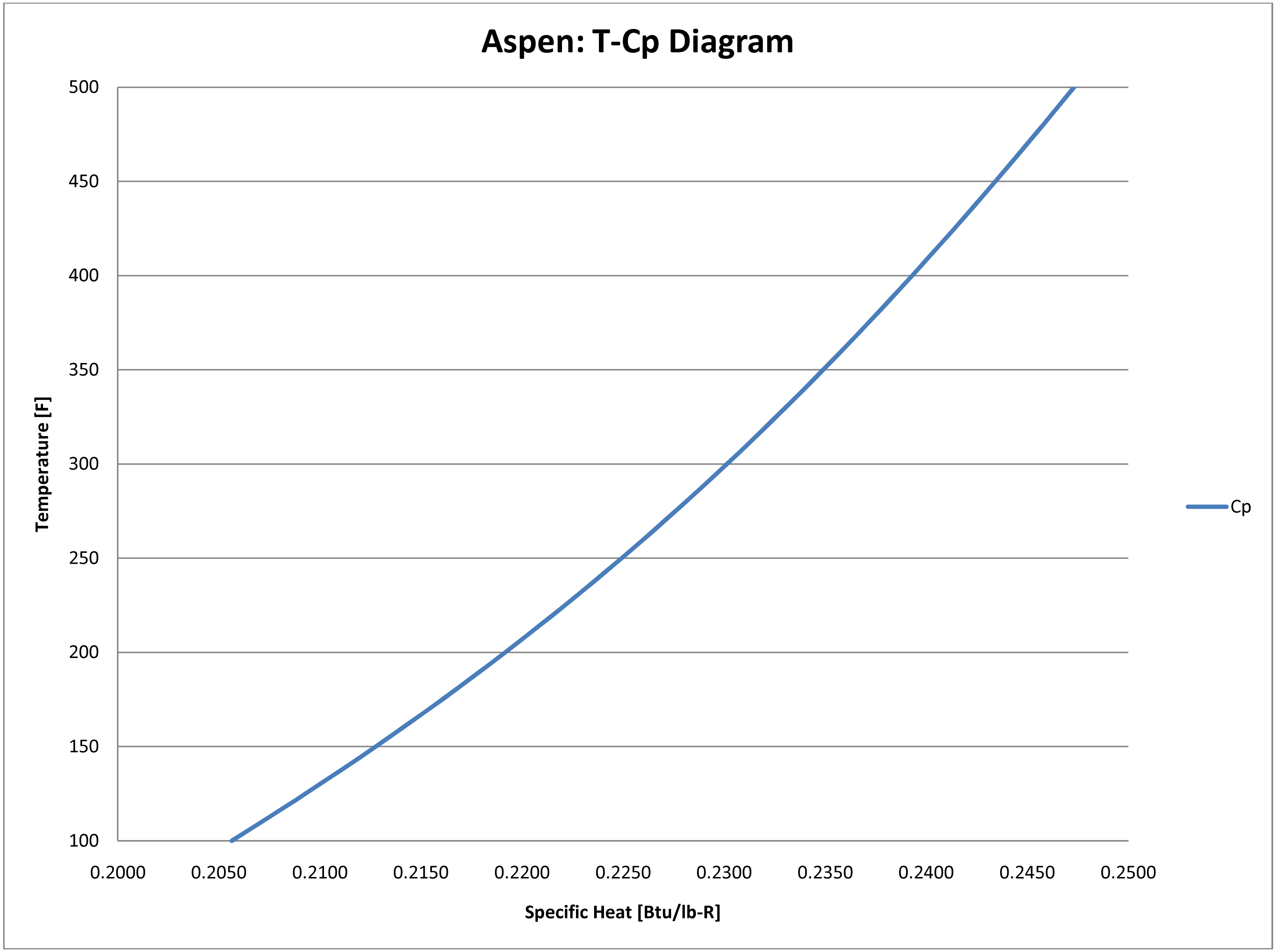


Figure 53. ***Aspen Plus*** **Specific Heat versus Temperature.**

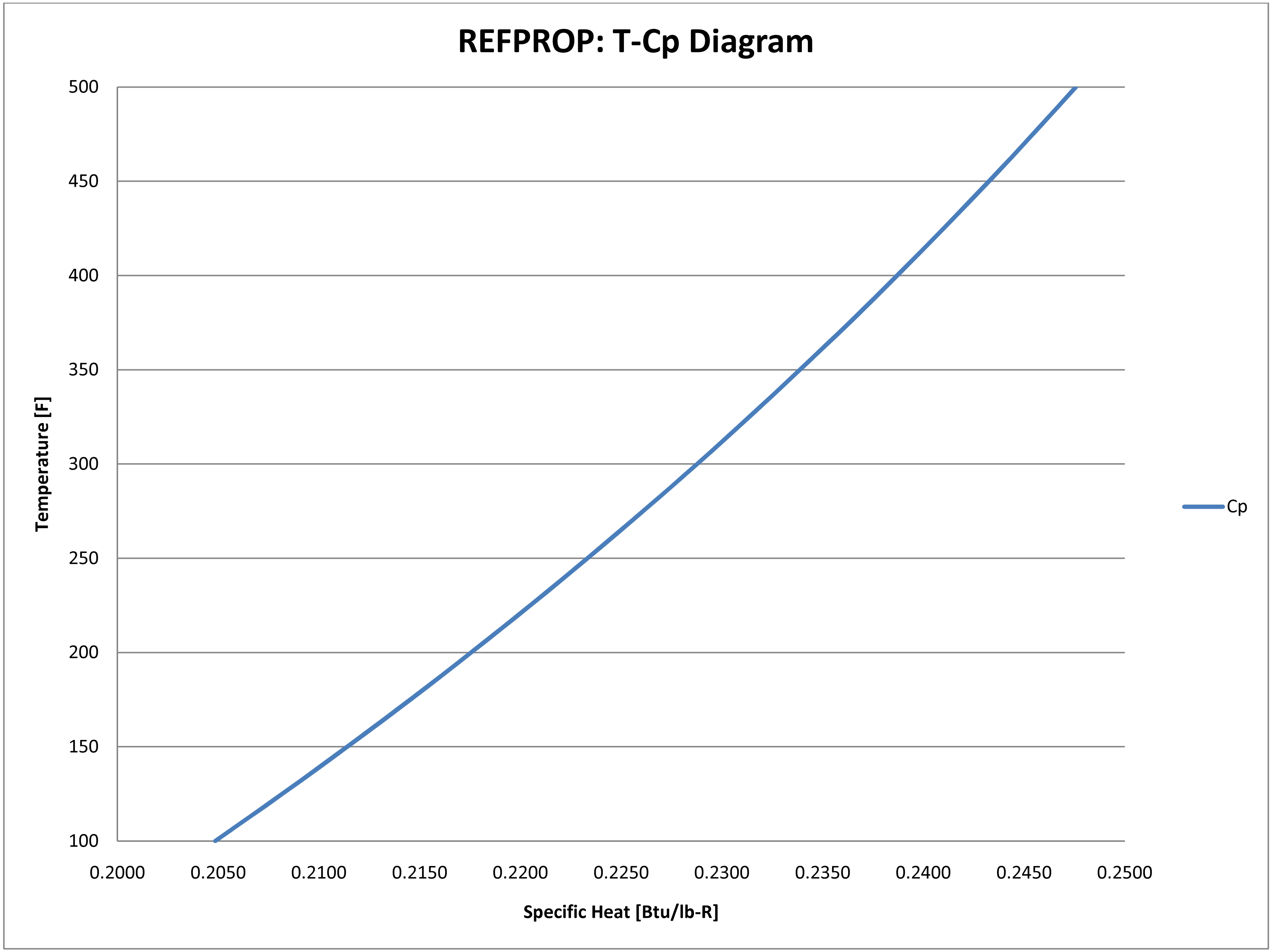


Figure 54. REFPROP Specific Heat versus Temperature.

Appendix E – REFPROP vs. Aspen Plus Compressor Data

Table 72. Ramgen Compressor and Intercooler Specifications and Results From Calculations Using REFPROP and *Aspen Plus* models.

Ramgen Compressor

Stage 1	Compressor		
	REFPROP	Aspen	Diff.
Mass Flow (lb/hr)	1,073,080		
Temp in (F)	100.0	100.0	
Pressure in (psia)	44.1	44.1	
Temp out (F)	**427.7**	**429.3**	
Pressure out (psia)	315.0	315.0	
Pressure Ratio	7.143	7.145	
Isentropic Eff.	0.85		
Mechanical Eff.	0.9704		
Mechanical Power (kW)	22,649	23,411	3.36%
Gas Power (kW)	21,979	22,718	3.36%
Stage 2	Compressor		
	REFPROP	Aspen	Diff.
Mass Flow (lb/hr)	1,067,360		
Temp in (F)	110.0	110.0	
Pressure in (psia)	310.0	310.0	
Temp out (F)	**458.1**	**463.0**	
Pressure out (psia)	2215.0	2215.0	
Pressure Ratio	7.145	7.145	
Isentropic Eff.	0.85		
Mechanical Eff.	0.9701		
Mechanical Power (kW)	21,043	22,183	5.42%
Gas Power (kW)	20,413	21,520	5.42%
TOTAL			
Total Mech. Power (kW)	43,692	45,594	4.35%
Total Gas Power (kW)	42,392	44,238	4.35%

Intercooler 1	Intercooling		
	REFPROP	Aspen	Diff.
CO_2 Temp in (F)	429.3	429.3	
CO_2 Temp out (F)	**110.0**	**110.0**	
H_2O Mass Flow (lb/hr)	263,885	263,885	
H_2O:CO_2 Mass Ratio	0.25	0.25	
H2O Pressure in (psia)	350.0	350.0	
H2O Temp in (F)	90.0	90.0	
H2O Temp out (F)	**414.3**	**419.5**	
Water Knockout (lb/hr)		5,720	
h-fg H2O @ P2,CO_2 (Btu/lb)	806.9		
Max Heat Avail. (Btu/hr)	87,838,488	89,013,445	1.34%
Post-Compressor Cooler 2	Intercooling		
	REFPROP	Aspen	Diff.
CO_2 Temp in (F)	463.0	463.0	
CO_2 Temp out (F)	**110.0**	**110.0**	
H_2O Mass Flow (lb/hr)	463,560	463,560	
H_2O:CO2 Mass Ratio	0.43	0.43	
H_2O Pressure in (psia)	500.0	500.0	
H_2O Temp in (F)	100.0	100.0	
H_2O Temp out (F)	**454.5**	**453.0**	
Max Heat Avail. (Btu/hr)	170,341,167	168,811,875	-0.90%

Table 73. Inline 4 Compressor and Intercooler Specifications and Results From Calculations Using REFPROP and *Aspen Plus* models.

Inline 4 Compressor

Stage 1	Compressor		
	REFPROP	Aspen	Diff.
Mass Flow (lb/hr)	1,073,080		
Temp in (F)	100.0	100.0	
Pressure in (psia)	44.1	44.1	
Temp out (F)	**424.3**	**425.9**	
Pressure out (psia)	289.3	289.3	
Pressure Ratio	6.560	6.563	
Isentropic Eff.	0.8125		
Mechanical Eff.	0.993		
Mechanical Power (kW)	22,464	22,668	-0.90%
Gas Power (kW)	22,306	22,510	-0.90%

Intercooler 1	Intercooling		
	REFPROP	Aspen	Diff.
CO_2 Temp in (F)	426.0	425.9	
CO_2 Temp out (F)	**110.0**	**110.0**	
H_2O Mass Flow (lb/hr)	262,057	262,351	
H_2O:CO_2 Mass Ratio	0.225	0.24	
H_2O Pressure in (psia)	350.0	350.0	
H_2O Temp in (F)	90.0	90.0	
H_2O Temp out (F)	**411.0**	**415.9**	
Water Knockout [lb/hr]		5,602	
h-fg H_2O @ P2,CO_2 (Btu/lb)	815		
Max Heat Avail. (Btu/hr)	86,280,542	87,469,588	-1.36%

Stage 2	Compressor		
	REFPROP	Aspen	Diff.
Mass Flow (lb/hr)	1,067,480		
Temp in (F)	110.0	110.0	
Pressure in (psia)	284.3	284.3	
Temp out (F)	**431.2**	**436.1**	
Pressure out (psia)	1720.3	1720.3	
Pressure Ratio	6.051	6.050	
Isentropic Eff.	0.8188		
Mechanical Eff.	0.992		
Mechanical Power (kW)	19,753	20,249	-2.45%
Gas Power (kW)	19,595	20,087	-2.45%

Intercooler 2	Intercooling		
	REFPROP	Aspen	Diff.
CO_2 Temp in (F)	436.0	436.1	
CO_2 Temp out (F)	**110.0**	**110.0**	
H_2O Mass Flow (lb/hr)	451,779	452,286	
H_2O:CO2 Mass Ratio	0.431	0.42	
H_2O Pressure in (psia)	350.0	350.0	
H_2O Temp in (F)	90.0	90.0	
H_2O Temp out (F)	**427.4**	**426.1**	
Max Heat Avail. (Btu/hr)	159,593,365	155,846,773	2.40%

Stage 3	Compressor		
	REFPROP	Aspen	Diff.
Mass Flow (lb/hr)	1,067,480		
Temp in (F)	110.0	110.0	
Pressure in (psia)	1715.3	1715.3	
Temp out (F)	**123.5**	**125.9**	
Pressure out (psia)	2219.6	2219.6	
Pressure Ratio	1.294	1.294	
Isentropic Eff.	0.8114		
Mechanical Eff.	0.998		
Mechanical Power (kW)	842	987	-14.72%
Gas Power (kW)	840	985	-14.72%
TOTAL			
Total Mech. Power (kW)	43,058	43,905	-1.93%
Total Gas Power (kW)	42,741	43,582	-1.93%

Table 74. Integrally Geared 1 with 149F Intercooling Compressor and Intercooler Specifications and Results From Calculations Using REFPROP and *Aspen Plus* models.

IG 1 Compressor with 149F Intercooling

Stage 1	Compressor		
	REFPROP	Aspen	Diff.
Mass Flow (lb/hr)	1,073,080		
Temp in (F)	100.0	100.0	
Pressure in (psia)	44.1	44.1	
Temp out (F)	**160.9**	**161.1**	
Pressure out (psia)	66.3	66.3	
Pressure Ratio	1.503	1.503	
Polytropic Eff.	0.8597		
Isentropic Eff.	0.854		
Mechanical Eff.	0.97		
Mechanical Power (kW)	3,956	4,074	-2.89%
Gas Power (kW)	3,838	3,952	-2.89%

Intercooler 1	Intercooling		
	REFPROP	Aspen	Diff.
CO_2 Temp in (F)	161.1	161.1	
CO_2 Temp out (F)	**149.0**	**149.0**	
H_2O Mass Flow (lb/hr)	43,540	43,540	
H_2O:CO_2 Mass Ratio	0.04	0.04	
H_2O Pressure in (psia)	85.0	85.0	
H_2O Temp in (F)	90.0	90.0	
H_2O Temp out (F)	**151.2**	**151.1**	
Max Heat Avail. (Btu/hr)	2,672,035	2,656,018	0.60%

Stage 2	Compressor		
	REFPROP	Aspen	Diff.
Mass Flow (lb/hr)	1,073,080		
Temp in (F)	149.0	149.0	
Pressure in (psia)	59.0	59.0	
Temp out (F)	**277.5**	**279.3**	
Pressure out (psia)	131.5	131.5	
Pressure Ratio	2.229	2.228	
Polytropic Eff.	0.8649		
Isentropic Eff.	0.862		
Mechanical Eff.	0.97		
Mechanical Power (kW)	8,714	9,073	-3.95%
Gas Power (kW)	8,453	8,801	-3.95%

Intercooler 2	Intercooling		
	REFPROP	Aspen	Diff.
CO_2 Temp in (F)	279.3	279.3	
CO_2 Temp out (F)	**149.0**	**149.0**	
H_2O Mass Flow (lb/hr)	175,714	175,714	
H_2O:CO2 Mass Ratio	0.16	0.16	
H_2O Pressure in (psia)	85.0	85.0	
H_2O Temp in (F)	90.0	90.0	
H_2O Temp out (F)	**269.5**	**269.2**	
Water Knockout (lb/hr)		0.0	
h-fg H_2O @ P2,CO_2 (Btu/lb)	876.7		
Max Heat Avail. (Btu/hr)	31,737,217	31,633,848	0.33%

Stage 3	Compressor		
	REFPROP	Aspen	Diff.
Mass Flow (lb/hr)	1,073,080		
Temp in (F)	149.0	149.0	
Pressure in (psia)	124.2	124.2	
Temp out (F)	**277.7**	**280.3**	
Pressure out (psia)	278.1	278.1	
Pressure Ratio	2.239	2.239	
Polytropic Eff.	0.8754		
Isentropic Eff.	0.876		
Mechanical Eff.	0.97		
Mechanical Power (kW)	8,494	8,884	-4.38%
Gas Power (kW)	8,239	8,617	-4.38%

Intercooler 3	Intercooling		
	REFPROP	Aspen	Diff.
CO_2 Temp in (F)	280.3	280.3	
CO_2 Temp out (F)	**149.0**	**149.0**	
H_2O Mass Flow (lb/hr)	192,692	192,692	
H_2O:CO_2 Mass Ratio	0.18	0.18	
H_2O Pressure in (psia)	85.0	85.0	
H_2O Temp in (F)	90.0	90.0	
H_2O Temp out (F)	**268.9**	**270.3**	
Water Knockout (lb/hr)		**1670.0**	
h-fg H_2O @ P2,CO_2 (Btu/lb)	819.1		
Max Heat Avail. (Btu/hr)	34,691,126	34,898,117	-0.59%

Stage 4	Compressor		
	REFPROP	Aspen	Diff.
Mass Flow (lb/hr)	1,071,410		
Temp in (F)	149.0	149.0	
Pressure in (psia)	270.8	270.8	
Temp out (F)	**275.1**	278.5	
Pressure out (psia)	572.0	572.0	
Pressure Ratio	2.112	2.112	
Polytropic Eff.	0.8343		
Isentropic Eff.	0.832		
Mechanical Eff.	0.97		
Mechanical Power (kW)	7,913	8,309	-4.77%
Gas Power (kW)	7,675	8,060	-4.77%

Intercooler 4	Intercooling		
	REFPROP	Aspen	Diff.
CO_2 Temp in (F)	278.5	278.5	
CO_2 Temp out (F)	**149.0**	**149.0**	
H_2O Mass Flow (lb/hr)	212,394	212,394	
H_2O:CO_2 Mass Ratio	0.20	0.20	
H_2O Pressure in (psia)	85.0	85.0	
H_2O Temp in (F)	90.0	90.0	
H_2O Temp out (F)	**266.6**	**268.5**	
Water Knockout (lb/hr)		**2420.0**	
h-fg H_2O @ P2,CO_2 (Btu/lb)	740.6		
Max Heat Avail. (Btu/hr)	37,734,393	38,079,677	-0.91%

Stage 5	Compressor		
	REFPROP	Aspen	Diff.
Mass Flow (lb/hr)	1,068,990		
Temp in (F)	149.0	149.0	
Pressure in (psia)	564.8	564.8	
Temp out (F)	**233.0**	236.1	
Pressure out (psia)	949.4	949.4	
Pressure Ratio	1.681	1.681	
Polytropic Eff.	0.8846		
Isentropic Eff.	0.890		
Mechanical Eff.	0.97		
Mechanical Power (kW)	4,602	4,856	-5.23%
Gas Power (kW)	4,464	4,710	-5.23%

Intercooler 5	Intercooling		
	REFPROP	Aspen	Diff.
CO_2 Temp in (F)	236.1	236.1	
CO_2 Temp out (F)	**149.0**	**149.0**	
H_2O Mass Flow (lb/hr)	208,137	208,137	
H_2O:CO2 Mass Ratio	0.19	0.19	
H_2O Pressure in (psia)	85.0	85.0	
H_2O Temp in (F)	90.0	90.0	
H_2O Temp out (F)	**226.7**	**226.1**	
Water Knockout (lb/hr)		**640.0**	
h-fg H_2O @ P2,CO_2 (Btu/lb)	664.4		
Max Heat Avail. (Btu/hr)	28,563,733	28,348,792	0.76%

Stage 6	Compressor		
	REFPROP	Aspen	Diff.
Mass Flow (lb/hr)	1,068,350		
Temp in (F)	149.0	149.0	
Pressure in (psia)	927.6	927.6	
Temp out (F)	**219.6**	223.0	
Pressure out (psia)	1439.6	1439.6	
Pressure Ratio	1.552	1.552	
Polytropic Eff.	0.8781		
Isentropic Eff.	0.907		
Mechanical Eff.	0.97		
Mechanical Power (kW)	3,347	3,671	-8.82%
Gas Power (kW)	3,247	3,561	-8.82%

Stage 7	Compressor		
	REFPROP	Aspen	Diff.
Mass Flow (lb/hr)	1,068,350		
Temp in (F)	223.0	223.0	
Pressure in (psia)	1439.1	1439.1	
Temp out (F)	**297.5**	300.7	
Pressure out (psia)	2219.2	2219.2	
Pressure Ratio	1.542	1.542	
Polytropic Eff.	0.8585		
Isentropic Eff.	0.917		
Mechanical Eff.	0.97		
Mechanical Power (kW)	3,742	4,315	-13.29%
Gas Power (kW)	3,629	4,186	-13.29%
TOTAL			
Total Mech. Power (kW)	40,768	43,182	-5.59%
Total Gas Power (kW)	39,545	41,886	-5.59%

Post-Compressor Cooler 7	Intercooling		
	REFPROP	Aspen	Diff.
CO_2 Temp in (F)	300.7	300.7	
CO_2 Temp out (F)	**110.0**	**110.0**	
H_2O Mass Flow (lb/hr)	563,849	563,849	
H_2O:CO2 Mass Ratio	0.53	0.53	
H_2O Pressure in (psia)	85.0	85.0	
H_2O Temp in (F)	90.0	90.0	
H_2O Temp out (F)	**294.0**	**290.7**	
Max Heat Avail. (Btu/hr)	115,955,584	113,913,503	1.79%

Table 75. Integrally Geared 1 with 110F Intercooling Compressor and Intercooler Specifications and Results From Calculations Using REFPROP and *Aspen Plus* models.

IG 1 Compressor with 110F Intercooling

Stage 1	Compressor		
	REFPROP	Aspen	Diff.
Mass Flow (lb/hr)	1,073,080		
Temp in (F)	100.0	100.0	
Pressure in (psia)	44.1	44.1	
Temp out (F)	**160.9**	**161.1**	
Pressure out (psia)	66.3	66.3	
Pressure Ratio	1.503	1.503	
Polytropic Eff.	0.8597		
Isentropic Eff.	0.854		
Mechanical Eff.	0.97		
Mechanical Power (kW)	3,956	4,074	-2.89%
Gas Power (kW)	3,838	3,952	-2.89%

Intercooler 1	Intercooling		
	REFPROP	Aspen	Diff.
CO_2 Temp in (F)	161.1	161.1	
CO_2 Temp out (F)	**110.0**	**110.0**	
H_2O Mass Flow (lb/hr)	191,305	191,305	
H_2O:CO_2 Mass Ratio	0.18	0.18	
H_2O Pressure in (psia)	85.0	85.0	
H_2O Temp in (F)	90.0	90.0	
H_2O Temp out (F)	**150.9**	**150.9**	
Max Heat Avail. (Btu/hr)	11,686,878	11,632,909	0.46%

Stage 2	Compressor		
	REFPROP	Aspen	Diff.
Mass Flow (lb/hr)	1,073,080		
Temp in (F)	110.0	110.0	
Pressure in (psia)	59.0	59.0	
Temp out (F)	**233.4**	**235.2**	
Pressure out (psia)	131.5	131.5	
Pressure Ratio	2.229	2.228	
Polytropic Eff.	0.8649		
Isentropic Eff.	0.862		
Mechanical Eff.	0.97		
Mechanical Power (kW)	8,145	8,479	-3.94%
Gas Power (kW)	7,901	8,225	-3.94%

Intercooler 2	Intercooling		
	REFPROP	Aspen	Diff.
CO_2 Temp in (F)	235.2	235.2	
CO_2 Temp out (F)	**110.0**	**110.0**	
H_2O Mass Flow (lb/hr)	250,306	250,306	
H_2O:CO_2 Mass Ratio	0.23	0.23	
H_2O Pressure in (psia)	85.0	85.0	
H_2O Temp in (F)	90.0	90.0	
H_2O Temp out (F)	**222.4**	**225.1**	
Water Knockout (lb/hr)		3650.0	
h-fg H_2O @ P2,CO_2 (Btu/lb)	876.7		
Max Heat Avail. (Btu/hr)	33,260,229	33,861,575	-1.78%

Stage 3	Compressor		
	REFPROP	Aspen	Diff.
Mass Flow (lb/hr)	1,069,430		
Temp in (F)	110.0	110.0	
Pressure in (psia)	124.2	124.2	
Temp out (F)	**233.4**	**236.3**	
Pressure out (psia)	278.1	278.1	
Pressure Ratio	2.239	2.239	
Polytropic Eff.	0.8754		
Isentropic Eff.	0.876		
Mechanical Eff.	0.97		
Mechanical Power (kW)	7,839	8,201	-4.42%
Gas Power (kW)	7,603	7,955	-4.42%

Stage 4	Compressor		
	REFPROP	Aspen	Diff.
Mass Flow (lb/hr)	1,067,560		
Temp in (F)	110.0	110.0	
Pressure in (psia)	270.8	270.8	
Temp out (F)	**230.8**	234.8	
Pressure out (psia)	572.0	572.0	
Pressure Ratio	2.112	2.112	
Polytropic Eff.	0.8343		
Isentropic Eff.	0.832		
Mechanical Eff.	0.97		
Mechanical Power (kW)	7,221	7,588	-4.84%
Gas Power (kW)	7,004	7,360	-4.84%

Intercooler 3	Intercooling		
	REFPROP	Aspen	Diff.
CO_2 Temp in (F)	236.3	236.3	
CO_2 Temp out (F)	**110.0**	**110.0**	
H_2O Mass Flow (lb/hr)	244,880	244,880	
H_2O:CO_2 Mass Ratio	0.23	0.23	
H_2O Pressure in (psia)	85.0	85.0	
H_2O Temp in (F)	90.0	90.0	
H_2O Temp out (F)	**225.3**	**226.2**	
Water Knockout (lb/hr)		**1870.0**	
h-fg H_2O @ P2,CO_2 (Btu/lb)	819.1		
Max Heat Avail. (Btu/hr)	33,245,054	33,382,485	-0.41%

Intercooler 4	Intercooling		
	REFPROP	Aspen	Diff.
CO_2 Temp in (F)	234.8	234.8	
CO_2 Temp out (F)	**110.0**	**110.0**	
H_2O Mass Flow (lb/hr)	263,152	263,152	
H_2O:CO_2 Mass Ratio	0.25	0.25	
H_2O Pressure in (psia)	85.0	85.0	
H_2O Temp in (F)	90.0	90.0	
H_2O Temp out (F)	**226.1**	**224.7**	
Water Knockout (lb/hr)		**750.0**	
h-fg H_2O @ P2,CO_2 (Btu/lb)	740.6		
Max Heat Avail. (Btu/hr)	35,935,925	35,495,387	1.24%

Stage 5	Compressor		
	REFPROP	Aspen	Diff.
Mass Flow (lb/hr)	1,066,810		
Temp in (F)	110.0	110.0	
Pressure in (psia)	564.8	564.8	
Temp out (F)	**190.5**	194.3	
Pressure out (psia)	949.4	949.4	
Pressure Ratio	1.681	1.681	
Polytropic Eff.	0.8846		
Isentropic Eff.	0.890		
Mechanical Eff.	0.97		
Mechanical Power (kW)	4,083	4,317	-5.41%
Gas Power (kW)	3,961	4,187	-5.41%

Intercooler 5	Intercooling		
	REFPROP	Aspen	Diff.
CO_2 Temp in (F)	194.3	194.3	
CO_2 Temp out (F)	**110.0**	**110.0**	
H_2O Mass Flow (lb/hr)	319,510	319,510	
H_2O:CO_2 Mass Ratio	0.30	0.30	
H_2O Pressure in (psia)	85.0	85.0	
H_2O Temp in (F)	90.0	90.0	
H_2O Temp out (F)	**187.1**	**184.3**	
Water Knockout (lb/hr)		**120.0**	
h-fg H_2O @ P2,CO_2 (Btu/lb)	664.4		
Max Heat Avail. (Btu/hr)	31,106,383	30,093,402	3.37%

Stage 6	Compressor		
	REFPROP	Aspen	Diff.
Mass Flow (lb/hr)	1,066,690		
Temp in (F)	110.0	110.0	
Pressure in (psia)	927.6	927.6	
Temp out (F)	**176.0**	179.7	
Pressure out (psia)	1439.6	1439.6	
Pressure Ratio	1.552	1.552	
Polytropic Eff.	0.8781		
Isentropic Eff.	0.907		
Mechanical Eff.	0.97		
Mechanical Power (kW)	2,757	3,044	-9.42%
Gas Power (kW)	2,675	2,953	-9.42%

Intercooler 6	Intercooling		
	REFPROP	Aspen	Diff.
CO_2 Temp in (F)	179.7	179.7	
CO_2 Temp out (F)	**110.0**	**110.0**	
H_2O Mass Flow (lb/hr)	733,837	733,837	
H_2O:CO_2 Mass Ratio	0.69	0.69	
H_2O Pressure in (psia)	85.0	85.0	
H_2O Temp in (F)	90.0	90.0	
H_2O Temp out (F)	**177.4**	**169.7**	
Max Heat Avail. (Btu/hr)	64,104,944	58,420,451	9.73%

Stage 7	Compressor		
	REFPROP	Aspen	Diff.
Mass Flow (lb/hr)	1,066,690		
Temp in (F)	110.0	110.0	
Pressure in (psia)	1439.1	1439.1	
Temp out (F)	**141.1**	145.2	
Pressure out (psia)	2219.2	2219.1	
Pressure Ratio	1.542	1.542	
Polytropic Eff.	0.8585		
Isentropic Eff.	0.917		
Mechanical Eff.	0.97		
Mechanical Power (kW)	1,363	1,793	-23.96%
Gas Power (kW)	1,322	1,739	-23.96%
TOTAL			
Total Mech. Power (kW)	35,364	37,496	-5.68%
Total Gas Power (kW)	34,303	36,371	-5.68%

Post-cooler 7	Intercooling		
	REFPROP	Aspen	Diff.
CO_2 Temp in (F)	145.2	145.2	
CO_2 Temp out (F)	**110.0**	**110.0**	
H_2O Mass Flow (lb/hr)	631,028	631,028	
$H_2O:CO_2$ Mass Ratio	0.59	0.59	
H_2O Pressure in (psia)	85.0	85.0	
H_2O Temp in (F)	90.0	90.0	
H_2O Temp out (F)	**133.6**	**135.2**	
Max Heat Avail. (Btu/hr)	27,634,282	28,473,213	-2.95%

Appendix F – Thermal Integration Case Results

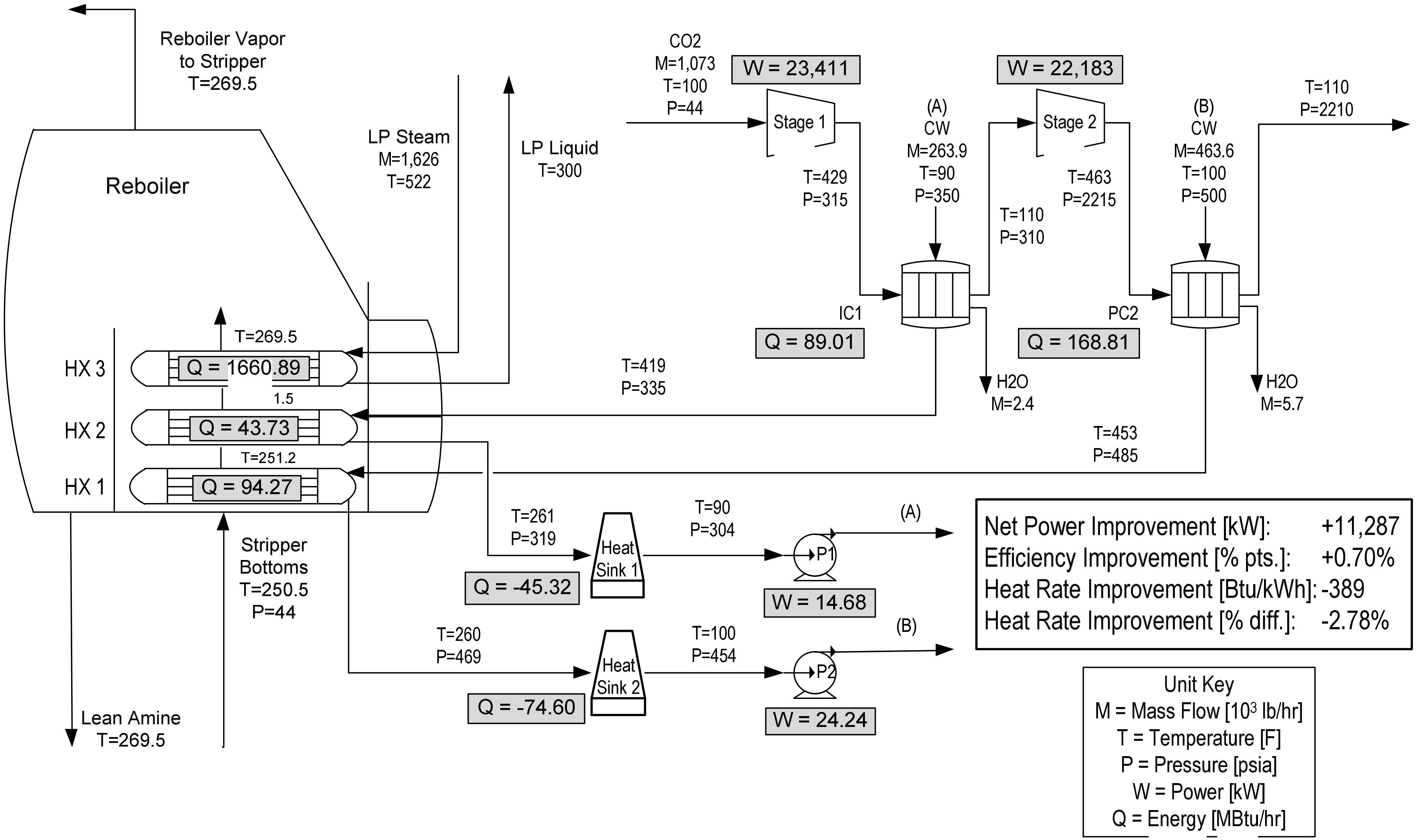


Figure 55. Ramgen IC1 and PC2 to Stripper Reboiler Thermal Integration Results.

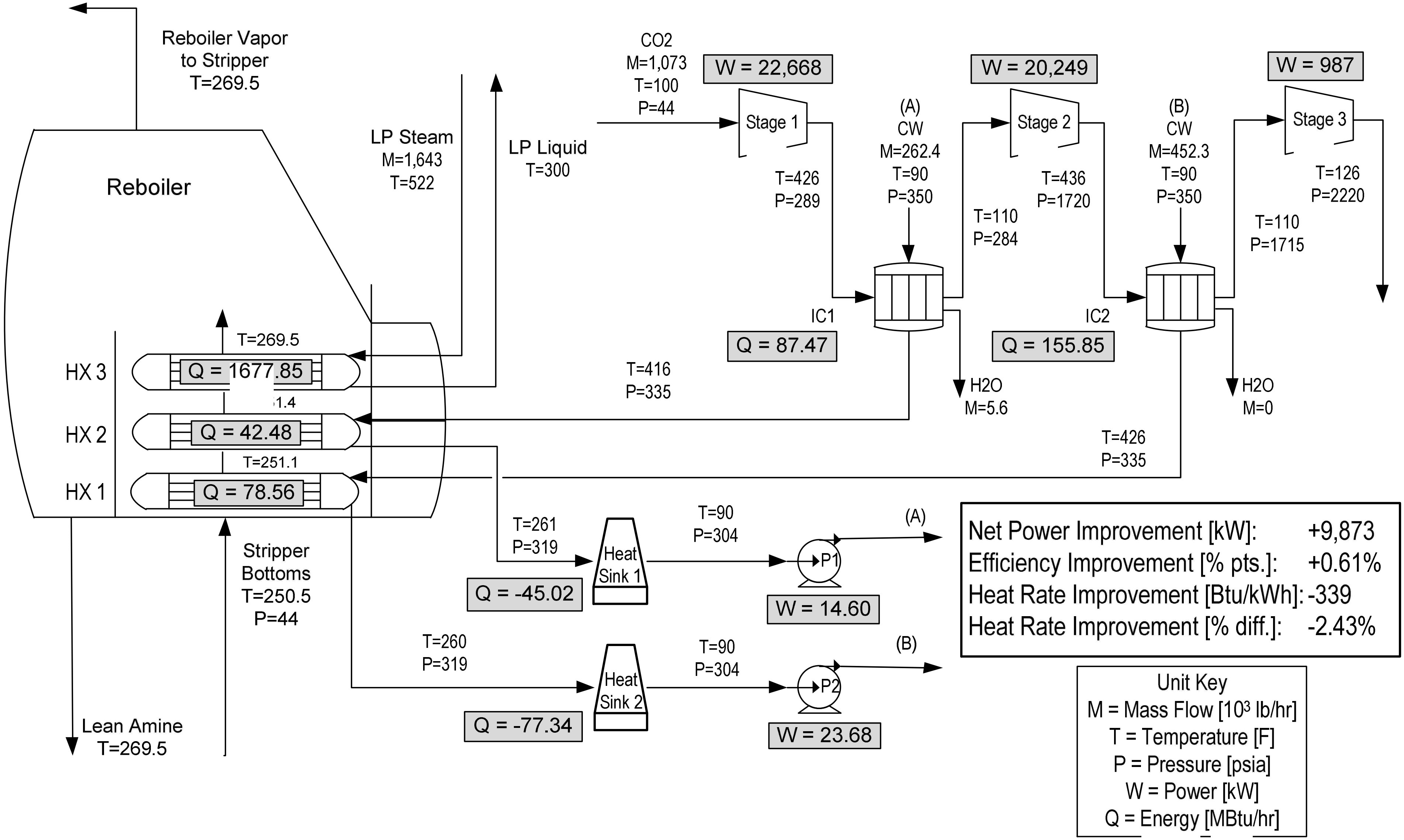


Figure 56. Inline 4 IC1 and IC2 to Stripper Reboiler Thermal Integration Results.

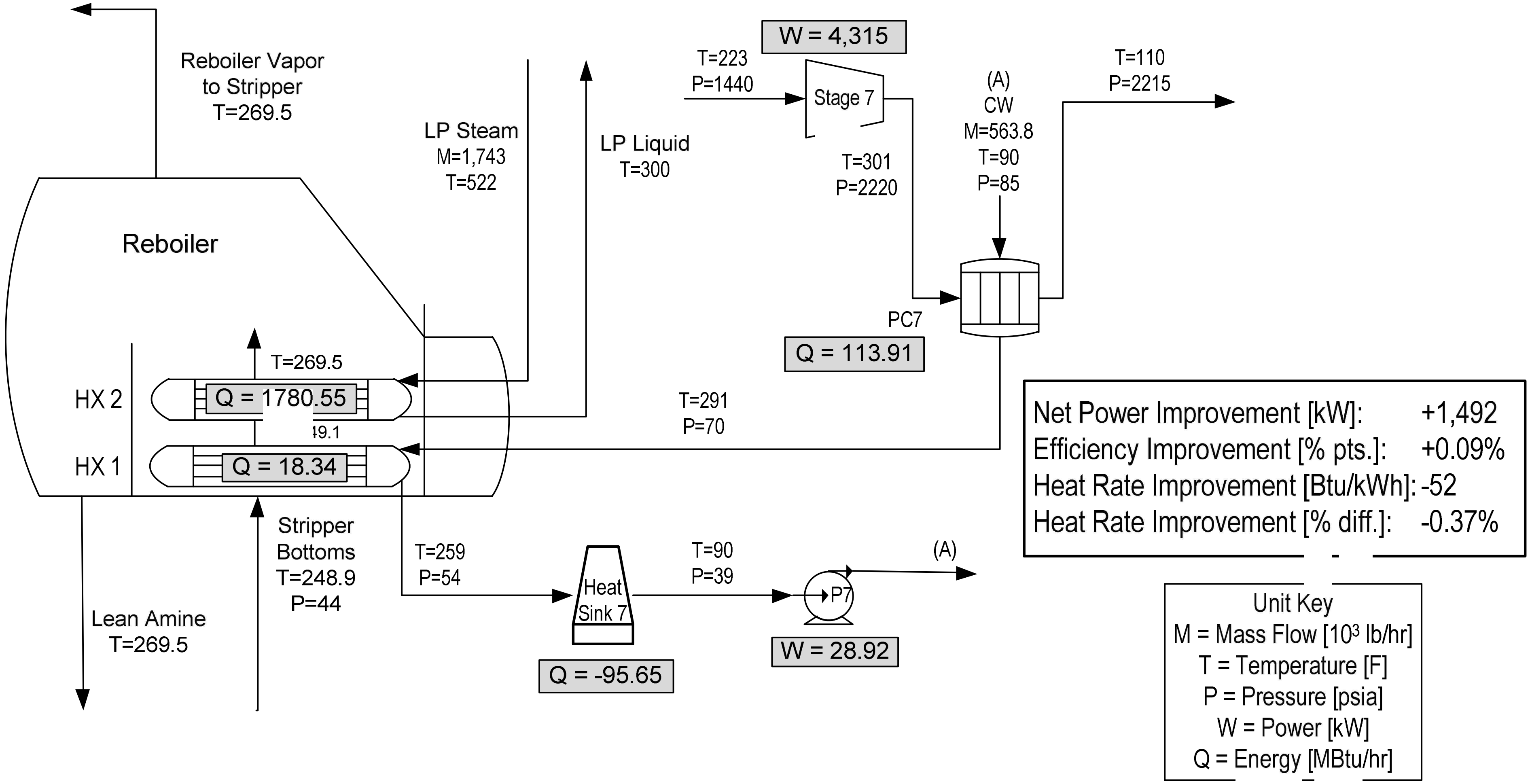


Figure 57. Integrally Geared 1 – 149 PC7 to Stripper Reboiler Thermal Integration Results.

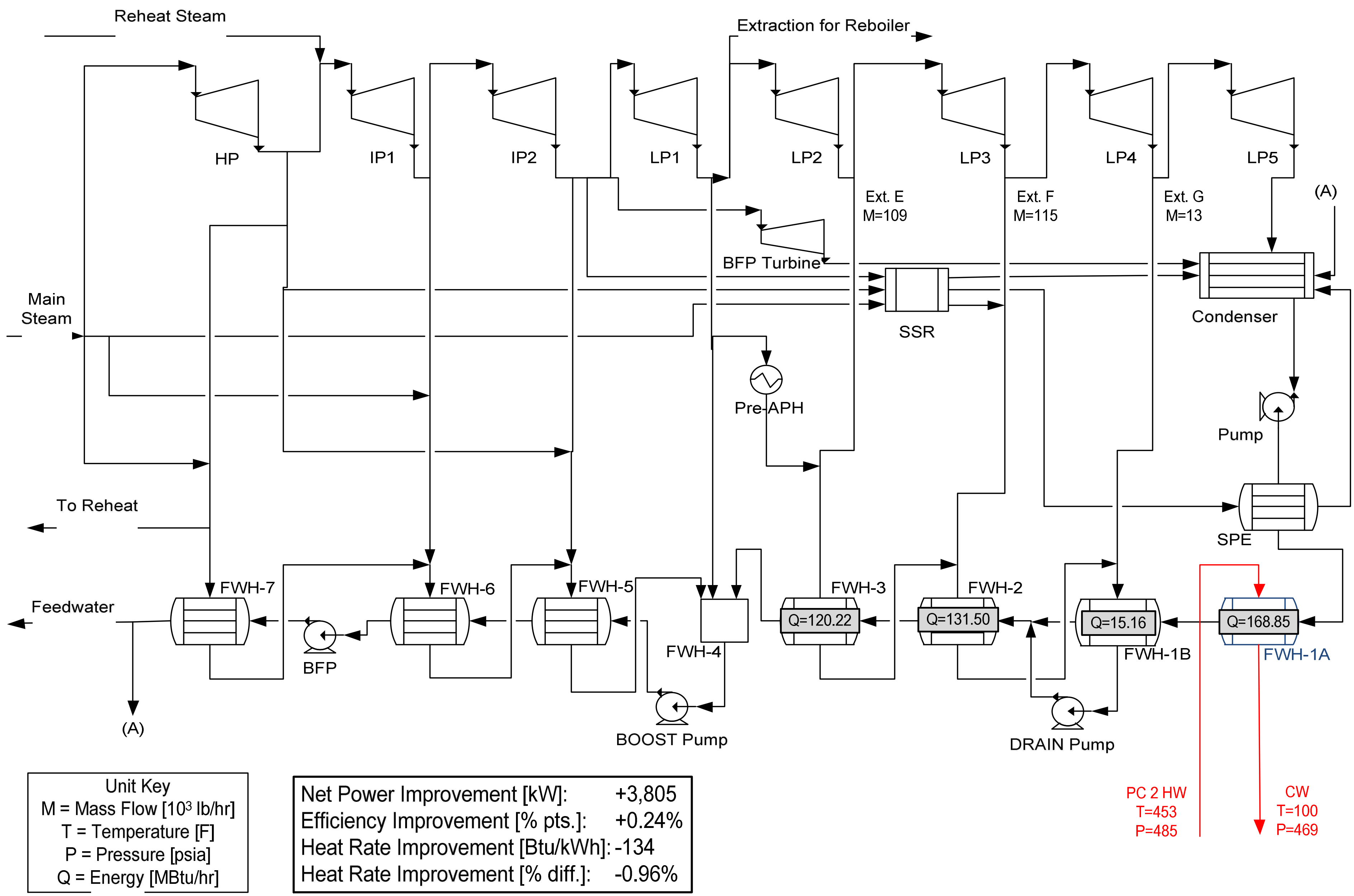


Figure 58. Ramgen PC2 to Feedwater Heater 1 Thermal Integration Results.

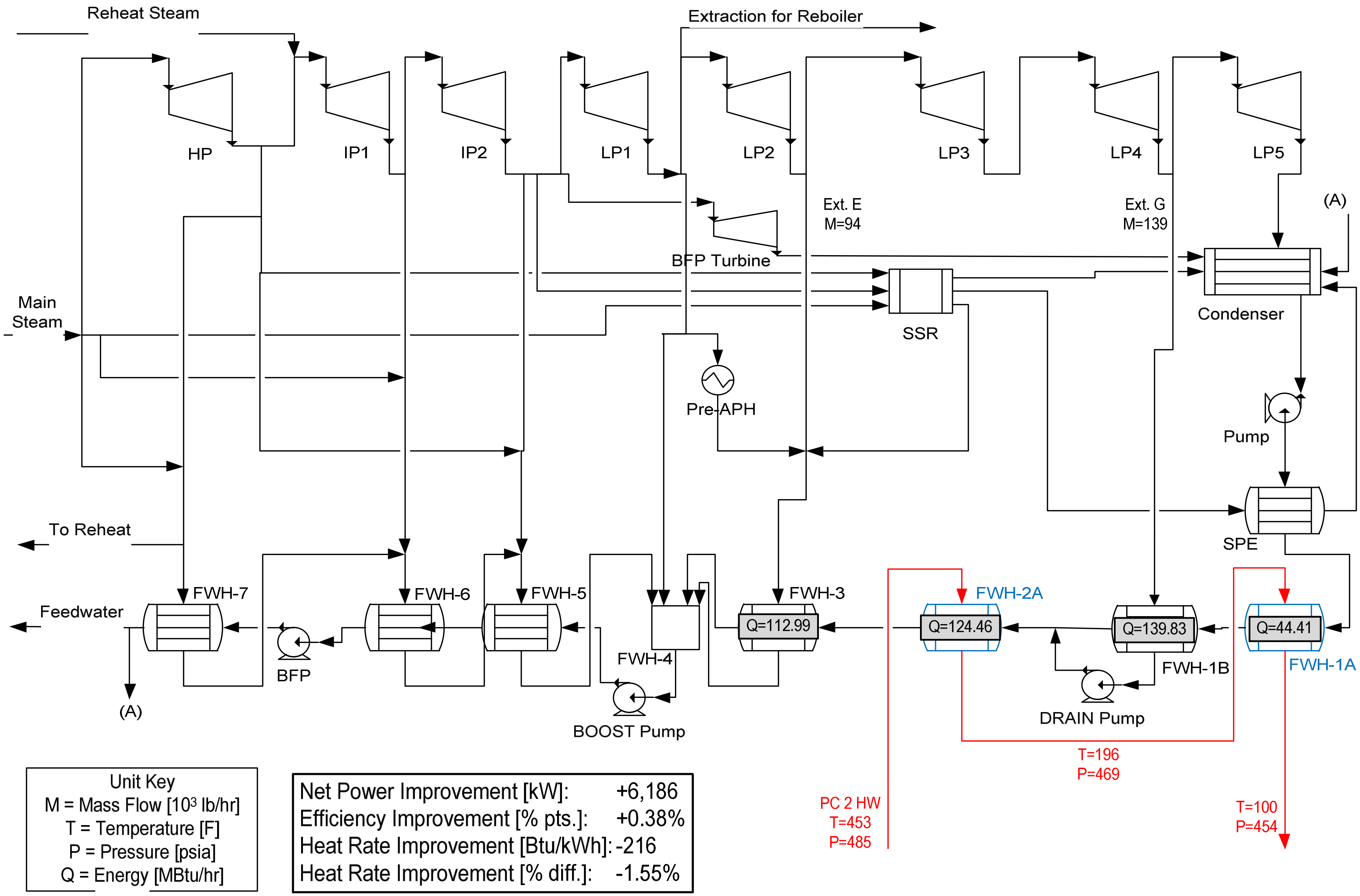


Figure 59. Ramgen PC2 to Feedwater Heater 2 Thermal Integration Results.

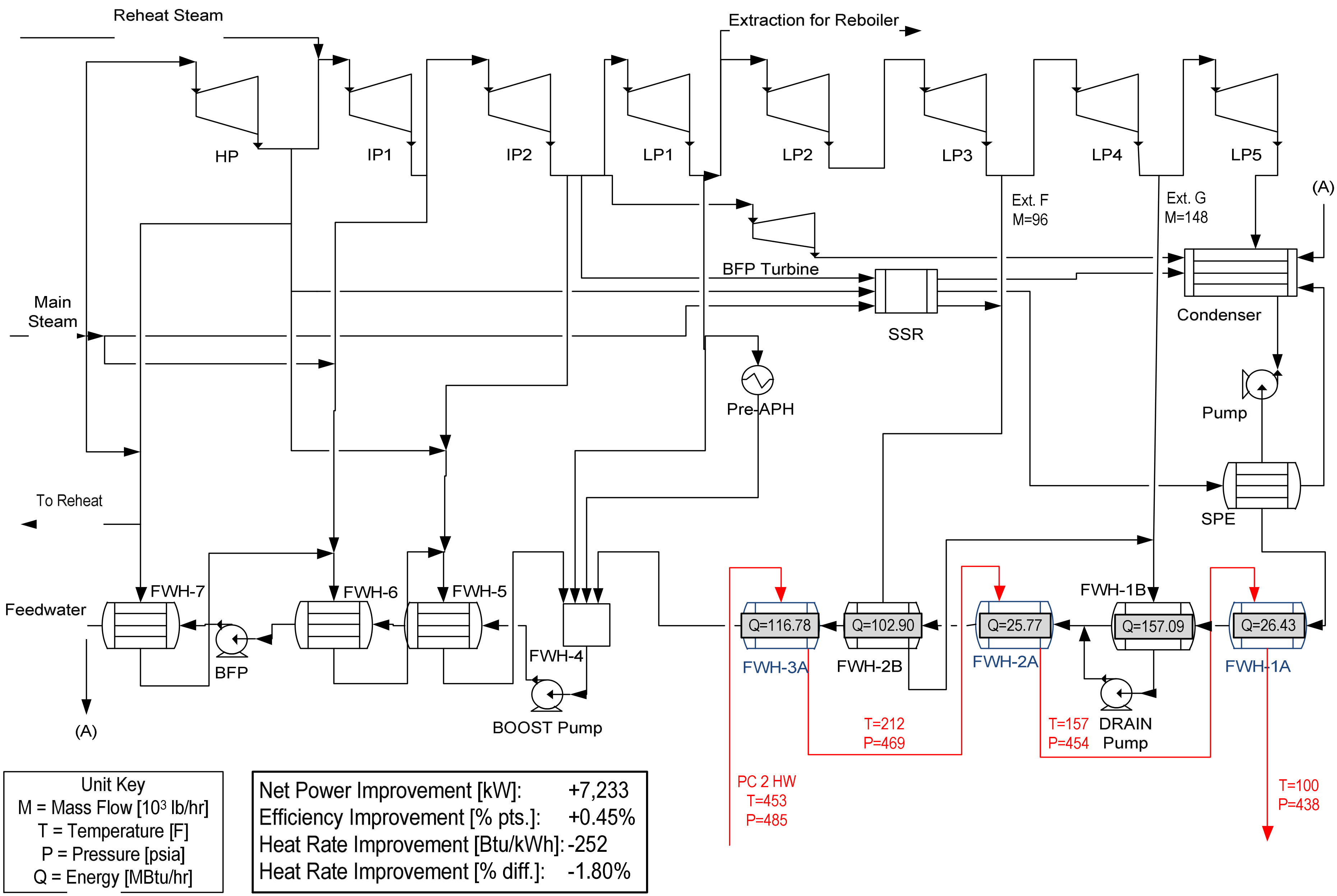
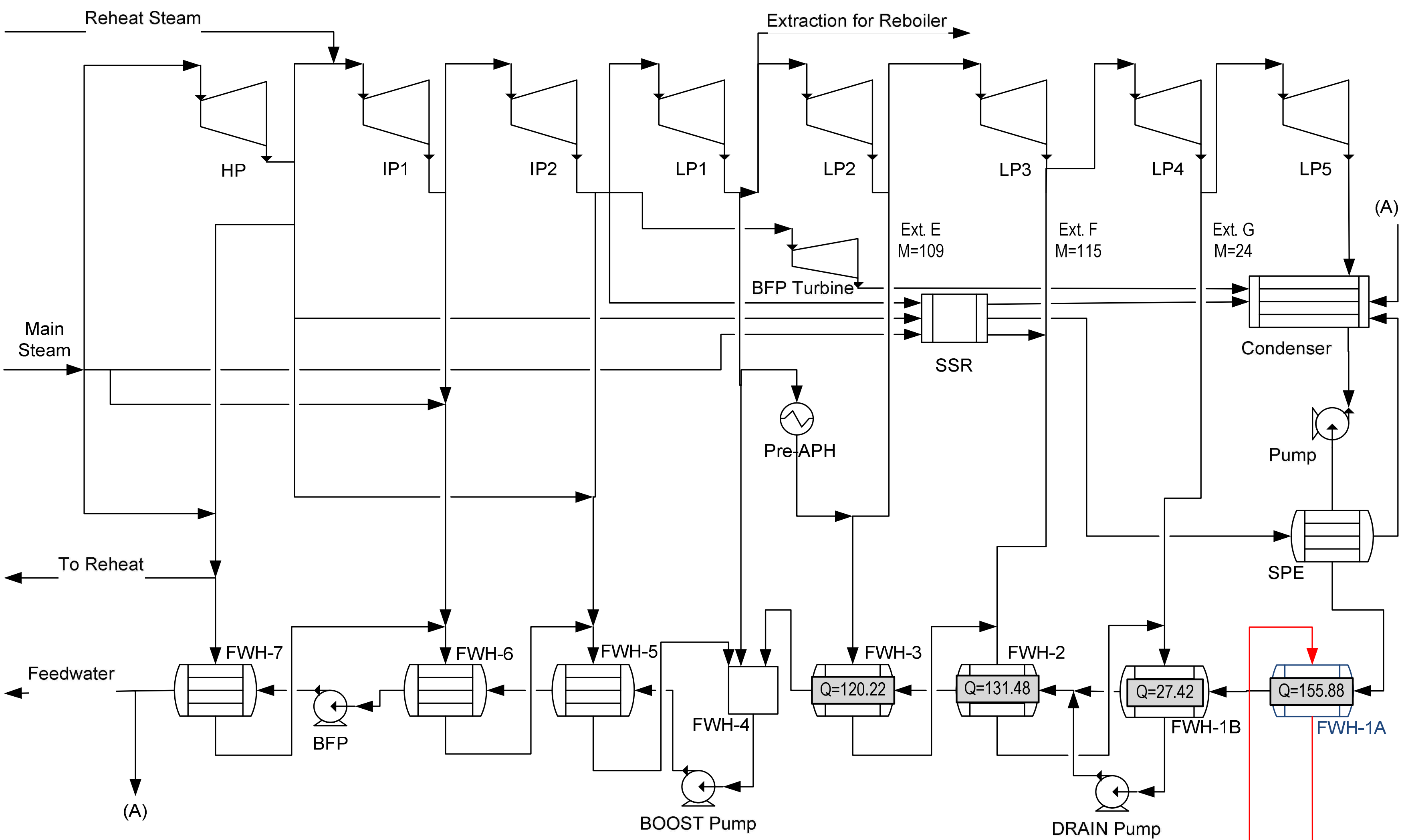


Figure 60. Ramgen PC2 to Feedwater Heater 3 Thermal Integration Results.



Unit Key
M = Mass Flow [10³ lb/hr]
T = Temperature [F]
P = Pressure [psia]
Q = Energy [MBtu/hr]

Net Power Improvement [kW]:	+3,555
Efficiency Improvement [% pts.]:	+0.22%
Heat Rate Improvement [Btu/kWh]:	-124
Heat Rate Improvement [% diff.]:	-0.89%

Figure 61. Inline 4 IC2 to Feedwater Heater 1 Thermal Integration Results.

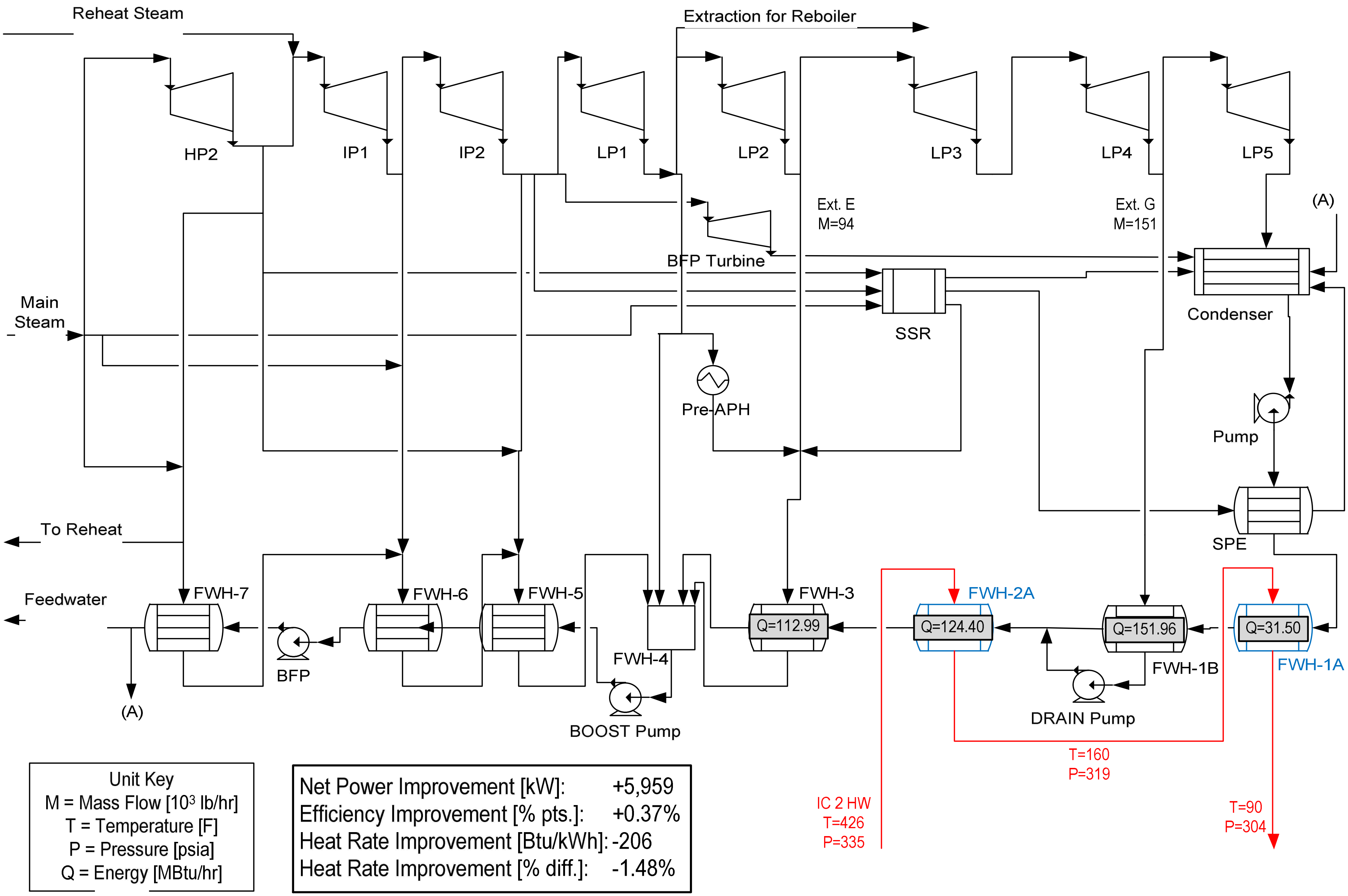


Figure 62. Inline 4 IC2 to Feedwater Heater 2 Thermal Integration Results.

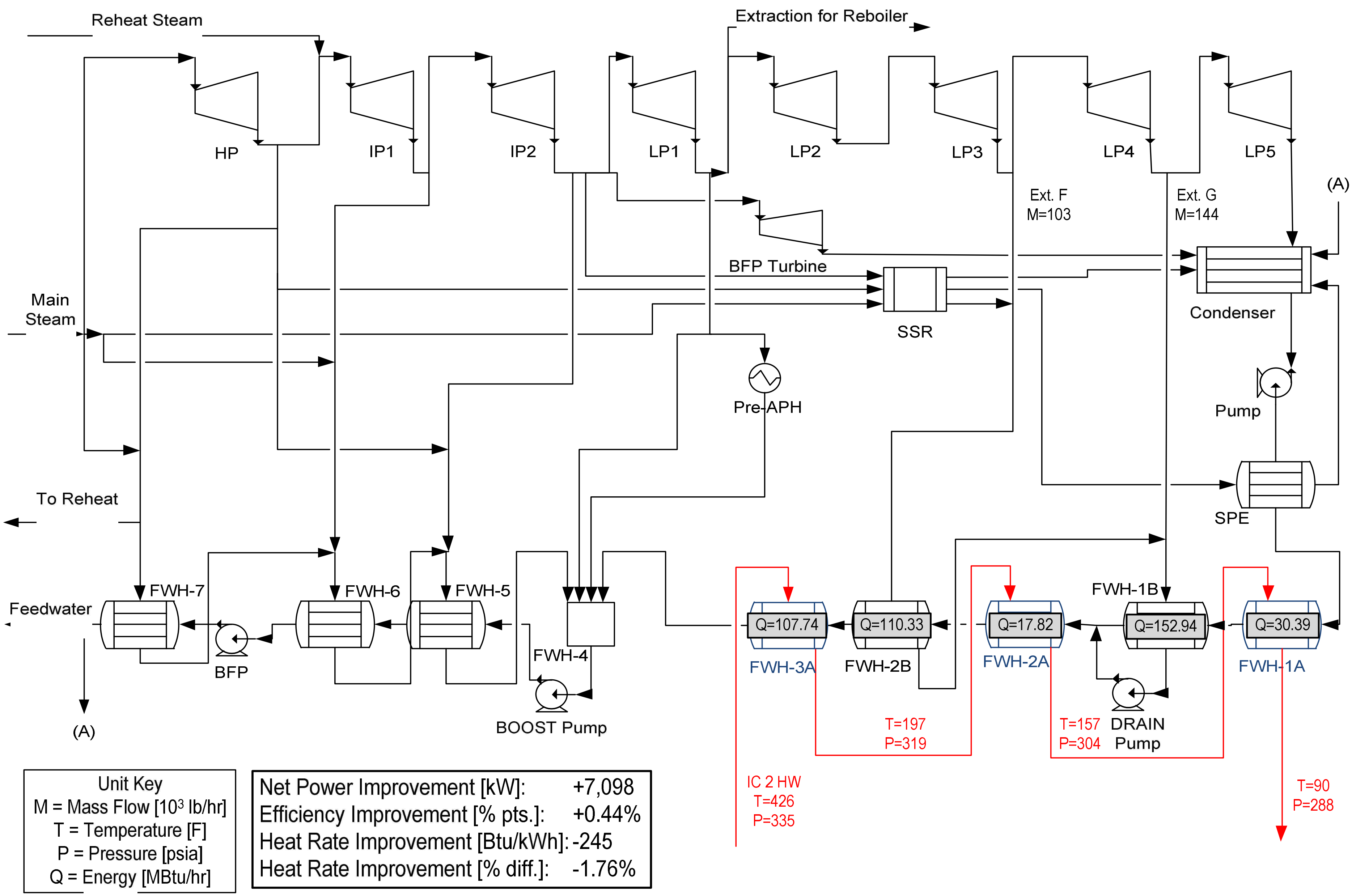


Figure 63. Inline 4 IC2 to Feedwater Heater 3 Thermal Integration Results.

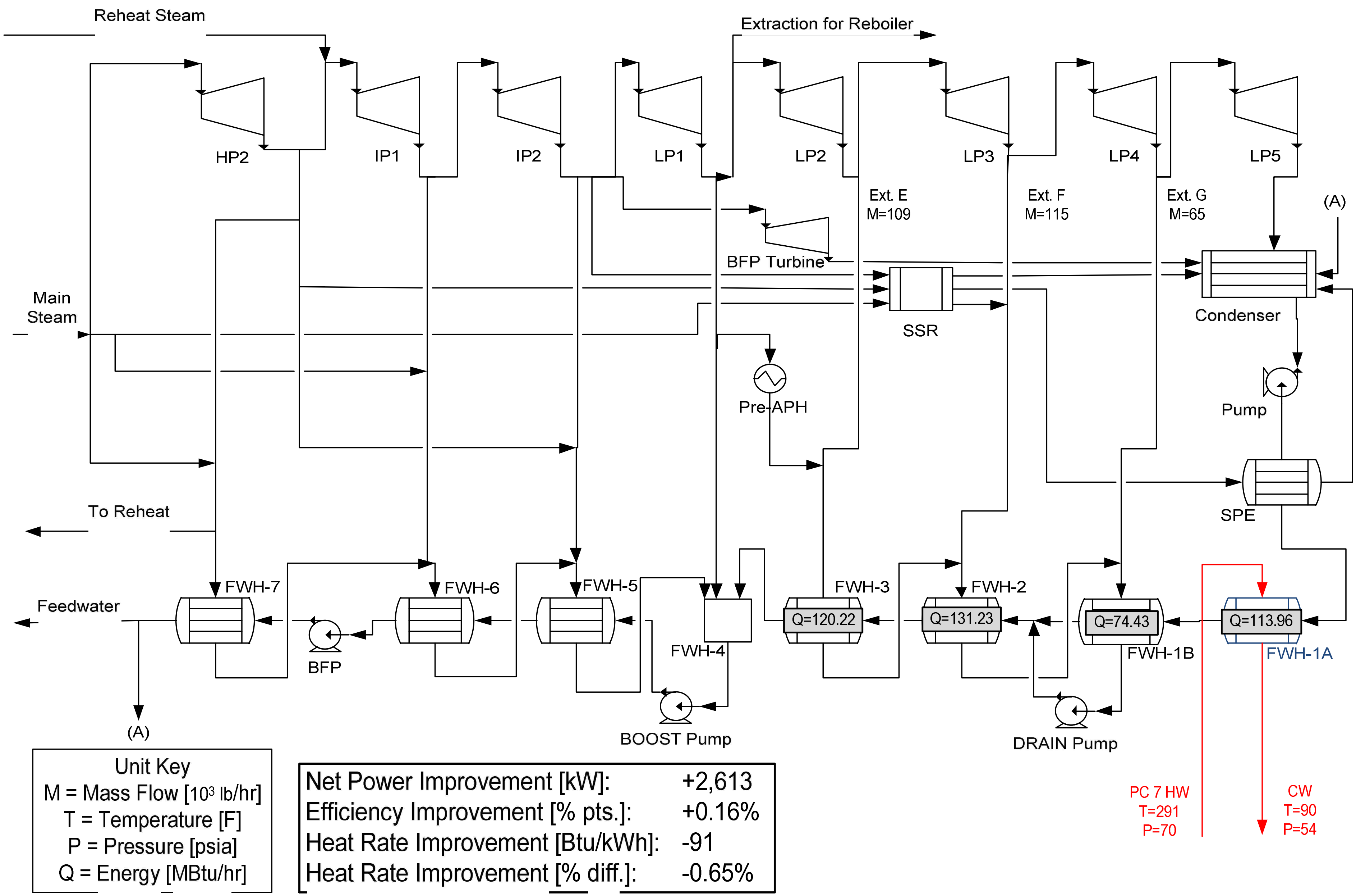
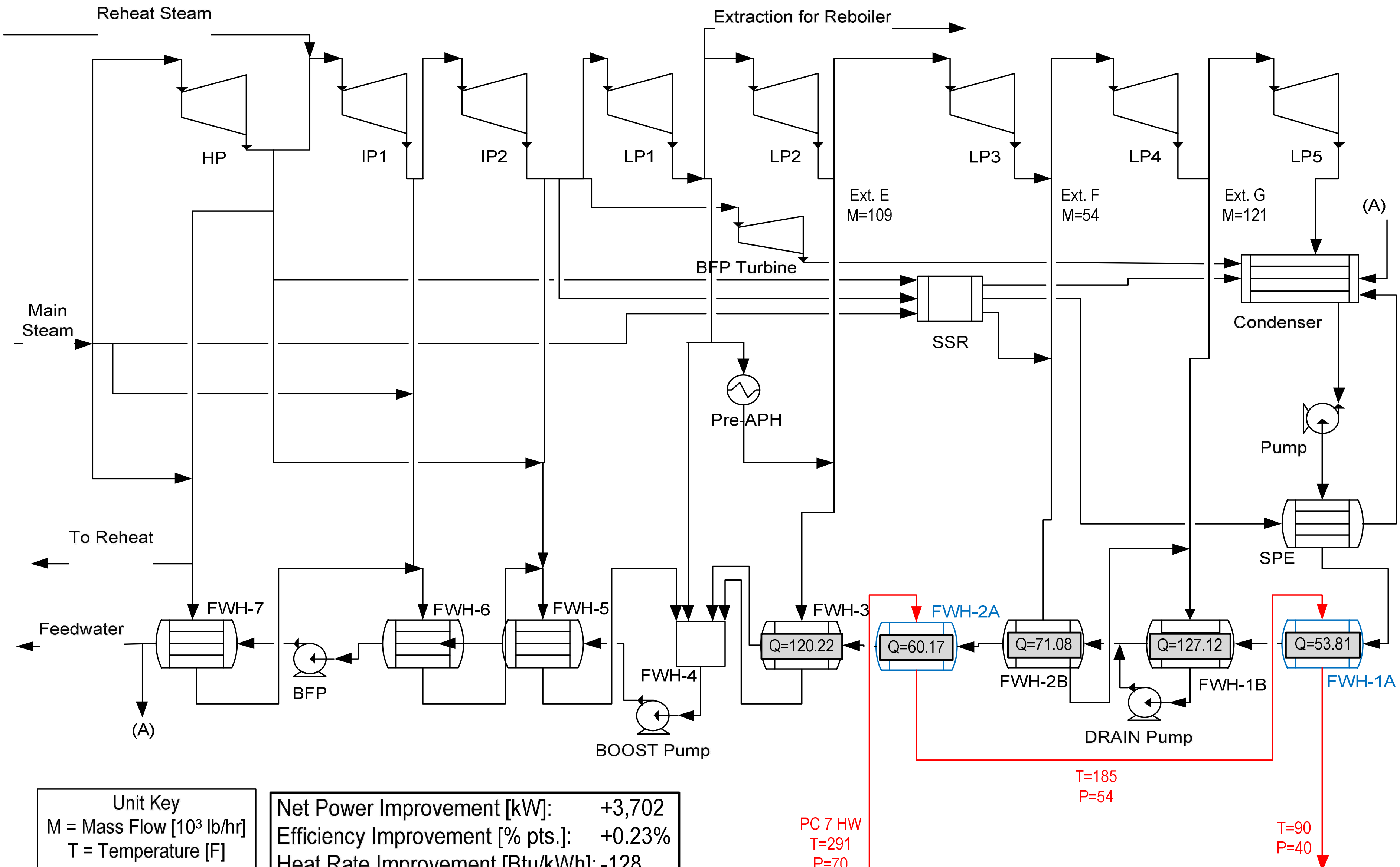


Figure 64. Integrally Geared 1 – 149 PC7 Feedwater Heater 1 Thermal Integration Results.



Unit Key
M = Mass Flow [10^3 lb/hr]
T = Temperature [F]
P = Pressure [psia]
Q = Energy [MBtu/hr]

Net Power Improvement [kW]:	+3,702
Efficiency Improvement [% pts.]:	+0.23%
Heat Rate Improvement [Btu/kWh]:	-128
Heat Rate Improvement [% diff.]:	-0.93%

Figure 65. Integrally Geared 1 – 149 PC7 to Feedwater Heater 2 Thermal Integration Results.

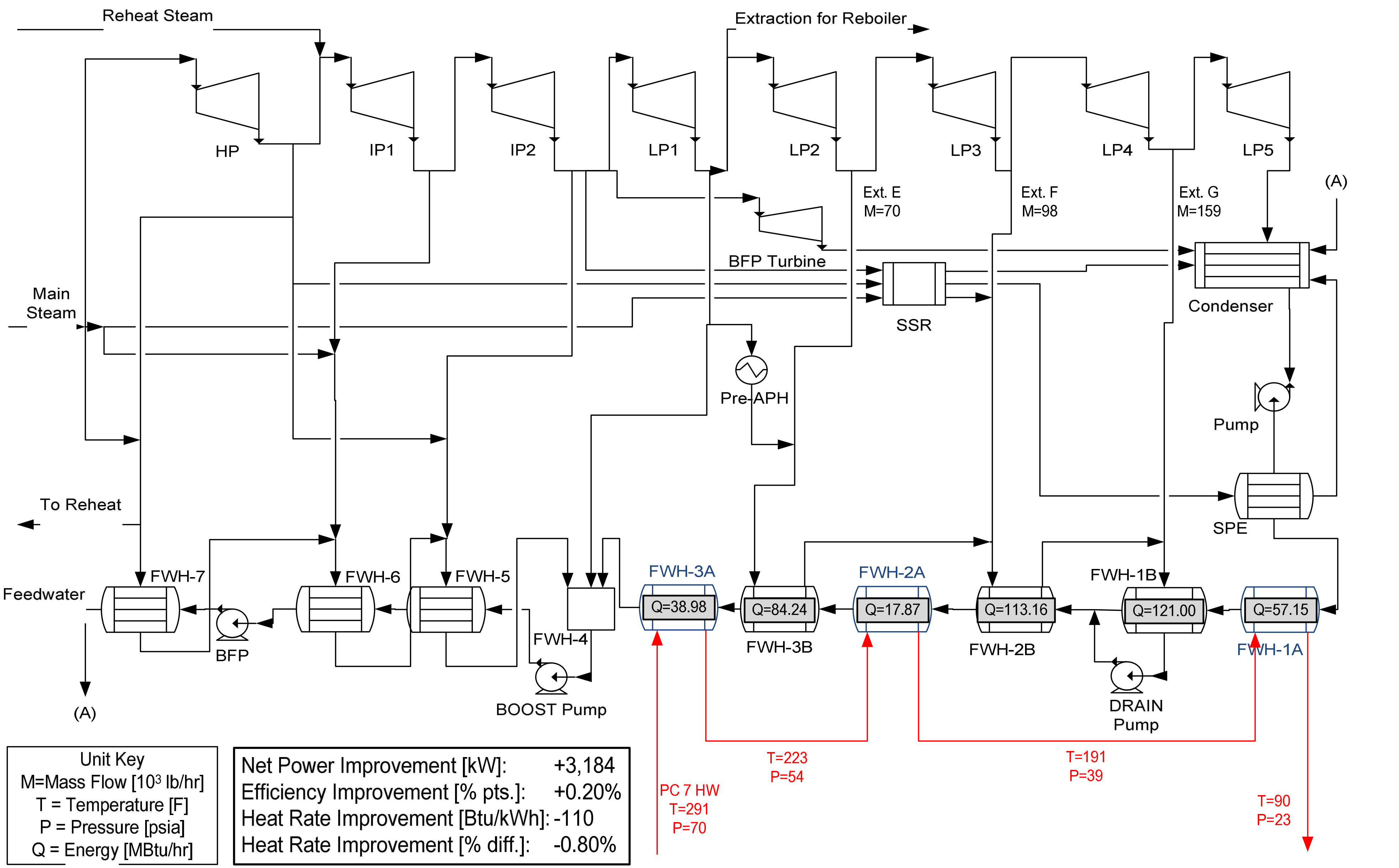


Figure 66. Integrally Geared 1 – 149 PC7 to Feedwater Heater 3 Thermal Integration Results.

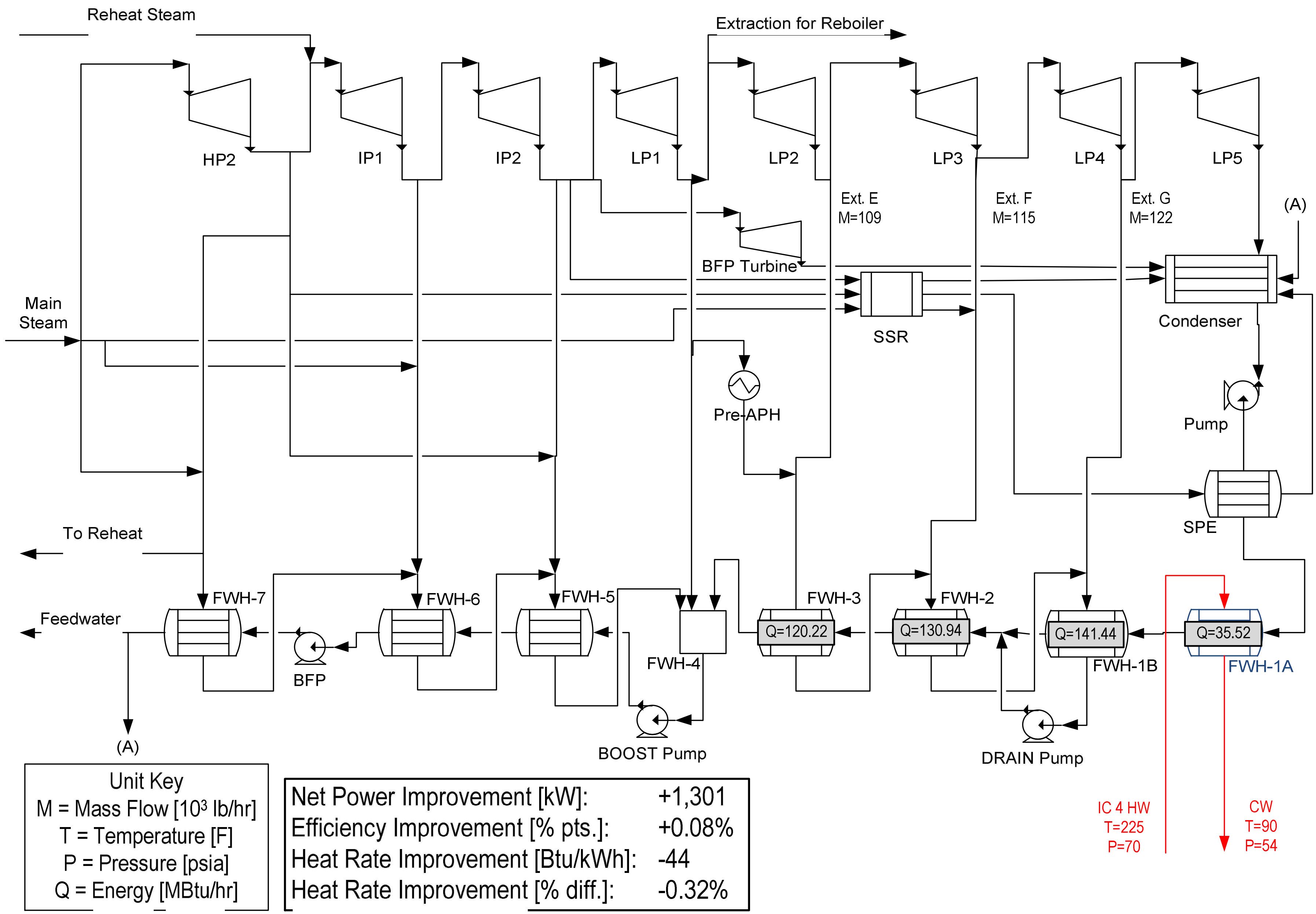
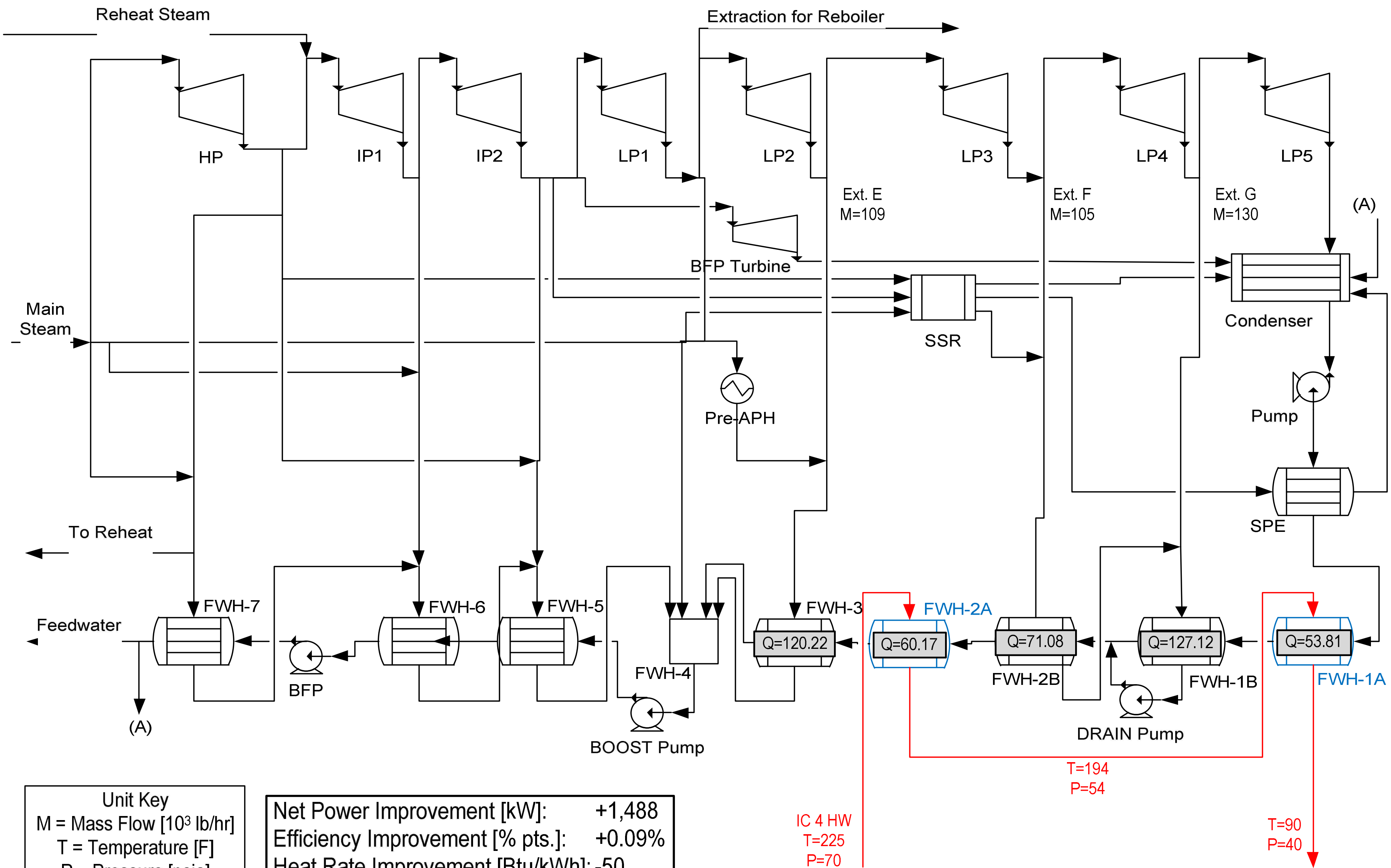


Figure 67. Integrally Geared 1 – 110 IC4 to Feedwater Heater 1 Thermal Integration Results.



Unit Key
M = Mass Flow [10^3 lb/hr]
T = Temperature [F]
P = Pressure [psia]
Q = Energy [MBtu/hr]

Net Power Improvement [kW]:	+1,488
Efficiency Improvement [% pts.]:	+0.09%
Heat Rate Improvement [Btu/kWh]:	-50
Heat Rate Improvement [% diff.]:	-0.37%

Figure 68. Integrally Geared 1 – 110 IC4 to Feedwater Heater 2 Thermal Integration Results.

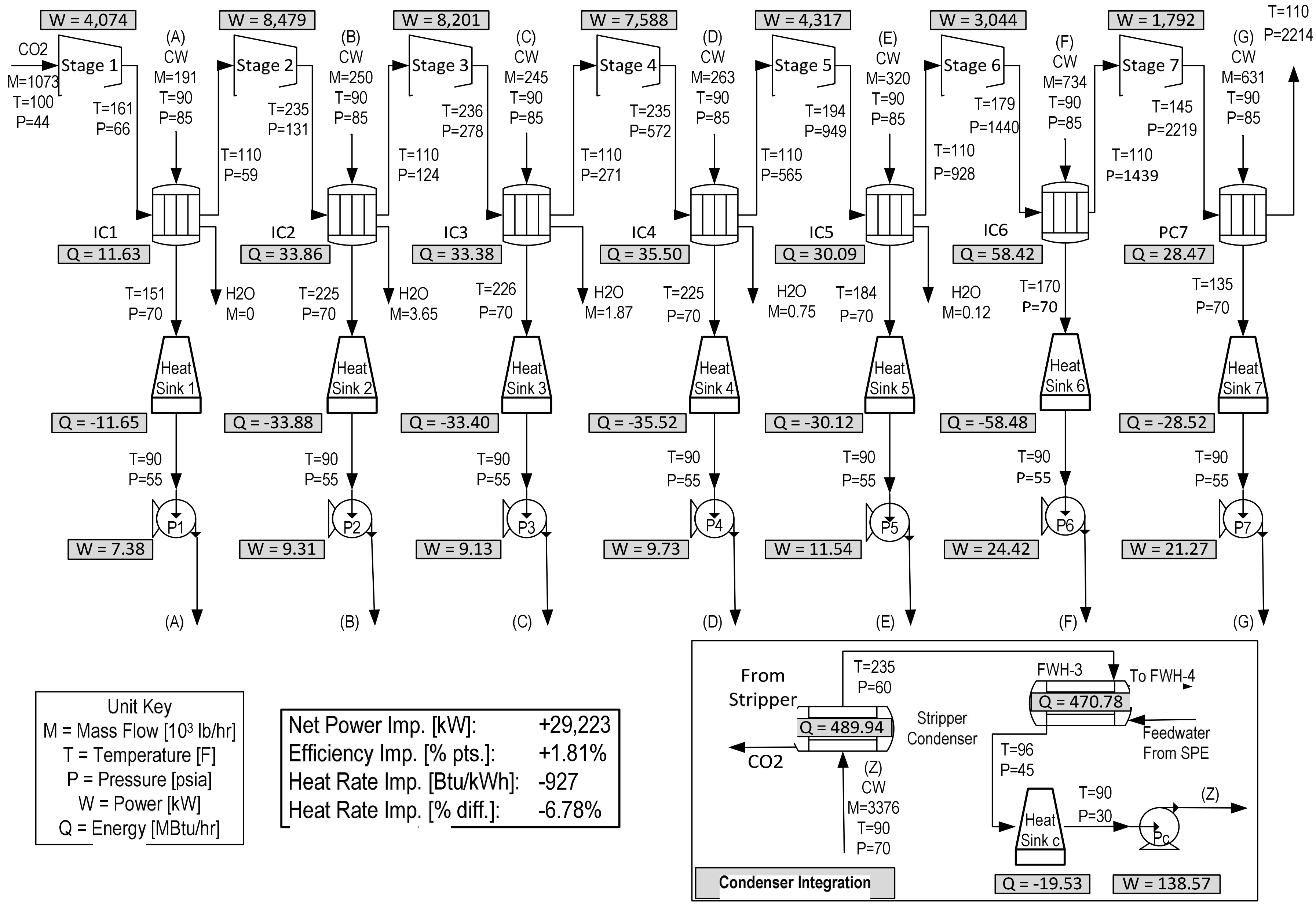


Figure 69. Stripper Condenser to FWH-3, with IG1 – 110 Base Case, Thermal Integration Results.

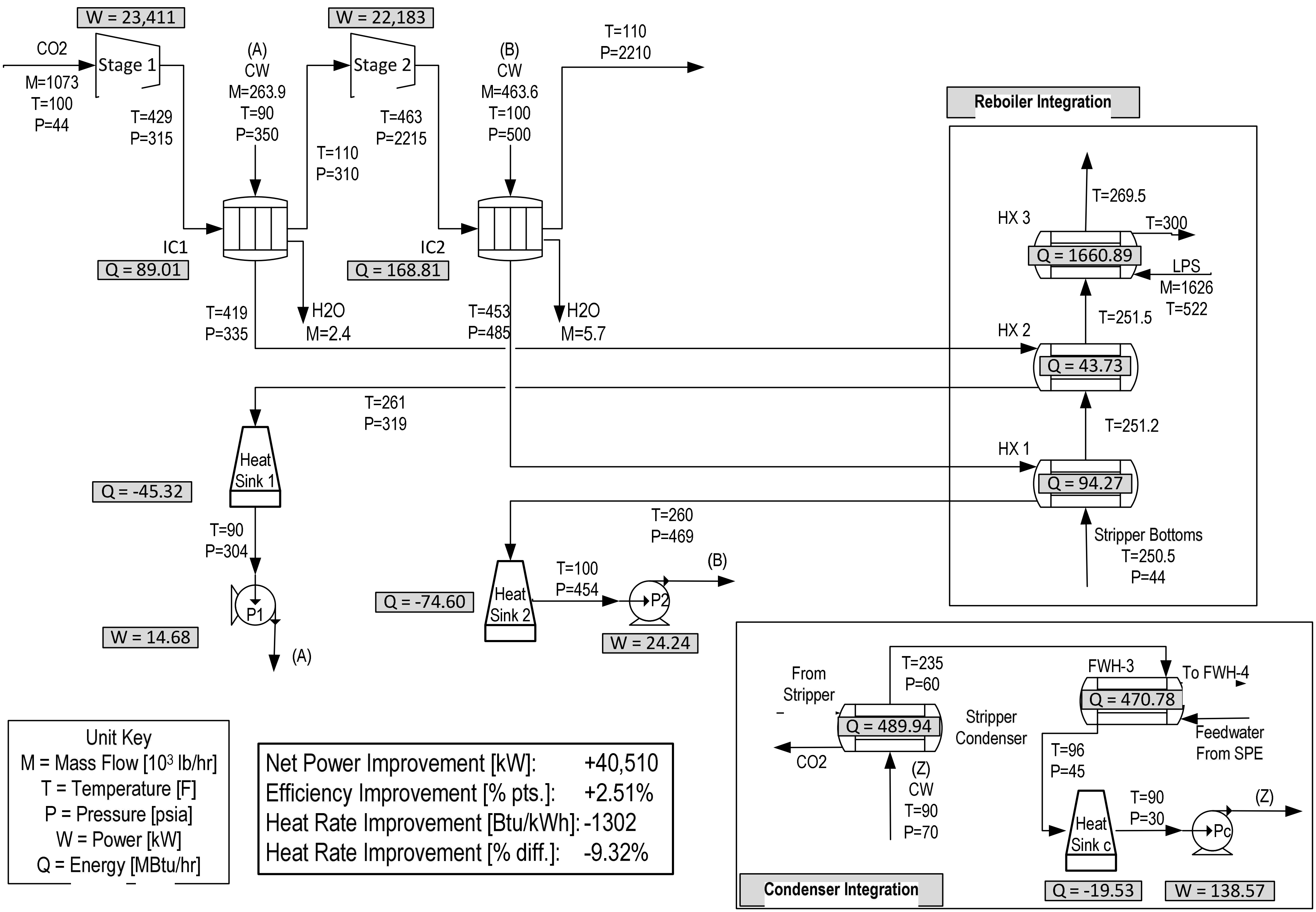


Figure 70. Combination of Ramgen Compressor to Reboiler and Condenser to FWH-3 Thermal Integration Results.

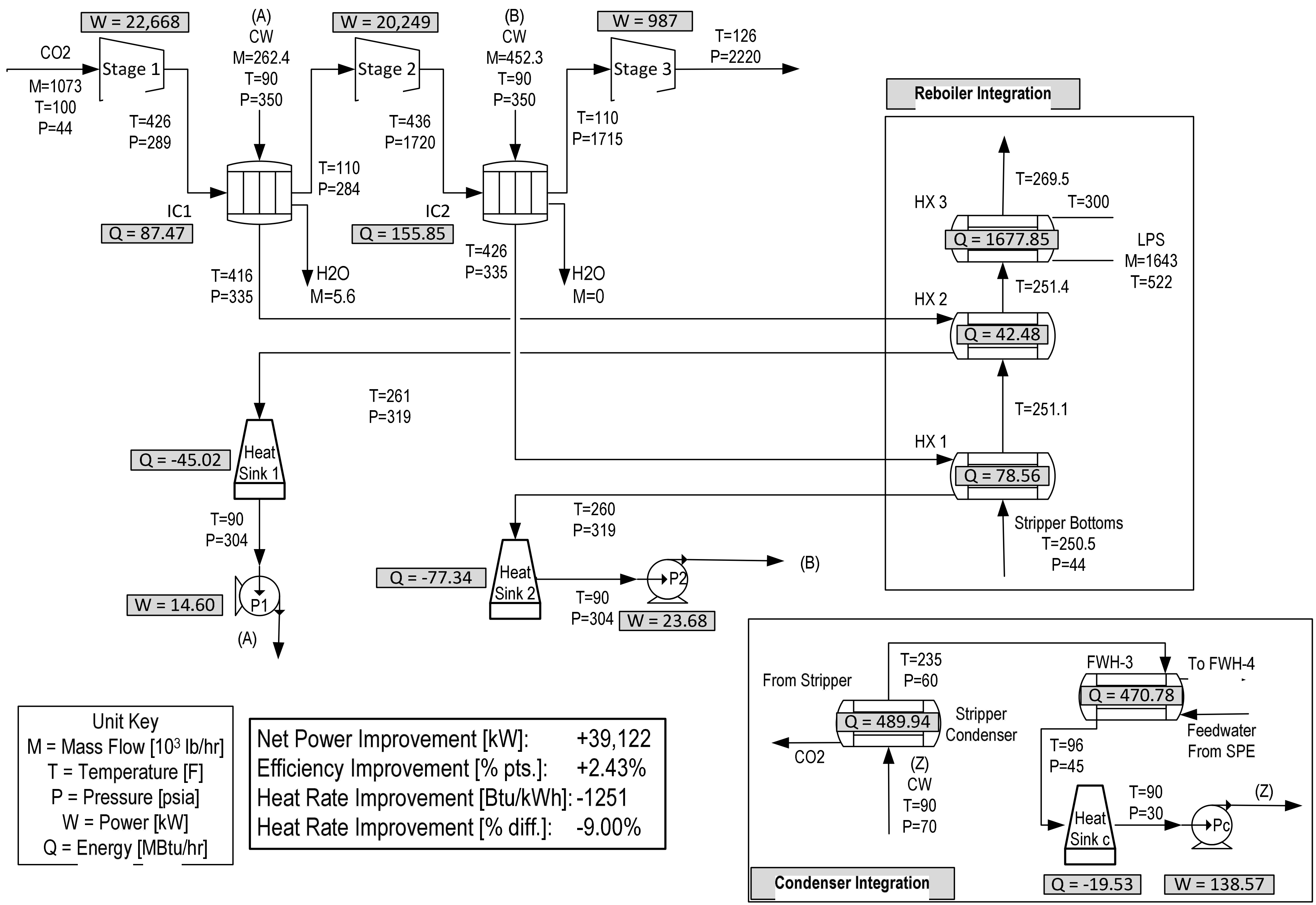


Figure 71. Combination of Inline 4 Compressor to Reboiler and Condenser to FWH-3 Thermal Integration Results.

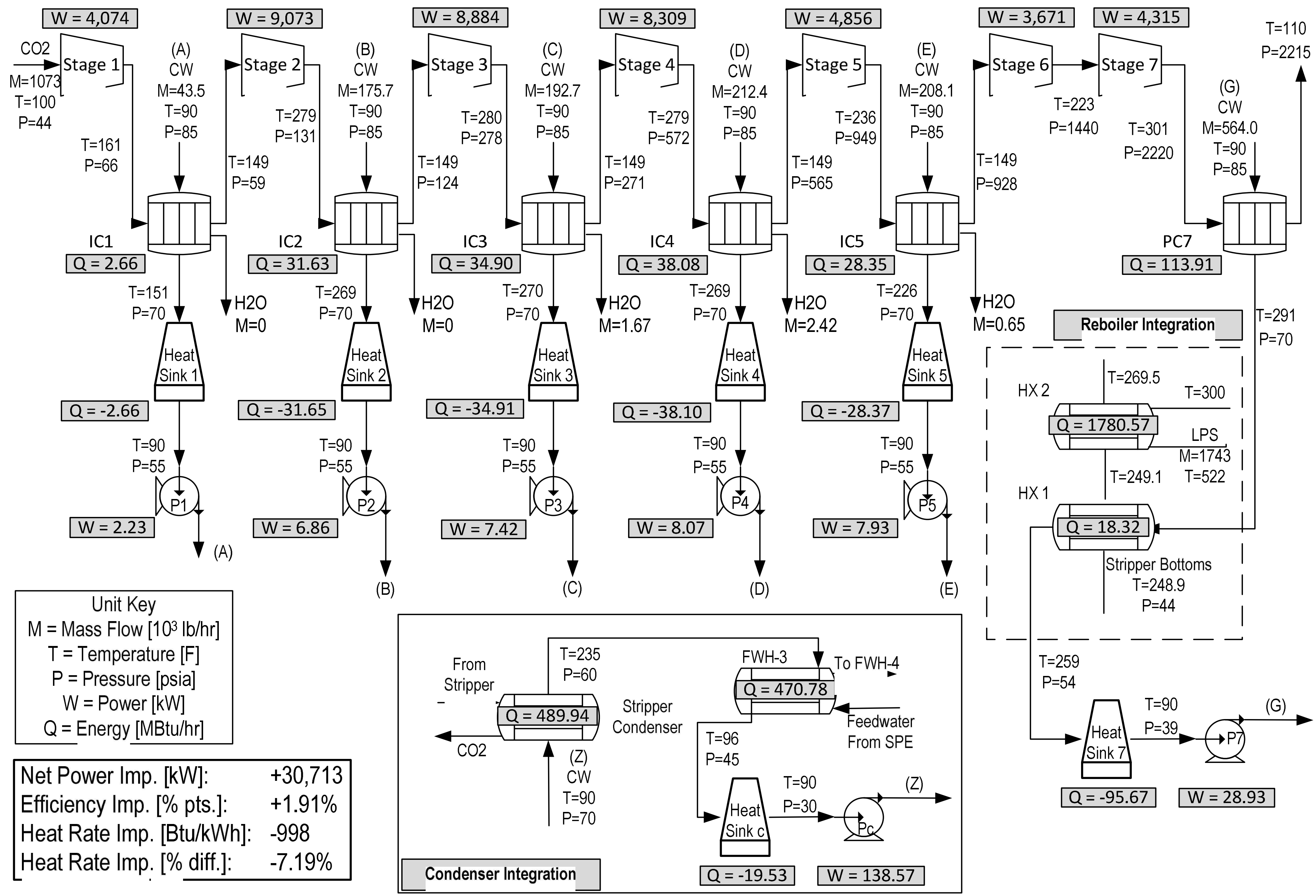


Figure 72. Combination of IG1 – 149 Compressor to Reboiler and Condenser to FWH-3 Thermal Integration Results.

Vita

Erony Whyte Martin grew up in Brunswick, Maine. She completed her undergraduate studies in Environmental Science and Biology at Ursinus College, Collegeville, Pennsylvania. She began her studies of mechanical engineering at Lehigh University in 2008 and began working for the Lehigh University Energy Research Center in May 2009. Erony currently works for LifeAire Systems, based in Allentown, Pennsylvania.