Engineering and Economic Analysis of an Advanced Ultra-Supercritical Pulverized Coal Power Plant with and without Post-Combustion Carbon Capture

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ABSTRACT

As the U.S. Environmental Protection Agency (EPA) and permitting agencies in countries around the world set limits on greenhouse gas emissions from power plants, it becomes more challenging for coal-based technologies to remain competitive. Adding carbon dioxide (CO_2) capture to a pulverized coal (PC) generating unit to meet these emission requirements decreases the plant's net power output and increases its capital and operating cost. One of the key strategies to mitigate this "penalty" for carbon capture is to increase the efficiency of the generating unit. Raising efficiency decreases coal consumption per megawatt-hour (MWh) of electricity produced, so less CO_2 is emitted. Increasing unit efficiency also decreases the operating costs associated with purchasing and handling coal, disposing of ash, and purchasing and preparing limestone for flue gas desulfurization and ammonia for selective catalytic nitrogen oxides (NO_X) reduction. The plant's water consumption is also decreased.

Although power plant designers can employ several approaches to increase the efficiency of a PC unit, the most significant gain is obtained by increasing the high-pressure and intermediate-pressure turbine inlet steam temperatures. Increasing the temperature above today's "state of the art" ultra-supercritical (USC) steam conditions requires the substitution of high nickel content alloys in place of ferritic steel alloys in the highest temperature and pressure sections of the boiler, steam turbine, and interconnecting piping. Such use of nickel alloys allows main and reheat steam temperatures in the range of 705–760°C (1300–1400°F). Plants operating at these steam conditions are referred to as advanced ultra-supercritical (A-USC) units.

In 2001, the U.S. Department of Energy (DOE) and the Ohio Coal Development Office (OCDO) launched a research program to develop and certify the nickel alloys required to operate a PC plant at A-USC steam conditions. Initial work focused on boiler components and was conducted by a consortium of U.S. companies and research organizations, including DOE's National Energy Technology Laboratory (NETL), OCDO, Energy Industries of Ohio (EIO), EPRI, ALSTOM Power, Babcock and Wilcox, Foster Wheeler, Oak Ridge National Laboratory, and Riley Power. A subsequent steam turbine consortium was formed, including many of the same organizations and General Electric. The consortia conducted tasks covering conceptual design, material properties, steamside oxidation, fireside corrosion, welding, fabricability, design data and rules, coatings, rotor/disc testing, blade/airfoil alloy testing, valve casting and testing, rotor alloy welding and characterizing, and casing welding and repair.

This report evaluates the economics and performance of two A-USC PC power plants:

- Case 1 is a conventionally configured A-USC PC power plant with superior emission controls, but without CO_2 removal.
- Case 2 adds a post-combustion carbon capture (PCC) system to the plant from Case 1, using the design and heat integration strategies from EPRI's 2015 report, "Best Integrated Coal Plant." The capture design basis for this case is "partial," to meet EPA's proposed New Source Performance Standard, which was initially proposed as 500 kg-CO_2/MWh (gross) or 1100 lb-CO_2/MWh (gross), but modified in August 2015 to 635 kg-CO_2/MWh (gross) or 1400 lb-CO_2/MWh (gross).

This report draws upon the collective experience of consortium members, with EPRI and General Electric leading the study. General Electric provided the steam cycle analysis as well as

the steam turbine design and cost estimating. EPRI performed integrated plant performance analysis using EPRI's PC Cost model.

Burning Powder River Basin (PRB) coal, the Case 1 configuration has an estimated efficiency of 41.4% on a higher heating value (HHV) basis. The arrangement of the boiler in PC Cost for this case is based on a "traditional" boiler configuration. Using on these assumptions, the plant in Case 1 has a capital cost of $2,933/kW and a levelized cost of electricity (LCOE) of $84.70/MWh.

Case 2 added a near term "advanced" solvent for CO_2 removal. The plant also included a high degree of heat integration to maximize the net power output while meeting CO_2 emissions regulations. With this design approach, for a proposed emission standard of 500 kg-CO_2/MWh (1100 lb-CO_2/MWh), plant efficiency dropped by only 2.5 percentage points to 38.9% (HHV basis). The capital cost of the A-USC plant increased by only 8%, but on a $/kW (net) basis, the cost increased to $3,370/kW (a 15% increase) due to the decrease in net power. The LCOE with CO_2 removal increased to $100.20/MWh. Analyses for the 635 kg-CO_2/MWh (1400 lb-CO_2/MWh) emission standard are under way. Preliminary results suggest a plant efficiency of 40.3% (HHV), a capital cost of $3,190/kW, and an LCOE of $93.25/MWh.

The high price of nickel piping has led engineering organizations to develop boiler and turbine configurations that minimize the amount of nickel piping required in the design. Two initial alternative designs are discussed in this report. Preliminary economic analyses suggest such designs could reduce the cost of an A-USC boiler by more than 10% relative to a conventional configuration, making A-USC technology competitive with the current "state of the art" USC PC units.

ACRONYMS AND SYMBOLS

AQCS	air quality control system
ASME	American Society of Mechanical Engineers
A-USC	advanced ultra-supercritical
DCS	distributed control system
DOE	U.S. Department of Energy
ESP	electrostatic precipitator
FD	forced draft
FGD	flue gas desulfurization
FSH	final superheater
FWH	feedwater heater
h_o	heat transfer coefficient at tube OD
h_{io}	heat transfer coefficient at tube ID referred to tube OD
HHV	higher heating value
HMB	heat and material balance
HP	high pressure
IP	intermediate pressure
LCOE	Levelized cost of electricity
LHV	lower heating value
LP	low pressure
MCR	maximum continuous rating
MEA	monoethanolamine
NETL	National Energy Technology Laboratory
OCDO	Ohio Coal Development Office
OD	outside diameter
PA	primary air
PC	pulverized coal
PCC	post-combustion CO_2 capture

PRB	Powder River Basin
SA	secondary air
SCR	selective catalytic reduction
TPC	total plant cost
USC	ultra-supercritical

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1
INTRODUCTION

This report describes the performance and cost of pulverized coal (PC) power plants with ultra-supercritical (USC) and advanced ultra-supercritical (A-USC) steam conditions, both without and with post-combustion capture (PCC) of CO_2. It draws upon past and current work by EPRI, General Electric, and members of the Advanced Materials for Ultra-Supercritical Boiler and Steam Turbine Consortia funded by the U.S. Department of Energy and the Ohio Coal Development Office. Results are presented for two design cases:

- Case 1 is a "conventionally configured" A-USC PC power plant (i.e., without CO_2 removal or novel equipment arrangements). New for 2015 is engineering and costing of the steam cycle and steam turbine by General Electric. Integrated plant performance analyses were conducted by EPRI; boiler performance and costing drew upon EPRI's PC Cost model.
- Case 2 adds a PCC system to the A-USC power plant from Case 1, using PCC design and heat integration strategies from EPRI's 2015 "Best Integrated Coal Plant" report.[1] GE also provided input on how to best integrate the steam cycle with the PCC system to optimize overall unit performance. The CO_2 capture rate is "partial," meaning to the extent necessary to comply with the U.S. Environmental Protection Agency's proposed New Source Performance Standard, which was initially proposed as 500 kg-CO_2/MWh (gross) or 1100 lb-CO_2/MWh (gross), but modified in August 2015 to 635 kg-CO_2/MWh (gross) or 1400 lb-CO_2/MWh (gross).

Design, performance, and cost information in this report also draws from other recent engineering-economic analyses by EPRI and members of the Advanced Materials for Ultra-Supercritical Boiler and Steam Turbine Consortia:

- A 2008 EPRI report[2] described the design, performance, and economics of a 750 MW (net) 1300°F (~700°C) series USC PC burning Powder River Basin (PRB) subbituminous coal. This design did not include CO_2 capture.
- A 2011 EPRI report[3] estimated the energy penalty associated with adding 90% CO_2 capture to the A-USC design from the 2008 report. The PCC system design employed a 30% (wt) monoethanolamine (MEA) solvent; design details are described in a 2010 EPRI report examining PCC for 1100°F (~600°C) USC PC units.[4]

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[1] *EPRI's Best Integrated Coal Plant with Post-Combustion Capture Case Study.* EPRI, Palo Alto, CA: 2015. 3002003740.

[2] *Engineering and Economic Evaluation of 1300°F Series Ultra-Supercritical Pulverized Coal Power Plants: Phase 1.* EPRI, Palo Alto, CA: 2008. 1015699.

[3] *Engineering and Economic Analysis of 1300°F Series USC Plant with Post-Combustion Capture.* EPRI, Palo Alto, CA: 2011. 1026645.

[4] *An Engineering and Economic Assessment of Post-Combustion CO_2 Capture for 1100°F Ultra-Supercritical Pulverized Coal Power Plant Applications: Phase II Task 3 Final Report.* EPRI, Palo Alto, CA: 2010. 1017515.

- A 2013 EPRI report[5] estimated the performance and cost of a "demonstration size" A-USC unit with a net output of 350–400 MW after CO_2 capture using a 30% (wt) MEA PCC system. The CO_2 capture design basis was not 90%, but rather targeted a stack emissions rate of 363 kg-CO_2/MW (net) or 800 lb-CO_2/MW (net), which is roughly equivalent to the emissions rate of a modern natural gas combined cycle unit without CO_2 capture.
- In 2013, EPRI evaluated a series of A-USC steam cycle temperatures and pressures for plants with and without PCC to identify "optimum" cycle conditions. This resulting report[6] was published in early 2014.
- In past evaluations of A-USC units with CO_2 capture, a PCC system using a 30% (wt) MEA solvent served as the design basis, primarily due to the large amount of information in the public domain. With reports of numerous advanced amine solvents having lower regeneration energy requirements, EPRI chose for its 2015 "Best Integrated Coal Plant" report[7] a "near term" advanced CO_2 removal solvent based on the research results from the National Carbon Capture Center in Wilsonville, AL. The overall plant performance with this PCC solvent was significantly better than that for the same plant using the previous MEA-based PCC design and, as noted above, serves as the basis for the Case 2 in this report.

A-USC Drivers

A significant component of the global drive to decrease emissions from pulverized coal plants focuses on increasing the efficiency of the generating unit. This reduces both the amount of coal required per megawatt-hour (MWh) of energy output and the plant's emissions, including CO_2. Where CO_2 capture is required to meet a regulatory emissions limit, the higher efficiency reduces the size and capital costs of the capture equipment. Additional benefits of higher efficiency include the decreased operating costs associated with purchasing coal, ash disposal, limestone for the flue gas desulfurization (FGD) unit, ammonia for the selective catalytic reduction (SCR) unit, reduced water consumption, and, where applicable, decreased costs for transport and storage of the CO_2.

State-of-the-art PC plants employ ultra-supercritical steam conditions, traditionally defined by EPRI as temperatures in excess of 593°C (1100°F). The maximum steam temperature typically used with the currently available ferritic steels is 610°C (1130°F) for the main steam and 621°C (1150°F) for the reheat steam. A-USC steam conditions are at temperatures above those of USC, typically in the range of 705–760°C (1300–1400°F). Realizing these higher steam temperatures requires a transition to high-nickel alloys in the boiler, steam piping, and steam turbine.

To date, significant progress has been made in identifying, evaluating, and qualifying the alloys needed for the construction of the critical components of coal-fired boilers and steam turbines capable of operating at higher efficiencies than ultra-supercritical plants. As discussed below, the scope of RD&D efforts encompasses the world's major coal-burning regions, and the materials

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[5] *Engineering and Economic Analysis of a 1300°F Series USC Demonstration Plant with Natural Gas Equivalency Post-Combustion Capture*. EPRI, Palo Alto, CA: 2013. 1026644.

[6] *Advanced Ultra-Supercritical Steam Cycle Optimization.* EPRI, Palo Alto, CA: 2014. 3002001788.

[7] *EPRI's Best Integrated Coal Plant with Post-Combustion Capture Case Study*. EPRI, Palo Alto, CA: 2015. 3002003740.

and fabrication technologies to build a first-of-a-kind demonstration A-USC plant have now become available.

RD&D Programs and Progress

The materials development programs discussed below are united by their aim to pursue higher steam conditions for coal-fired power plants. However, as Table 1-1 shows,[8] the focus of these diverse research efforts reflect variances in fuels and operating conditions, as well as the objectives of coal fleets and jurisdictions they serve.

Table 1-1
Comparison of Topics Pursued by International A-USC RD&D Programs

Research Topic	EU	US	Japan	China
Retrofit to older units	No	Yes	Yes	No
Cyclic operation	Yes	Yes	No	No
Oxy-combustion	Yes	Yes	No	No
High-sulfur coal firing on fireside corrosion	No	Yes	No	No
Biomass co-firing on fireside corrosion	Yes	No	Yes	No
Waste co-firing on fireside corrosion	Yes	No	No	No
Coatings	Yes	Yes	Yes	No
New 1200ºF (650ºC) steels	Yes	No	Yes	Yes
New nickel/iron alloys	No	No	Yes	Yes
New nickel alloys	No	Yes	Yes	No
Welded rotors	Yes	No	Yes	Yes

Europe

In the late 1990s, a consortium of major European power generators, equipment suppliers, and materials makers spurred the formation of the Components Test (ComTes) program. A component test facility, COMTES700, installed at E.ON’s Scholven Power Plant in Gelsenkirchen, Germany, provided materials exposure information at temperatures up to 700°C (1290°F) for large-scale boiler components and valves. The current ENCIO (European Network for Component Integration and Optimization) project is aimed at qualifying materials, components, and manufacturing processes, as well as erection and repair concepts, as a follow-up of COMTES700 activities with the objective of erecting and operating a new Test Facility.

[8] Adapted from: Nicol, Kyle. “Status of Advanced Ultrasupercritical Pulverised Coal Technology in 2013,” presented at the IEA Second AUSC Workshop, Rome, Italy, October 2014.

Unit 4 of ENEL's Andrea Palladio Power Station in Fusina, Italy, has been selected for installation of the Test Facility. Four test loops are planned for 20,000 hours of operation at 700°C (1290°F).[9]

Test Loop 1: Development of pipe repair concept

Complementary to COMTES700, different heat treatments (pre-/post-) will be applied and a trial weld of each material combination and/or welding method will be tested destructively. Trial welds will be carried out, and those without any crack indication will then be selected to produce components to be installed in the test loop.

Different nondestructive test (NDT) methods will be applied and tested to assure high standards of quality.

Nondestructive surface tests will be performed frequently during operation for all test welds. A final repair with orbital tungsten inert gas (TIG) narrow gap and electrode welding will be executed on additionally aged pieces of A617B (having been in operation in COMTES700).

Test Loop 2: Test of Hot Isostatic Pressing (HIP) parts and weldments as well as life-time monitoring

HIP technology holds promise for fabrication of T-pieces, valve bodies, and turbine parts. Although this technology is commercially applied in other fields, it has not yet been adopted for boiler and turbine pressure parts in power plants (alloy 617B or alloy 625). HIP fabrication may substitute for expensive castings, yielding savings for applications at 700°C (1290°F) steam conditions.

NDT will be performed frequently during operation for all test welds. After the end of operation, final repair welds will be executed on all dismantled pipes of Test Loop 2.

The creep behavior of alloy 617B will be monitored by running tests under respective load and temperature, as well as by using a thin-wall piece designed for ~30,000 hours (measuring and monitoring of the creep online).

Test Loop 3: Test of different Ni-based alloys and elements

The optimized chemical composition of A617B, known as A617OCC, will be used to explore possible improvements in weldability through decreased formation of chromium carbides. Additionally, an optimized melting process will be implemented to reduce the amount of impurities in the ingot. Such an optimization has the potential to make welds more reliable. This is also expected to be an option to reduce relaxation-cracking and hot-cracking occurrences. Due to the new melting process, the improved weldability may lead to fewer and simpler pre- and post-weld heat treatment requirements, providing a possible cost reduction for 700°C (1290°F) steam cycle applications.

Other Ni-based alloys such as A263, HR6W, and A625 will be tested and the weldments of the material combinations A617B OCC–HR6W and A263–A625 cast will be tested and compared.

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[9] Di Gianfrancesco, A. et al. "Encio Project: An European Approach to 700°C Power Plant." presented at the Seventh International Conference on Advances in Materials Technology for Fossil Power Plants, Waikoloa, Hawaii, October 2013.

This is necessary as the combinations A263–A625 cast (neither with nor without heat treatment) and A617B OCC–HR6W have not yet been tested and investigated with the required heat treatment.

NDT will be performed frequently during operation for test welds. After the end of operation, final repair welds will be executed on dismantled pipes of Test Loop 3, which are long enough for this purpose, followed by microstructural investigations and mechanical and creep testing.

Test Loop 4: Test of turbine cast material and weldments

This will provide a platform to test thick-wall welds in the range of the real pipe dimensions of a demonstration plant with material combination alloy 617 OCC–alloy 625 cast. This combination was not tested at real dimensions in COMTES700.

Although there was no negative indication about the behavior of the welded material in the combination alloy A617B forged–A625 cast material, the dimensions of the test weld in COMTES700 had been at relatively thin wall thickness (80 mm or 3.15 in). Thus, thick-walled welds in the range of the real cast dimensions of a demonstration plant with material combinations A617B OCC–A625 cast need to be tested. The weldments A625 cast–A625 cast with wall thickness in the range of 150 mm (5.9 in) was not foreseen in earlier designs for this component, but with the post-weld heat treatment needed for A617B OCC–A625 cast, this design also needs to be tested.

United States

In 2001, U.S. DOE/NETL—in conjunction with the Ohio Coal Development Office (OCDO), major boiler- and turbine-equipment manufacturers, and other key groups[10] including EPRI—launched a research program to develop and certify nickel alloys to achieve boiler and turbine steam conditions up to 760°C/35 MPa (1400°F/5000 psi). Table 1-2 lists the alloys selected for evaluation.[11]

Table 1-2
Nickel-Based Alloys under Evaluation

Alloy	Component	Comments
Alloy 263	Castings, Rotor	Back-up cast alloy to Haynes 282, good castability and weldability, lower strength but good ductility
CCA617	Superheater/Reheater, Pipe	Higher strength than Inconel 740, but not enough data to change American Society of Mechanical Engineers (ASME) code stress values, not suitable for high-sulfur coals, only certain welds are successful, strain-age cracking concerns, low-strength limits applicability for turbine rotor
Haynes 230	Superheater/Reheater, Pipe	Successful welding trials, maximum-size limitations for pipe may limit applicability

[10] ALSTOM Power, Babcock and Wilcox, Foster Wheeler, General Electric, Oak Ridge National Laboratory, and Riley Power, Inc.

[11] *Combustion Technology Status 2014.* EPRI, Palo Alto, CA: 2014. 3002003618.

Alloy	Component	Comments
Haynes 282	Castings, Rotor	Higher creep strength than Inconel 740, relatively insensitive to starting microstructural condition, good forging ‘window’ for rotor, can be cast for valves and casings
Inconel 740/740H	Superheater/Reheater, Pipe	Highest strength alloy in ASME code to achieve up to 760°C (1400°F), excellent fireside corrosion resistance, successful fabrication and welding, prime candidate for boiler components, cannot be air cast for valves and shells
Nimonic 105	Bolts, Blades	Highest creep-strength alloy, only considered for bolting and blading (non-welded components)
Waspalloy	Rotor, Bolts, Blades	Back-up alloy with good turbine history, cannot be welded reliably, poor ductility

A significant milestone for the consortium was achieved in 2011 with the successful approval of Inconel Alloy 740 by the American Society of Mechanical Engineers Boiler & Pressure Valve Vessel Code Committee (Code Case 2702). This material is rated for continuous operation at steam conditions of 760°C (1400°F).

The consortium’s boiler project has operated a steam test loop manufactured with materials of interest at Southern Company’s Plant Barry in Alabama for more than 17,000 hours at temperatures exceeding 760°C (1400°F). The loop contained a combination of 94 specimens with eight different superalloys and three different surface coatings. Upon removal at the end of testing, the components appeared to be in good condition and to have retained their mechanical integrity. The components are undergoing an extensive corrosion, oxidation, and material properties examination at Alstom’s Material Technology Center in Chattanooga, Tennessee.[12]

With the successful test of the steam loop at Plant Barry, the next logical step is to move toward a commercial demonstration of the technology. The consortium is now developing a Component Test (ComTest) Program to remove the identified risk barriers to full-scale implementation of advanced materials. The five major areas of development cover:[13]

- **Membrane Wall—**This component will be to use optimized manufacturing routes for T91/92 and other alternative materials to prove manufacturability and operation of an A-USC membrane wall and reduce the risk of early life issues. Main steam from the host utility will be cooled before flowing through the membrane wall section, which will operate at temperatures expected in an A-USC boiler.
- **Superheater—**The ComTest superheater will be installed inside the boiler and used to heat the main steam temperature up to 760°C (1400°F). As a result, the material will be exposed to the high temperatures and gas/ash constituents found in a utility boiler. Because the boiler may cycle during daily load changes, a gas-fired auxiliary superheater will be installed at the

[12] http://www.alstom.com/press-centre/2014/12/major-milestone-achieved-in-the-development-of-advanced-ultra-supercritical-steam-power-plants/

[13] *Combustion Technology Status 2014*, EPRI, Palo Alto, CA: 2014. 3002003618.

outlet of the superheater to accommodate swings in load and provide a constant temperature for downstream equipment.

- **High-Temperature, High-Pressure Steam Valve Testing—**The ComTest steam turbine will operate at a reduced pressure to increase steam volumetric flow. However, it is important to be able to design and operate the steam stop and control valves at the temperature and pressures expected during commercial A-USC operation. It is anticipated that two valve systems will be installed in parallel and cycle continuously during the test. Producing these valves, as well as the other valves needed for ComTest, will allow for the qualification of the vendors to produce high-quality valves from nickel alloys and determine the performance of the valves at these operating conditions.
- **Desuperheater and Thick-Wall Section—**Steam temperature control will be required for A-USC boilers. Injection of water in the desuperheater will produce a temperature gradient at the nozzle connection as well as along the length of the device. The A-USC desuperheater will be based on a conventional design, but with advanced materials to determine their reliability in this service. The thick-wall test section will be a component similar to a header with tube penetrations. By building the thick-walled component, the supply chain of these header sections will be developed and shop manufacturing processes and procedures will be proven. The desuperheater will be used to thermally cycle a thick-wall test section to evaluate its durability.
- **Steam Turbine—**The steam turbine for ComTest will operate with an inlet temperature of 760°C (1400°F). ComTest objectives include the selection of materials and manufacturing technologies required for reliable steam turbine operation, evaluation of component life, inspection of any oxidation and/or deposits that develop over the testing period, and the evaluation of steam path, sealing, and bearing design.
- As originally conceived, all of these components would be installed in, and adjacent to, a utility USC power plant. However, as the consortium searched for hosts for the ComTest, Youngstown Thermal in Youngstown, Ohio, expressed interest in hosting the lower-pressure components, primarily the A-USC steam turbine. Youngstown Thermal is currently a district heating facility that provides saturated steam to its customers in the downtown area. The envisioned modification would accommodate the ComTest high-temperature (i.e., A-USC) turbine employing a turbine inlet valve pressure of 585 psig and an exhaust pressure of 170 psig. Proposals for this project have been submitted to DOE and are under review as of mid-2015. Discussions continue between the consortium and potential utility hosts about testing the steam generator components of the ComTest program that require high-pressure steam.

Japan

Since 2008, major Japanese equipment and materials manufacturers, utility companies, and two research institutes have been cooperating to develop 700°C (1290°F) class A-USC technology with the support from the Japanese government.

Boiler Technology Development. HR6W, HR35, Alloy 617, Alloy 263, Alloy 740, and Alloy141 are candidate Ni-based alloys developed under the Japanese program for use at temperatures higher than 650°C (1200°F).

High-boron 9Cr steel, low-carbon 9Cr steel, and SAVE 12AD are ferritic steels for use at temperatures below 650°C (1200°F). These materials are being tested to verify their characteristics regarding creep rupture, fatigue, oxidation, and corrosion. Welding and bending tests have been conducted to ascertain the manufacturability of the materials.

HR6W is being developed by Nippon Steel and Sumitomo Metal for pipes and tubes in A-USC applications, with an emphasis on good corrosion resistance to combustion gas in the boiler and good creep strength for pressurized pipes and tubes at 700°C (1290°F). The 100,000-hr creep rupture stress at 700°C (1290°F) is expected to be about 90 MPa, meeting the program target for large steam piping materials.

In 2015 and 2016, boiler components such as superheaters, pipes, valves, and turbine casing will be tested using an actual boiler.

Turbine Technology Development. Three Ni-based alloys are being developed for use at temperatures higher than 700°C (1290°F), namely FENIX-700, LTES, and TOS1X. Each is expected to meet a target 100,000-hour creep rupture stress at 700°C (1290°F) of about 90 MPa (13,000 psi). Development of FENIX-700 aims to allow building of a rotor heavier than 10 tons without segregation in the material. Both LTES and TOS1X are targeted for weights of about 10 tons, which will be welded to steel parts to make a 30- to 40-ton rotor.

FENIX-700, which has superior long-term stability at 700°C (1290°F), was developed from Alloy 706 by reducing Nb content and increasing Ti and Al content. The 100,000-hour creep rupture strength at 700°C (1290°F) is expected to be higher than 100 MPa (14,500 psi).

LTES700R is a Ni-based alloy developed by MHI. This alloy was developed to have a thermal expansion coefficient similar to 12Cr steel, so it conforms well to conventional steels. In addition, the creep rupture strength of LTES700R is higher than the target for 700°C (1290°F) class rotor material. Originally, LTES700 was developed for small parts, such as casing bolts. LTES700R was developed from LTES700 for large steam turbine rotors. Welding technology is crucial for this material, and numerous tests have been conducted, including the welding of dissimilar materials.

TOS1X was developed from Alloy617. The earlier version of TOS1X, which is now called TOS1X-I, is expected to have approximately 200 MPa (29,000 psi) of 100,000 hour creep rupture strength at 700°C (1290°F). A piece of forged material, 1000 mm (39 in) in diameter and weighing 7 tons, has been made successfully using TOS1X-I. TOS1X-II was developed from TOS1X-I by increasing the Al and Ti content. TOS1X-II is also expected to have about 200 MPa (29,000 psi) of 100,000 hour creep rupture strength at 700°C (1290°F). A 13-ton piece of forged material of TOS1X-II has been made successfully.

Three rotors made of the three candidate rotor materials will be tested in 700°C (1290°F) atmosphere and at actual speed from 2014 to 2016. The rotors will be heated by electric heaters in a vacuum chamber and driven by an electric motor.

China

In 2010, China launched the National 700°C USC Coal-Fired Power Generation Technology Innovation Consortium, an 18-member body consisting of material research institutes, power plant equipment manufacturers, and power companies. The Consortium's R&D plan calls for a ten-year program culminating in a demonstration project.

Target steam parameters are 35 MPa/700°C/720°C (5075 psia/1290°F/1330°F) for a 600 MW unit. The program includes developing indigenous materials as well as modifying imported materials for further enhancement, including optimization of G115 and Inconel 740H.[14]

The Chinese test facility (CTH 700) will be similar to the European ComTes700 but with a two-pass 320 MW boiler. The pressure of the existing boiler will be raised to 25 MPa (3625 psia) and the temperature to 725°C (1340°F). The steam flow rate will be limited to 3 kg/s (6.6 lb/s). The construction of the test loop was scheduled to start in 2014. Test data will be collected for 100,000 hours.

India

A joint A-USC Project consortium between the Indira Gandhi Center for Atomic Research (IGCAR), Bharat Heavy Electrical Limited (BHEL, an equipment manufacturer), and the National Thermal Power Corporation (NTPC, India's largest power generation utility) entails a materials testing program specific to the combustion of Indian coals.

The consortium is working to select materials and develop welding and fabrication technologies, as well as collaborating with national/international institutions to support the A-USC project. They have designed and fabricated a steam loop, similar to the Plant Barry loop, with plans to install it into a boiler in 2014 and operate for two years. The preliminary conceptual design of the boiler is complete. Materials selected for use in the high-temperature zone of the boiler are SS 304HCu and Alloy 617M.[15]

The consortium envisions building an 800 MW plant with 300 bar/700°C/700°C (4350 psi/1290°F/1290°F) steam conditions with an efficiency target of 46% (HHV basis),[16] over a seven-year timeframe following government approval of funding.

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[14] Zhang, Dongke, Ed. *Ultra-Supercritical Coal Power Plants: Materials, Technologies, and Optimization,* Elsevier, August 2013.

[15] Gandy, D. and J. Shingledecker, eds. *Advances in Materials Technology for Fossil Power Plants: Proceedings from the Seventh International Conference,* October 2013, Waikoloa, Hawaii.

[16] http://www.projectsmonitor.com/daily-wire/india-may-have-advanced-ultra-super-critical-power-unit-by-2021/

2
DESIGN BASIS SPECIFICATIONS

Site Location

The plant is of indoor construction and is located in Kenosha, Wisconsin. The site is clear and level in a Seismic Zero Zone and 30-meters (100-feet) deep pile foundations are required. Available at the site boundary are rail and transmission access, raw water supplied from Lake Michigan, and natural gas is available.

Ambient Conditions

Annual average ambient air conditions shown in Table 2-1 are required to calculate material balances and thermal efficiencies, to develop system designs, and to size equipment.

Table 2-1
Annual Average Ambient Conditions

Site Elevation Above Mean Sea Level	183 m	600 ft
Atmospheric Pressure	1 bar	14.4 psia
Annual Average Ambient Air Dry Bulb Temperature	15.6°C	60°F
Maximum Ambient Dry Bulb Temperature	35°C	95°F
Maximum Ambient Wet Bulb Temperature	23.9°C	75°F
Seismic Zone	0	0

Environmental Requirements

The stack gas emission limits for the unit are listed in Table 2-2:

Table 2-2
Stack Gas Emissions Limits

Pollutants		Emission Limits (HHV Basis)
PM_{10}	~10 mg/m^3	0.01 lb/MBtu
$PM_{2.5}$	~13 mg/m^3	0.013 lb/MBtu
SO_2	~30 mg/m^3	0.03 lb/MBtu
NO_X	~30 mg/m^3	0.03 lb/MBtu
VOC	~10 mg/m^3	0.0025 lb/MBtu
Mercury	90% capture	

The permitted levels for wastewater disposal or discharge are shown in Table 2-3. Following treatment, the wastewater complies with the U.S. Environmental Protection Agency (U.S. EPA) Effluent Guidelines and Standards. (Interim Detailed Study Report for the Steam Electric Power Generating Point Source Category, November 2006).

Table 2-3
Wastewater Discharge Permit

Makeup Water	Makeup for potable, process, and de-ionized water are drawn from municipal sources.
Process Wastewater	Water associated with combustion activity and storm water that contacts equipment surfaces is collected and treated for disposal through a permitted discharge.
Sanitary Waste Disposal	Design includes a packaged domestic sewage treatment plant with effluent discharged to the industrial wastewater treatment system. Sludge is hauled off site. Package plant is sized for 1,500 gallons per day.
Water Discharge	Blow-down is treated for pH, chloride and metals. Evaluation of other constituents is needed to determine if additional treatment is required before discharge.

In-plant noise levels must not exceed 90 dBA for an 8-hour exposure. Plant perimeter noise levels must not exceed 65 dBA during the day and 55 dBA at night.

Design Data

Coal Specification

The fuel delivered by rail is Wyoming PRB subbituminous coal, with characteristics detailed in Table 2-4.

Table 2-4
Coal and Ash Analysis for Wyoming Subbituminous Coal

Proximate Analysis Weight, % As Received	
Moisture	30.24
Ash	5.32
Volatile	31.39
Fixed Carbon	33.05
Ultimate Analysis Weight, % As Received	
Carbon	48.18
Hydrogen	3.31
Nitrogen	0.70
Chlorine	0.01
Sulfur	0.37
Oxygen	11.87
Ash	5.32
Moisture	30.24
Heating Value, As Received	
HHV, kJ/kg (Btu/lb)	19,400 (8340)
LHV, kJ/kg (Btu/lb)	17,900 (7710)
Ash Softening Temperature for Reducing Conditions	
°C (°F)	1190 (2170)
Ash Mineral Analysis Weight %	
SiO	31.38
Al_2O_3	15.12
TiO_2	1.11
Fe_2O_3	5.36
CaO	23.56
MgO	4.68
Na_2O	1.48
K_2O	0.31
P_2O_5	0.86
SO_3	14.67
Undetermined	1.46

Limestone and Reagents

The nominal design composition of the limestone to be used in the FGD is shown in Table 2-5.

Table 2-5
Limestone Design Composition

Design Limestone Analysis, wt. % (as received)	
$CaCO_3$	94.1
$MgCO_3$	3.3
SiO_2	0.6
Water	2.0
Balance	0.0
Total	100

Water Source and Wastewater

The raw water for the cooling tower makeup is from Lake Michigan with a typical quality shown in Table 2-6. The distance from the plant to the raw water source is assumed to be 0.8 km (0.5 mile.

Table 2-6
Raw Water Quality for Cooling Tower Makeup

	mg/l	**mg/l $CaCO_3$**
Silica (SiO_2)	6.8	—
Calcium (Ca)	76.0	189.0
Magnesium (Mg)	16.0	66.0
Sodium (Na)	20.0	44.0
Potassium (K)	2.9	3.7
Bicarbonate (HCO_3)	246.0	202.0
Sulfate (SO_4)	56.0	58
Chloride (Cl)	26.0	37.0
Nitrate (NO_3)	6.9	5.6
Total dissolved solids	457.0	—
Total hardness	—	255.0
pH	8.0	
Ionic strength (meg/l)	9.2×10^{-3}	
Temperature range, °C (°F)	5–27 (40–80)	

Potable water and makeup water for process and de-ionized water for the plant facility is drawn from municipal sources. The distance from the plant to the tie-in point is assumed less than 1.5 km (1.0 mile). The municipal water quality is assumed to be the same as the raw water presented above.

Plant Operating Criteria

Duty

The plant operates primarily as a base-load unit, with infrequent stop-start cycles. It is designed with reliability features to minimize forced outages and achieve high availability.

Capacity Factor

The PC plants have an annual capacity factor of 80%. Annual capacity factor is defined as the actual annual production divided by the plant rated capacity times 8760 hours.

Plant Design Life

The plant design is based on using components suitable for a 30-year life, with provision for periodic maintenance and replacement of critical parts.

Mechanical Design Criteria

This section establishes the criteria for the mechanical design of the PC power plant components and systems.

Boiler Description

The Boiler Island scope and general design basis are summarized below.

- Greenfield, balanced-draft unit fired with subbituminous coal, and designed for base-loaded operation with 30-year life.
- The gross power of the steam turbine is 825 MW without CCS.
- Low-NO_X axial-swirl burners with over-fired air and SCR are used to achieve emission limits of 30 mg/m^3 (0.03 lb/MBtu).
- The steam conditions entering the HP turbine are 242.3 bar (3515 psia) and 732°C (1350°F). The reheat temperature is 760°C (1400°F).
- The steam piping pressure drop and attemperation assumptions are shown in Table 2-7.

Table 2-7
Steam Piping Pressure Drop and Attemperation Assumptions

Feedwater heater outlet to the boiler inlet	0.5%
Through the economizer, boiler and superheater	8%
Main steam from the boiler outlet to the turbine inlet	6%
Cold reheat, exiting the turbine to the boiler inlet	2%
Reheat steam through the boiler	4%
Hot reheat steam from the boiler outlet to the turbine	2%
Loss from the IP turbine to the LP turbine	<1%
Superheater uses two-stage attemperation, % of BFW used for attemperation	4%

- There is a separator, recirculation pump and start-up system, and economizer.
- A bottom-ash system (submerged chain conveyor) to remove ash from the hopper throat feeding it into a water-filled trough.
- Soot-blowing system and mechanical draft cooling tower.
- Single fans for FD and PA.
- Seven operating coal mills are installed at the side of the furnace.
- The main design performance data of the A-USC boiler are summarized in Table 2-8. Selection of main and reheat steam pressure and temperature are described in Section 6.

Table 2-8Table 2-8
Summary of Boiler Performance

	Main Steam
Flow, kg/h x 10^3 (lb/h x 10^3)	1818 (4008)
Temperature leaving boiler °C (°F)	733 (1352)
Pressure leaving boiler, bar (psia)	347 (3585)
	Reheat Steam
Flow, kg/h x 10^3 (lb/h x 10^3)	1563 (3375)
Temperature leaving boiler °C (°F)	760 (1400)
Pressure leaving boiler, bar (psia)	51 (740)
	Feedwater
Temperature entering boiler °C (°F)	305 (580)
Pressure entering boiler, bar (psia)	273 (3865)
	Heat Duties
Heat to Main Steam, MW, (MBtu/hr)	1293 (4412)
Heat to Reheat Steam, MW (MBtu/hr)	285 (972)
Total Heat to Steam, MW (MBtu/hr)	1567 (5348)

Boiler Efficiency, %	86.6
Heat In Fuel, HHV, MW (MBtu/hr)	1822 (6218)
Fuel Fired, kg/hr (lb/hr)	338,000 (746,000)
CO_2 Emission without PCC, kg/hr (lb/hr)	596,000 (1,313,000)

Figure 2-1 and Figure 2-2 are diagrams depicting the hot flue gas and steam/water flows, temperatures, and pressures calculated by the AspenPlus™ model of the steam generator in U.S. Customary and SI units, respectively. Figure 2-3 presents the general process flow diagram of the boiler island. Appendix A contains the heat and mass balance information for Case 1 and Case 2.

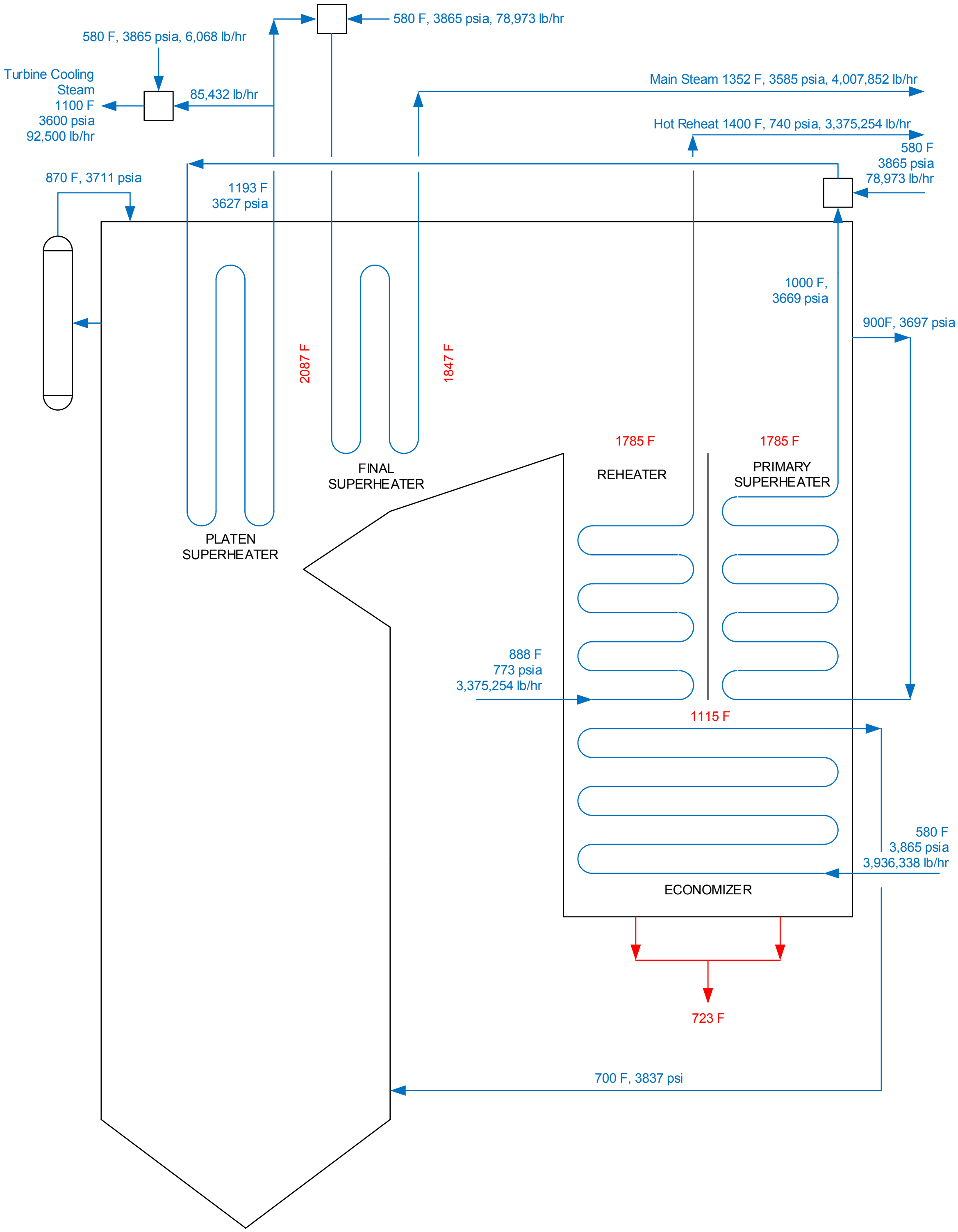

Figure 2-1
Flows, Temperatures, and Pressures through the Boiler (U.S. Customary Units)

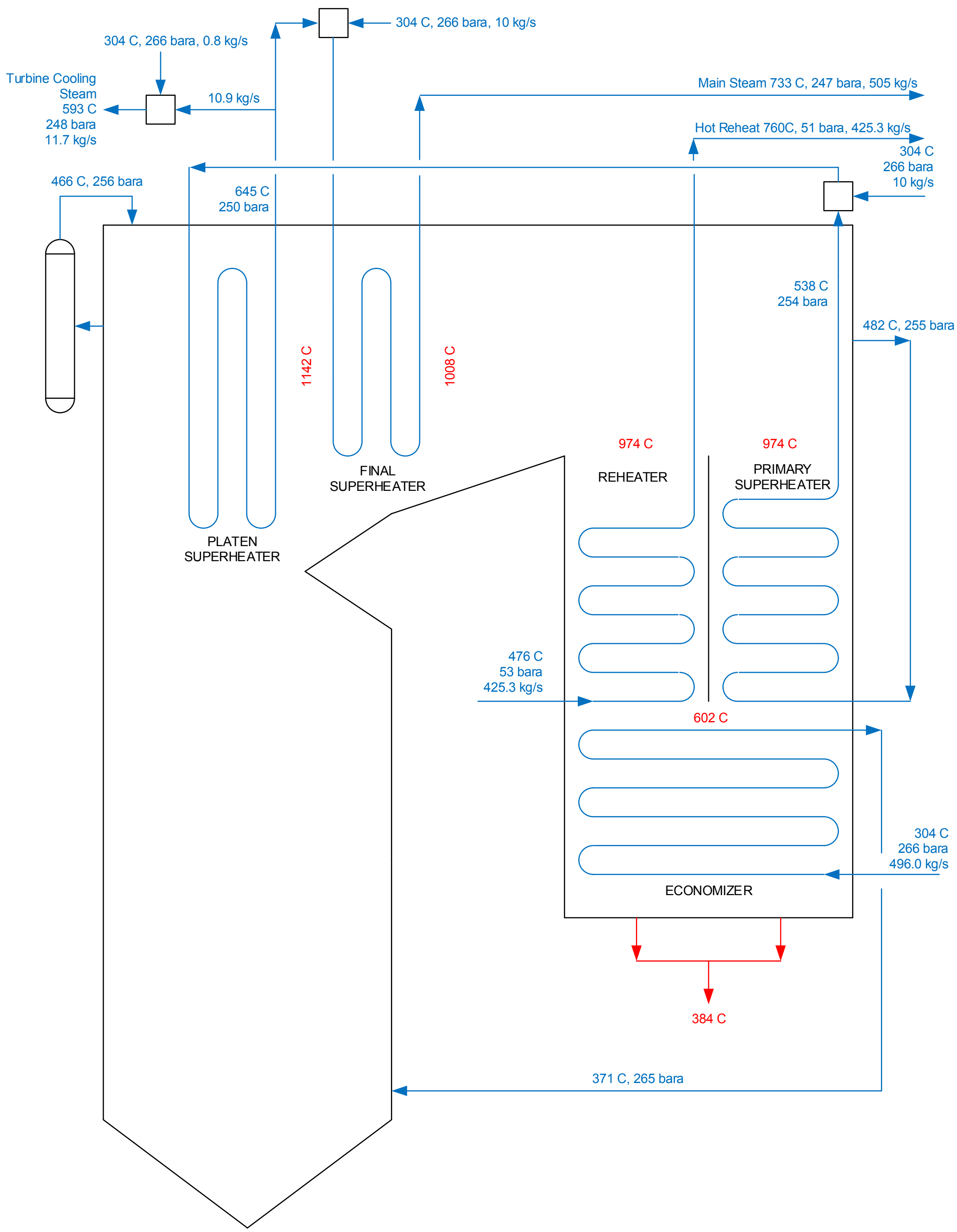

Figure 2-2
Flows, Temperatures, and Pressures through the Boiler (SI Units)

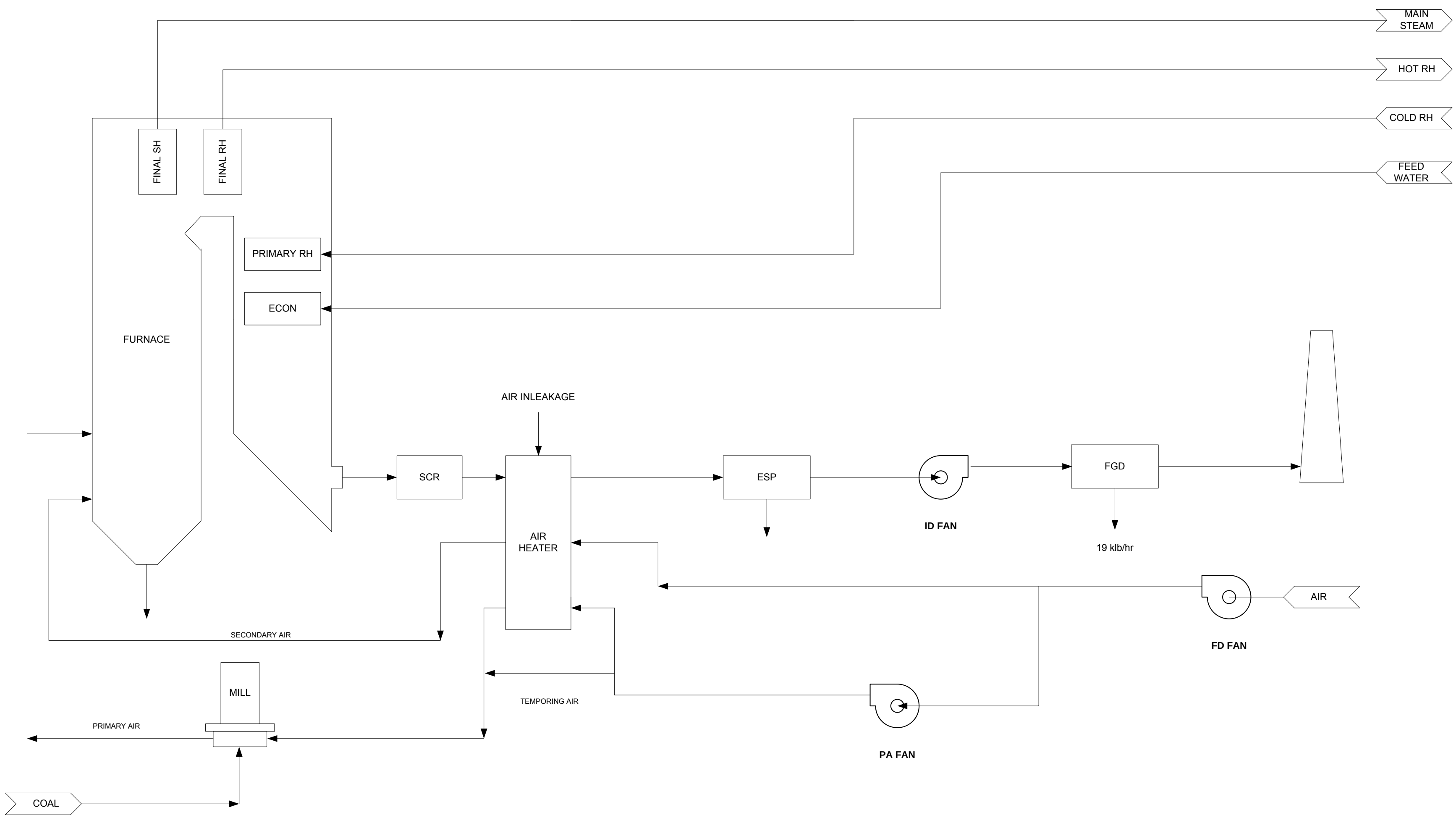

Figure 2-3
Boiler Island Process Flow Diagram

FD and PA Fans

One axial-flow FD fan with variable-pitch blade control provides the combustion air. The air is drawn from within the boiler house to make use of the heat lost from the boiler surfaces and to assist with ventilation and cooling. The fan is selected with sufficient margins on volume and pressure to meet all specified modes of operation.

Likewise, one axial-flow PA fan with variable-pitch, blade control draws air from the FD fan discharge to dry the coal in the mills and convey it to the burners. The fan is selected with sufficient margins on volume and static head to meet all specified modes of operation.

Air from the FD fan is split into the primary and secondary air streams, both of which are preheated in the regenerative air heater before passing to the furnace. A booster fan passes the PA to the milling plant to dry the coal before it is pulverized and conveyed to the burners for combustion.

Heat released in the furnace is transferred to the steam-water circuits providing evaporative, superheat, and reheat duty. Exhaust gases from the economizer pass through the SCR unit and enter the regenerative air heater, where the final extraction of heat from the flue gas takes place to preheat the primary and secondary air streams. The ESP downstream of the air heater removes fly ash entrained in the flue gas, followed by an FGD plant that removes the majority of the SO_2. The scrubbed flue gas in the non-PCC design (Case 1) is discharged to the atmosphere via the stack.

Air Heater

The design for PRB coal targets an outlet flue gas temperature of approximately 130°C (265°F). It is assumed in this model that air in-leakage of 10% occurs in the air heater.

Coal Mills (Pulverizers)

The "milling plant" comprises the vertical mills together with associated bunkers, coal feeders, outlet chutes, PC pipe work, and seal air fans. Part load is achieved by turning down the mills and shutting off burners. One mill supplies all the burners at a single elevation (front and rear wall), with each burner supplied by its own PC pipe from the mill outlet. The mill inlet air temperature is controlled by air bypassing the regenerative air-heater as shown in Figure 2-3.

Condenser and Cooling System

Wet mechanical draft cooling towers are used to provide cooling water for the condenser and other heat loads. The cooling water design conditions are shown in Table 2-9:

Table 2-9
Cooling Tower Design Conditions

	Max Summer 35°C (95°F)	Avg. Ambient 16°C (60°F)
Maximum Supply Temperature, °C (°F)	29.4 (85)	22.2 (72)
Maximum Return Temperature, °C (°F)	38.8 (100)	As calculated
Minimum Supply Pressure, bara (psia)	3.8 (55)	3.8 (55)
Return Pressure Drop, bara (psi)	2.75 (40)*	2.75 (40)*

(*) Minimum return pressure at grade at the cooling tower is 1.5 bar, a (30 psia). Value shown represents minimum at PCC battery limit which included 0.7 bar (10 psi) return header pressure drop allowance.

Air Quality Control System

The AQCS design is developed to achieve the emission levels tabulated below.

Pollutant	Emission Levels mg/m^3 (lb/MBtu)	Emission Intensity kg/MWh (lb/MWh)
PM_{10}	10 (0.010)	0.038 (0.08)
$PM_{2.5}$	13 (0.013)	0.045 (0.10)
SO_2	30 (0.03)	0.11 (0.24)
NO_X	30 (0.03)	0.11 (0.24)

The AQCS technologies used are summarized below.

NO_X control	SCR (included in boiler island)
Particulate removal, PM_{10}	ESP
SO_2 removal	Wet FGD
Mercury capture	Halogen injection into boiler promoting mercury oxidation over SCR catalyst with co-capture in FGD. Possible supplemental capture using activated carbon injection ahead of ESP.

Condensable particulate emissions, $PM_{2.5}$, are primarily SO_3, and are calculated based on the system chemistry. For the PRB coal, the uncontrolled $PM_{2.5}$ emissions are estimated to be 18 mg/m^3 (0.018 lb/MBtu), and a wet FGD typically achieves 30% removal, reducing $PM_{2.5}$ emissions to 13 mg/m^3 (0.013 lb/MBtu).

SCR System

The SCR is designed to reduce NO_X to 30 mg/m^3 (0.03 lb/MBtu) at full load, and 25% Benson load operating conditions. A single SCR reactor is mounted between the economizer outlet and the regenerative air heater gas inlet. The reactor is orientated for downward flue gas flow and incorporates two catalyst layers for initial loading, with space for the addition of a further layer to facilitate catalyst management.

The reagent is liquid anhydrous ammonia pumped from storage tanks to 2 x 100% vaporizers, and then injected into dilution air from dedicated fans.

Mercury Control

Because oxidized mercury (Hg^{2+}) is soluble, a large percentage of it is absorbed in the wet FGD. Conversely, because elemental mercury (Hg^0) is insoluble, the wet FGD does not capture it effectively. The oxidation state of the mercury entering the FGD has been found to be strongly dependent on the coal chemistry and system configuration.

SCR catalysts are designed to reduce NO_X but also have the dual function of oxidizing mercury. This can be a significant co-benefit because SCR installed upstream of an FGD can result in a higher overall mercury capture. Halogens promote the oxidation reaction and capture of mercury in the FGD.

To compensate for the low halogens in subbituminous coal, bromine can be introduced directly into the furnace or into the flue gas stream to promote oxidation and enhance mercury removal in the FGD.

Re-emission of up to 15% of the mercury captured in the FGD has been observed, but this can be nearly eliminated by the use of suitable additives, as included in the current design.

If 90% mercury capture cannot be achieved by the above steps, sorbent injection ahead of the ESP can be utilized as a polishing stage. Only small quantities of sorbent are required for this purpose. However, the fly ash collected by the ESP now contains captured mercury, and EPA rulings state that this material can no longer be used in cement kilns due to the re-emission of the captured mercury.

Electrostatic Precipitators

The ESP components included in the design are:

- Gas distribution devices
- Collecting system
- Collecting plate rappers
- High-voltage system
- Discharge electrodes
- Insulator compartments
- Discharge electrodes rappers
- Control system

FGD System

The limestone slurry contacts the flue gas in the FGD absorber to remove the SO_2. A generic spray tower design is used, although several alternative designs are available. Gypsum generated in the process can easily be disposed of on site.

The flue gas exits the ID fans and is ducted to the vertical up-flow spray tower absorber. The hot flue gas enters the lower portion of the tower through the inlet nozzle, turns upward, and flows counter-currently through multiple levels of spray headers and associated nozzles. The scrubbing slurry cools the flue gas to the adiabatic saturation temperature at which the SO_2 is absorbed.

The cleaned flue gas enters the vertical flow, two-stage mist eliminator section located in the absorber flue gas exit, which removes carryover mist by inertial contact. The mist eliminator consists of a primary stage for capturing large particles and a secondary stage for capturing wash water droplets and finer particles. Both stages are kept free of slurry deposits by a water-wash spray system directed to the upstream and downstream faces of the first stage and the upstream face of the second stage. The cleaned flue gas then exits the absorber and enters the chimney designed for wet operation.

The slurry falls into the reaction tank located at the base of the tower, where injected air completes the scrubbing reaction by oxidizing the calcium sulfite into calcium sulfate (gypsum). Agitators in the tank ensure the air and slurry are well mixed and that the solids remain in suspension.

Recycle pumps taking suction from the reaction tank pump the slurry to the headers and nozzles in the spray tower for further reaction with the flue gas. The absorber has a spare recycle pump and associated header. This can be brought into service to increase SO_2 capture if so required.

Limestone slurry from the Limestone Preparation System is added to the reaction tank to neutralize and regenerate the scrubbing slurry. A bleed system removes the appropriate amount of slurry from the reaction tank to maintain process equilibrium. The absorber has two bleed pumps (one operating and one spare) to transfer slurry to the gypsum dewatering and storage facility.

Limestone Handling

Limestone for the FGD system is received by trucks that unload directly onto the uncovered storage pile. The limestone is moved by dozer to the platform reclaim conveyor that transports the limestone to the crusher. A series of conveyors delivers the crushed limestone to one of two storage silos. Dust collection devices are installed in the vicinity of the crusher, transfer locations, and on the limestone silos.

Limestone Preparation

Belt feeders deliver the crushed limestone from the storage silos to 2 x 100% capacity horizontal wet, ball-mill grinding systems, one in operation and on standby. Before entering the mills the limestone is mixed with process recycle water. The ground limestone passes into the mill slurry tank and exits at the opposite end of the mill to be pumped to hydrocyclones that remove the coarse limestone. The underflow, containing the larger particles enters a distributor box and flows by gravity to the ball mill to be reground. The overflow containing the fine slurry with 30% solids passes to a distributor box and flows by gravity into the reagent storage tank. Slurry pumps transport the limestone slurry to the absorber reaction tank based on a demand signal from the absorber.

Gypsum Dewatering

The solid waste handling system is designed to accommodate the collection of FGD byproduct gypsum from the boiler and transport it off-site for disposal. It is assumed that there is no resale value for the FGD byproduct. The system receives slurry containing approximately 15% solids from the FGD system absorber bleed pump and produces a gypsum product containing 85% solids.

The bleed slurry is pumped in a continuous loop to the hydrocyclones located adjacent to the absorber. An on/off valve, controlled by density, admits slurry to the hydrocyclones for initial separation of the gypsum from the water. The underflow containing 50% solids flows by gravity to a tank located adjacent to the absorber and is then pumped to a surge tank located in the gypsum dewatering area. To prevent solids settling, all tanks are furnished with an agitator. The partially dewatered slurry is pumped to the vacuum belt filters to dewater the gypsum, which is discharged onto a transport conveyor and sent to either of two uncovered storage piles near the dewatering building. The filtrate is pumped to the filtrate tank and returned to the absorber and reagent preparation systems for process use.

Balance of Plant

Raw Coal Receiving, Storage, and Handling

This system unloads, conveys, prepares, and stores the coal delivered to the plant. The scope of the system is from the rotary car dumper and coal receiving hoppers to the pulverizer fuel inlets.

The PRB coal is delivered to the site by unit trains of 100-ton rail cars. Each unit train consists of 100, 100-ton rail cars. The unloading is by a rotary dumper, which unloads the coal to two receiving hoppers. Coal from each hopper is fed directly into a vibratory feeder. The 150 x 0 mm (6 x 0 inch) coal from the feeder is discharged onto a belt conveyor. The coal is then transferred to a second conveyor that transfers the coal to the reclaim area. The conveyor passes under a magnetic plate separator to remove tramp iron, and then to the reclaim pile.

Coal from the reclaim pile is fed by two vibratory feeders, located under the pile, onto a belt conveyor that transfers the coal to the coal surge bin located in the crusher tower. The coal is reduced in size to 75 x 0 mm (3 x 0 inch) by the first of two coal crushers. The coal then enters a second crusher that reduces the coal size to 25 x 0 mm (1 x 0 inch). The coal is transferred by conveyor to the transfer tower. In the transfer tower the coal is routed to the tripper, which loads the coal into one of the coal silos.

Solid Waste Handling Systems

Fly Ash Handling

This system is designed to collect fly ash from the ESP and transport it for disposal. On-site emergency ash storage is sized for 90 days. The final disposal is off site.

Other Ash Handling

The bottom ash handling system is designed to collect ash from the boiler and transport it for disposal.

The economizer ash handling system is designed to collect the ash from the economizer and transport it for disposal.

On-site emergency ash storage is sized for 90 days. The final disposal is off site.

Control Systems

Each of the systems comprising the A-USC PC power plant contains the instrumentation necessary to monitor the critical operating parameters. Each of the systems is controlled by a digital control system (DCS) that is integrated with an overall plant DCS to result in efficient and safe operation of the plant processes.

Electrical System

This system consists of the equipment necessary to receive and distribute power to the electrically driven components and building support systems in the PC power plant. It includes station service equipment, switchgear and motor control centers, conduit and cable trays, wire and cables, protective equipment (grounding, cathodic protection, etc.), and standby equipment.

Project Transportation Size Limitations

Overland Transportation Size

The maximum overland transportable dimension is 35 m (100 ft) long by 4.5 m (15 ft) wide by 4.5 m (15 ft) height (including carriage height). Maximum equipment height is 4.1 m (13.5 ft) assuming using 0.45 m (1.5 ft) height low-boy carriage. Maximum overland transportable weight is 109 tonnes (120 tons).

Barge Transportation Size

Although there is no maximum barge transportation size limitation, the project site is 10 km (6 miles) inland from Lake Michigan and thus overland transportation from the dock to job site is required. Maximum permissible barge transportation size may be restricted and will need to be reviewed on a case-by-case basis.

3
MODELING METHODOLOGY

The modeling for this project utilizes several resources:

- EPRI's PC Cost is used to size and cost the PC boiler system.
- AspenTech's AspenPlus^{TM} is used to perform the heat and material balance for the PC boiler system.
- General Electric's in-house software is used for the steam turbine cycle and design.
- The design of the CO_2 capture system in Case 2 is based on EPRI report 3002003740.[17]

Boiler – PC Cost Model

PC Cost is an Excel based spreadsheet costing tool developed by EPRI and its subcontractors that "evolved" for ~25 years. Its purpose is to allow engineers and planners to estimate the conceptual and preliminary costs of subcritical and supercritical PC power plants.[18] It performs heat and material balance calculations and performance calculations for the PC unit based on the fuel specification. It then estimates the cost of the major equipment subsystems by scaling from reference costs. These reference costs are based on budgetary quotes and/or in-house developed quotations for the boiler, turbine, air quality control systems (AQCS), and material handling equipment. These reference costs have been updated periodically over the life of the costing tool. The goal is to provide a ±30% cost estimate for the equipment and materials within the plant boundary, including all the direct and indirect costs for site preparation, earthwork, concrete and structural steel, building construction, major equipment, auxiliary equipment, piping, electrical/instrumentation/control equipment, construction labor, bulk materials, and subcontractors.

Sizing of the PC furnace is performed using proprietary methods. PC Cost then sizes the backpass of the boiler using conventional heat transfer methods to determine the surface area. The temperature and pressure of the steam/water is used to determine the wall thickness of the tubes and tube weights. The cost is determined from unit costs for the boiler components.

For the other plant equipment, the costs are scaled from reference plant costs using industry-accepted algorithms.

Modifications to PC Cost

In 2011, EPRI began developing a version of PC Cost to allow for the evaluation of A-USC PC units. Several modifications were implemented during this transition:

- It was decided to create a version of PC Cost that used AspenTech's AspenPlus^{TM} to perform the majority of the heat and material balance calculations.

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[17] *EPRI's Best Integrated Coal Plant with Post-Combustion Capture Case Study*. EPRI, Palo Alto, CA: 2015. 3002003740.

[18] Hoskins, Bill (URS) and Booras, George (EPRI), "Assessing the Cost of New Coal-Fired Power Plants," *Power Magazine*, October 2005, pp 24-28.

- To determine the cost of the boiler, PC Cost has to first calculate the weight of the individual boiler components (superheater, reheater, etc.). Using the operating conditions, PC Cost calculates a tube metal temperature and then based on an assumed tubing material calculates a wall thickness. To accommodate A-USC boilers, a wider range of alloys were included including nickel based alloys.
- In the original PC Cost, the individual sections of the boiler are made from only one material. A modification was made to allow materials to be chosen based on the estimated metal temperature of an individual row of tubes (perpendicular to the flue gas flow). This approach is described in more detail below.

AspenPlus™ HMB Model

PC Cost assumes a boiler configuration to perform its heat transfer calculations. This configuration and the corresponding AspenPlus™ model are shown in Figure 3-1. For both cases, the steam turbine heat balance was developed by GE and the steam/water flows were used as inputs to the PC Cost model. A temperature profile for the steam/water is assumed through the boiler backpass, and the enthalpy rise across each section is calculated and summed to estimate the coal and air flows. AspenPlus™ then calculates the heat and material balance and the flow rates, temperatures, pressures.

One key parameter for sizing the heat transfer surface in the backpass of the boiler is the temperature exiting the furnace. PC Cost uses proprietary algorithms for determining this temperature based on coal ash analysis and correlations for estimating the slagging properties of the coal as well as its corrosion and erosion potential.

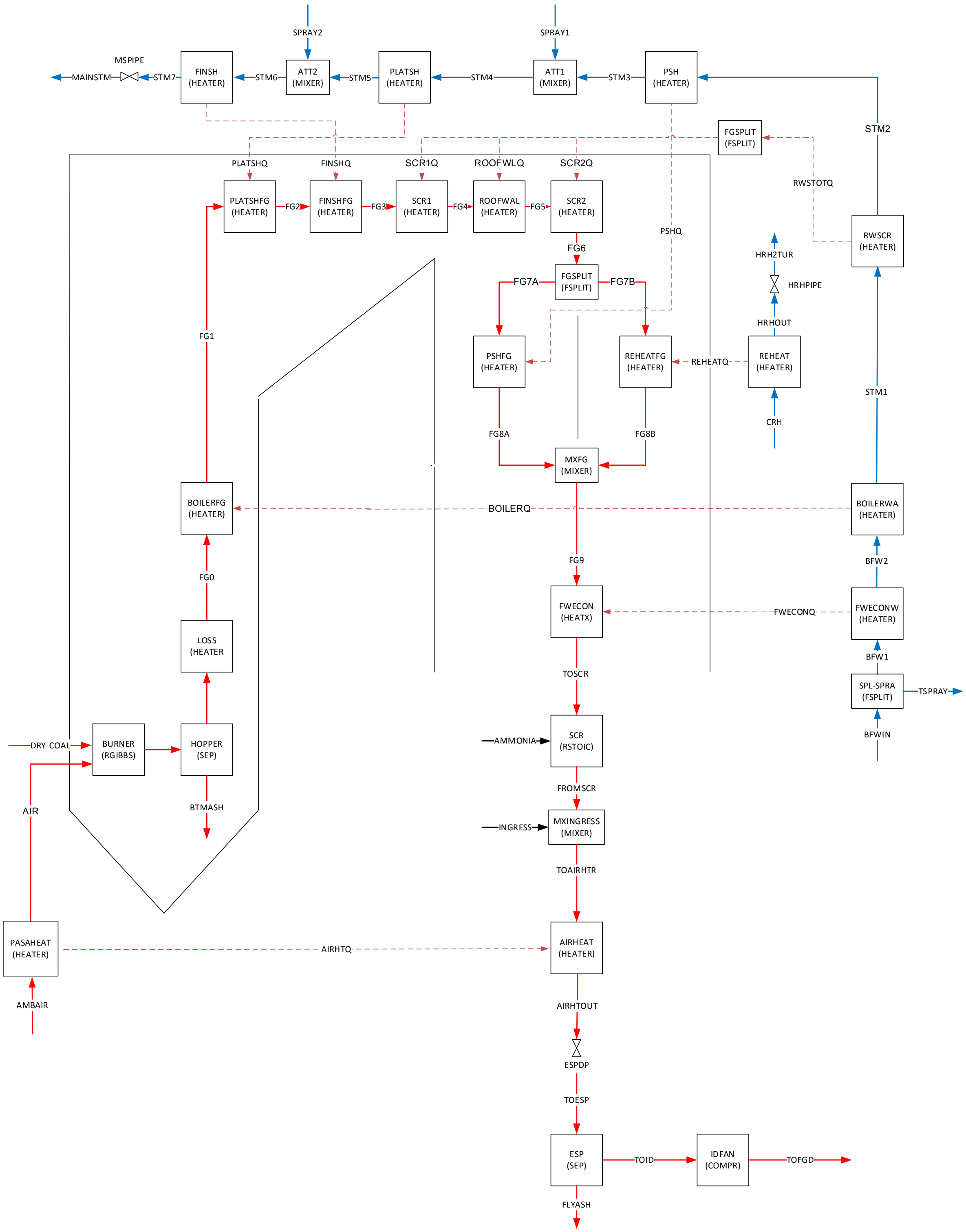

Figure 3-1
Original PC Cost Boiler Configuration

Heat Transfer Calculations

The assumed temperature profile of the steam/water circuit through the boiler set the enthalpy of the fluid at the inlet and outlet of each boiler section. Combined with the flow rate of the fluid, this determined the overall duty for each section of the boiler, and is used to determine the temperature of the flue gas.

AspenPlus™ provided values for the physical properties at the inlet and outlet of each section for the steam and the flue gas, and these values are averaged to calculate the inside and outside heat transfer coefficients. Once the individual heat transfer coefficients are calculated, the overall heat transfer coefficient and the log-mean temperature difference are calculated using standard heat transfer methods. Using this information, the area of each boiler section can be calculated.

PC Cost uses proprietary calculations based on the coal ash properties and the heat input to the furnace to determine the furnace dimensions (depth, width and height). This set the width of the convective pass. The height of the convective pass is set by the flue gas conditions and ash properties entering the finishing superheater (FSH).

Selection of Boiler Materials

In the original version of PC Cost only two materials are used for the boiler sections—carbon steel and T91. In the A-USC version of PC Cost, the following materials were added to the program to accommodate the higher steam temperatures:

- Austenitic stainless steels: Super 304H, HR3C, and HR6W
- Nickel alloys: IN617, HR230, and IN740

For A-USC cases, PC Cost was modified so that material selection within a boiler section is based on local design conditions, hence allowing the most appropriate material to be selected and minimize costs. The following approach is used for each section of the boiler:

- From the heat transfer calculations for a section, the total surface area, the tube OD, the tube spacing, and the convective section height and width are determined.
- Based on the total surface area and the tube OD, the total required length of tubing can be calculated.
- Using the height of the convective section as the length of a single tube, the number of tubes in a section can be calculated.
- The tube center-to-center spacing of the convective section is based on the design flue gas velocity. Using this spacing, the number of tubes in a row perpendicular to the flue gas flow was calculated. Each of these rows represents a "platen."
- From the total number of tubes calculated and the number of platens, the number of tubes in each platen parallel to the flue gas flow was calculated.
- To simplify the calculations, it was assumed that the steam temperature increase and flue gas temperature decrease are linear as they pass through the convective section. It was also assumed that the temperature distribution was uniform across the width of the convective section (perpendicular to the gas flow). For each row, the inlet and outlet flue gas temperatures were used to calculate an average flue gas temperature. Likewise, the inlet and outlet steam temperatures were used to calculate an average steam temperature for each row.

These values as well as the heat transfer coefficients were used to calculate a tube wall metal temperature.

- A temperature margin of 28°C (50°F) was added to this metal temperature to set the design metal temperature for the row being evaluated. For the design pressure, the pressure of the steam entering this section of the boiler is multiplied by 1.25.
- For each of the alloys being evaluated the program calculated the allowable stress at the design temperature based on the values in the ASME Boiler and Pressure Vessel code. If the design temperature is greater than the temperature allowed by Code for a given material, that material is removed from consideration for that row of tubes.
- A tube OD is assumed for each section of the backpass, and a tube wall thickness calculated based on the allowable stress. If the calculated tube wall thickness exceeded 20% of the OD, this material was allowed for this row of tubes. This criterion was set to maintain a reasonable pressure drop.
- From the tube length, diameter, and wall thickness, the weight of a single tube can be calculated. This is multiplied by the material costs utilized for this study.

The unit weight cost of the materials meeting the code design criteria was multiplied by the tube weight for a single tube, and the lowest cost material was chosen. It is assumed that this material was used for all of the tubes in a row perpendicular to the flue gas flow.

Steam Tubing and Piping

The material selection method for the steam piping was similar to that of the tubing. First, PC Cost assumed a velocity of 61 m/s (200 ft/sec) for the steam in the piping. Based on the volumetric flow rate of the steam and this assumed velocity, the inside cross-sectional area and the pipe ID are calculated.

The pipe wall thickness is calculated using an approach similar to the tubing approach discussed above. If the wall thickness for a selected material is greater than 20% of the tube OD, then the material is not allowed for the pipe. The same material unit weight costs are used to calculate the cost per meter of pipe. The material with the lowest cost is used for the design.

Auxiliary Loads

For the fans and pumps, the motor power is calculated from AspenPlus™ based on the flow rate calculated in the Heat and Material Balance (HMB). Values in the Boiler Island not calculated by AspenPlus™ (mill power, for example) are ratioed from values reported in EPRI report 1015699[19] based on the calculated flow rates.

Pump load and condenser duties for the steam turbine and feedwater system are calculated by AspenPlus™.

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[19] *Engineering and Economic Evaluation of 1300°F Series Ultra-Supercritical Pulverized Coal Power Plants: Phase 1*. EPRI, Palo Alto, CA: 2008. 1015699.

Steam Cycle Modeling

The steam cycle design and corresponding steam turbine configuration were modeled entirely through the use of General Electric's internally developed turbine design and thermodynamic heat balance modeling tools. GE steam turbine efficiencies are determined using precision field test results, laboratory analysis, and comprehensive aerodynamic modeling. Turbine efficiencies presented in this report are consistent with GE's current quoting practices and are representative of a cycle-specific turbine design. Development of the preliminary steam turbine thermodynamic design adhered to all GE design practices.

4

CASE 1: PERFORMANCE RESULTS WITHOUT CO_2 CAPTURE

Steam Turbine

One of the largest barriers to the potential market penetration of A-USC steam turbine technology is the expense associated with the nickel alloy materials necessary for such high temperature service. GE has therefore focused on developing a steam turbine design which reduces the need for high temperature material application and, subsequently, unit cost. This study assumes a tandem-compound four casing machine with a single flow high-pressure (HP) turbine section, a single flow intermediate-pressure (IP) turbine section, and a four flow two casing low-pressure (LP) turbine design. A representation of this turbine layout is shown in Figure 4-1.

Figure 4-1
Layout of HP, IP, and LP Steam Turbines

High-Pressure Turbine

The HP turbine of the evaluated A-USC steam cycle is a full arc admission single flow steam path contained within a double shell bolted horizontal joint configuration. Based on GE's coal-fired plant steam turbine design experience, the HP section steam path design is an impulse wheel and diaphragm turbine. Over half of the HP stages will require nickel alloy materials to satisfactorily operate at 732°C (1350°F) inlet temperature conditions. An extraction for a feedwater heater above reheater pressure is taken in the second half of the HP steam expansion.

To reduce the cost and manufacturing cycle of this steam turbine section, GE has applied a bolted rotor architecture of individual bucketed disks derived from GE's F-Class gas turbine, in lieu of a single rotor forging design common to most modern steam turbines. This bolted structure allows for smaller nickel alloy material rotor forgings than either a monoblock or welded rotor design. The material selected for these steam path components is Haynes 282.

The double shell architecture utilizes two HP inner shells. The inlet inner shell is made of the nickel alloy H282. The exhaust inner shell is made of a current state-of-the-art shell alloy. The use of two inner shells of different material localizes and limits the use of the expensive nickel alloy material.

High temperature steam is contained within the HP steam path through the use of an external cooling line which routes high pressure boiler steam, at a temperature lower than 593°C (1100°F) from the steam boiler to the HP inlet steam endpacking through a flow control valve configuration. At sufficient flow rates, cooling steam supplies the entirety of HP inlet seal steam. Cooling flows are expected to be less than two percent of the total boiler flow, eliminate the leakage of high temperature steam, and generate power through the entire HP section, minimizing the associated performance impact. This ensures the majority of the HP inlet endpacking remains below 593°C (1100°F), limiting high temperature alloy material to the HP steam path and initial packing rings adjacent the steam path. Endpacking rotor material with lower temperature capabilities is welded or bolted to the rest of the HP steam path.

Intermediate-Pressure Turbine

Similar in concept to the HP section, the IP turbine of the evaluated A-USC steam cycle is a single flow steam path contained within a double shell bolted horizontal joint configuration. The IP section steam path design is also an impulse wheel and diaphragm bolted rotor construction. Over half of these stages require high temperature-capable materials (H282) to satisfactorily operate at 760°C (1400°F) inlet temperature conditions. An extraction for a feedwater heater is taken halfway through the IP steam expansion.

Similar to the HP section, the double shell architecture employs two inner shells. The inlet inner shell is made of the nickel alloy H282. The exhaust inner shell is made of a current state-of-the-art alloy. The use of two inner shells of different material localizes and limits the use of the expensive nickel alloy material.

The IP exhaust pressure was selected considering steam cycle as well as turbine architecture implications. The IP exhaust pressure is 8.3 bara (120 psia). This pressure is chosen to allow for a single-flow IP steam path. At pressures lower than approximately 8.3 bara, the required active length of the last IP stage presents challenges that require a two-flow IP section design with shorter stage active lengths. This design suffers IP efficiency penalties due to the increased secondary and leakage losses of smaller stage designs and requires roughly twice the expensive high-temperature rotor material to operate at A-USC conditions. Plant value is diminished through both higher costs and poorer performance.

The IP inlet endpacking cooling strategy resembles that used in the HP inlet. High temperature steam is contained within the IP steam path through the use of a cooling line taken off the HP inlet endpacking by a flow control valve configuration. An attemperation line from the HP section exhaust is available, if necessary, to maintain temperatures below 593°C (1100°F). At sufficiently high flow rates, cooling steam supplies the entirety of IP inlet seal steam. The majority of the IP inlet endpacking remains below 593°C (1100°F), limiting nickel alloy material to the IP steam path and initial packing rings adjacent the steam path.

Low-Pressure Turbine

The four flow, two casing LP turbine applies a 1016 mm (40 inch) high efficiency last-stage bucket that leverages the latest designs used in GE combined cycle products. LP non-margin steam path stages are of traditional wheel and diaphragm impulse construction. As noted in the previous IP turbine description, steam is admitted into the LP section at higher pressures than seen in other A-USC conceptual designs. When considered alongside a 760°C IP section inlet temperature, high LP bowl temperatures require application of a welded rotor, two-casing design to avoid material temperature concerns in the first LP stage. Although costs of this design will increase over a monoblock rotor LP design, the savings in superalloy material reduction from limiting the IP section design to a single flow configuration more than offsets any increased LP section costs.

The LP section contains four or five extractions, depending on cycle inlet pressure, and exhaust conditions evaluated are chosen to reduce the exhaust losses of the applied last-stage bucket configuration.

Cycle Performance Determination

Steam cycle conditions, including reheat pressure, LP inlet pressure, and extraction pressures, were adjusted to increase performance for each cycle within the set of defined design constraints. GE steam turbine section efficiencies were determined using precision field test results, laboratory analysis, and comprehensive aerodynamic modeling. Turbine efficiencies presented in this report are consistent with GE's current quoting practices and are representative of a cycle-specific turbine design.

Table 4-1 summarizes additional steam cycle assumptions that affect modeled plant performance.

Table 4-1
Summary of Cycle Parameter Assumptions

Parameter Description	Value
Main Stop and Control Valves (MSCV) Pressure Drop	2.0%
Combined Reheat Valves (CRV) Pressure Drop	2.0%
Reheater Pressure Drop, Cold Reheat to CRV Inlet	8.0%
Flange Pressure Drop of Feedwater Heater Extractions	3.0%

Steam Cycle Main Inlet Pressure

The length of the steam turbine expansion from HP section inlet to LP section exhaust is a fundamental cycle performance parameter. It is well known that the two big drivers of Rankine cycle efficiency are main steam pressure and temperature. The current assumption for this study was the establishment of main steam temperature at 732°C (1350°F), with reheat temperature at 760°C (1400°F). Therefore, for a fixed LP exhaust condition and established main steam temperature, raising the main steam pressure was a key consideration for achieving desired objectives related to cycle efficiency and value.

Several steam turbine cycle balance of plant characteristics work against any gains achieved with higher main steam pressure. The increase in boiler feedpump power requirements associated with increased throttle pressure will substantially reduce steam cycle benefits associated with the increase in main steam pressure. An increase in inlet pressure will also reduce the HP steam path efficiency due to the lower inlet volumetric flow and resultant smaller steam path annulus. Selecting an appropriate steam turbine inlet pressure for an A-USC plant first requires balancing these factors to determine the net cycle performance impact.

To determine if increasing inlet pressure adds plant value, performance gains must be considered alongside an evaluation of the additional plant costs required to reach higher inlet pressures. Increasing steam pressure will increase the wall thickness of the boiler tubing and piping materials. At the high temperatures of A-USC conditions, the highest temperature material is Inconel 740, which cost about $100/kg ($45/lb) in mid-2015. Its use increases capital cost significantly as the pressure increases.

As part of this study, GE and EPRI evaluated three steam cycles by varying the main steam inlet pressures: 242 bar (3515 psia), 294 bar (4265 psia), and 346 bar (5015 psia). This assessment concluded that increasing the main inlet pressure above 242 bar (3515 psia) provides a diminishing return in cycle performance, as shown in Figure 4-2. The benefit to net steam turbine thermal efficiency from increasing inlet pressure from 242 bara to 294 bara is over twice that of increasing inlet pressure from 294 bara to 346 bara. Cycle efficiency values in Figure 4-2 are representative, and do not necessarily reflect final heat balance values.

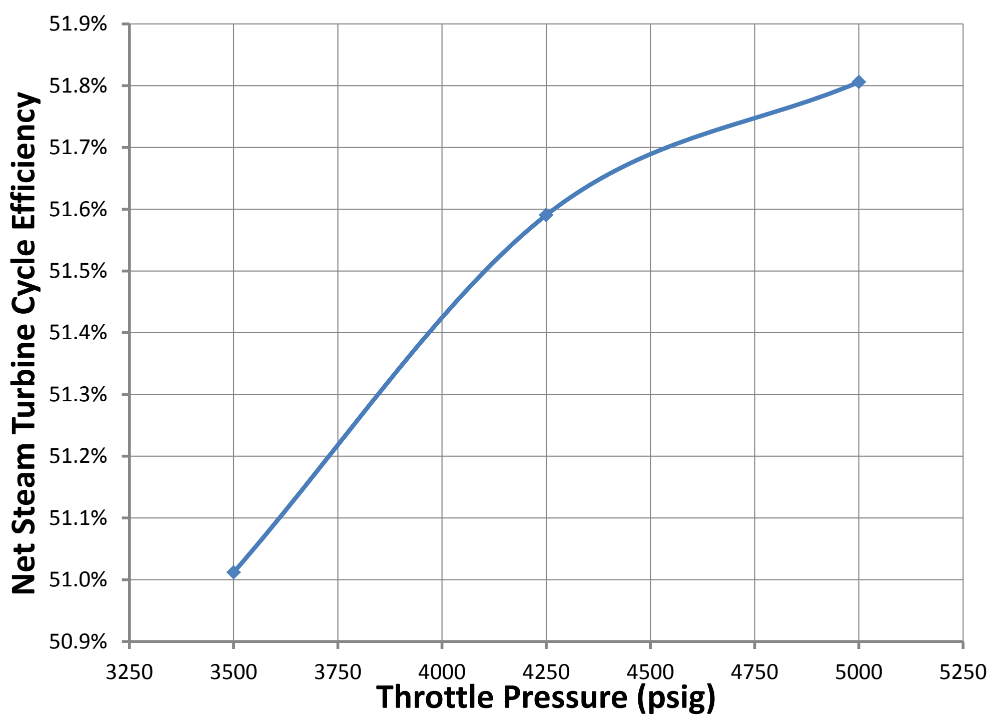

Figure 4-2
Impact of Main Steam Pressure on A-USC Cycle Efficiency

A capital cost increase for each operating pressure above 242 bar (3515 psia) was estimated, and compared to the efficiency improvement. The results are shown in Figure 4-3 and indicate that relative to a 242 bar (3515 psia) unit, the cost increase of the 294 bar (4265 psia) unit was approximately $350,000 per Btu/kWh. The cost increase of the 346 bar (5015 psia) unit was approximately $650,000 per Btu/kWh more than the 242 bar (3515 psia) case. Previous studies and recent commercial experience have shown that a 1 Btu/kWh of heat rate improvement is valued by U.S. utilities in the range of $50,000 to $150,000. As a result, these cost increases are higher than what the U.S. utility industry would traditionally pay for improvements in heat rate. Therefore, it was decided to use the 242 bar (3515 psia) steam cycle as the basis for Case 1.

The analysis showed that the main driver of the cost increases depicted in Figure 4-3 are the boiler, at 45–47%, and the main steam piping, at 17–21% of the cost increase for increased steam pressure.

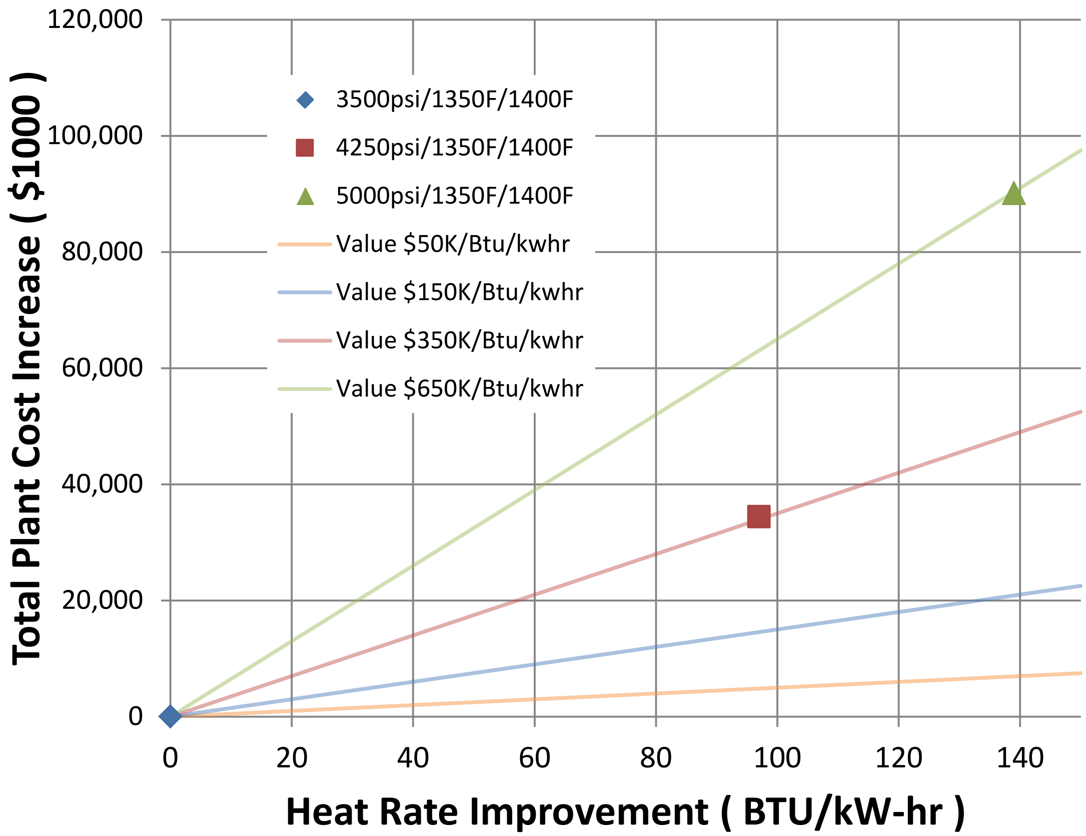

Figure 4-3
A-USC Plant Cost Increase vs. Heat Rate Improvement

This analysis is primarily based on a U.S. utility market for an A-USC PC unit with no carbon capture. In this scenario, the largest component of the LCOE is the capital cost, as shown in Figure 4-4, because a coal price of $1.80/MBtu was used for this study and there is no cost associated with CO_2 removal. For international utilities with high fuel prices, or for U.S. utilities building A-USC units with carbon capture facilities, it may be economic to increase the steam cycle pressure to increase the overall plant efficiency. Case 2 in Section 5 of this report evaluates a higher pressure cycle to determine if the higher cost is offset by the efficiency improvement.

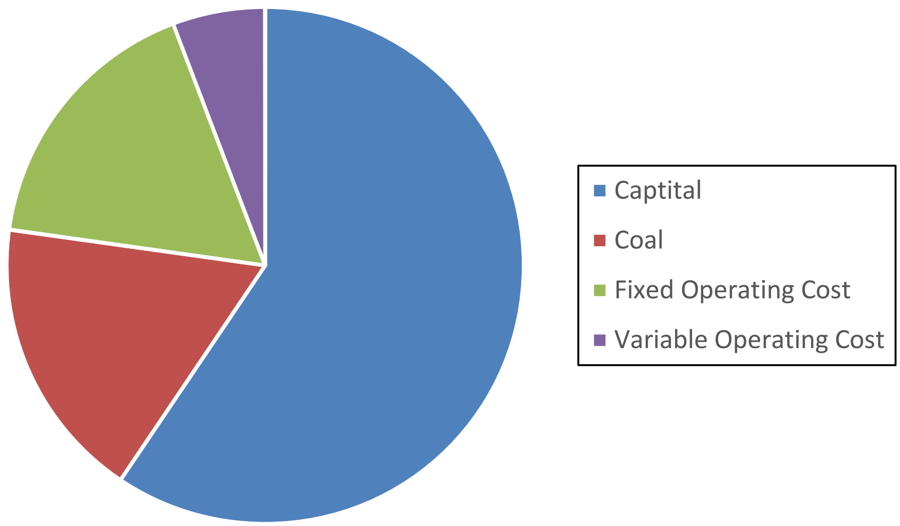

Figure 4-4
Levelized Cost of Electricity, Cost Component Distribution

Condensate System

The function of the condensate system is to pump condensate from the condenser hot well to the deaerator, through the gland steam condenser and low-pressure feedwater heaters. The system consists of the following major components:

- Water cooled condenser
- Three 50% capacity motor-driven condensate pumps
- One 100% capacity condensate polishing system
- One gland steam condenser

The steam from the LP turbine section exhausts into parallel-string condensers. The steam is condensed with cooling water at a maximum temperature of 29.4°C (85°F), and the condenser is designed to operate at 6.7 kPa (2 in-Hg).

The main condensate pumps are correspondingly divided into two sets. The first-stage set overcomes the pressure drop of the condensate polishing plant and the gland steam condenser. The second-stage set includes a booster pump set to overcome the relatively high pressure of the deaerator. A minimum flow recirculation line, discharging to the condenser, is provided to maintain the minimum flow requirement for the gland steam condenser and condensate pumps. A 100% in-line condensate polisher is used at all times to maintain the condensate quality required by the boiler.

Feedwater Heaters

The boiler feedwater heaters consist of LP and HP heater trains. Selected design parameters are presented in Table 4-2, and the steam cycle diagram is shown in Figure 4-5. The main condensate LP heater train is equipped with four heaters, FWH 1 through FWH 4. The deaerator is equipped with a storage tank that provides buffering storage capacity for the boiler feedwater system. The heating steam for the deaerator is from the IP turbine exhaust extraction.

Table 4-2
Feedwater Heater Design Parameters

Name	Type	Terminal Temperature Difference	Drain Temperature Difference
LP Heater 1 - FWH 1	Shell & tube	2.2°C (4°F)	5°C (9°F)
LP Heater 2 - FWH 2	Shell & tube	2.2°C (4°F)	5°C (9°F)
LP Heater 3 - FWH 3	Shell & tube	2.2°C (4°F)	5°C (9°F)
LP Heater 4 - FWH 4	Shell & tube	2.2°C (4°F)	5°C (9°F)
Deaerator	Direct Contact	-	-
HP Heater 1 - FWH 6	Shell & tube	-1.7°C (-3°F)	5.6°C (10°F)
HP Heater 2 - FWH 7	Shell & tube	-1.7°C (-3°F)	5.6°C (10°F)
HP Heater 3 - FWH 8	Shell & tube	-1.7°C (-3°F)	5.6°C (10°F)
Topping Desuperheater	Shell & tube	-	-

The HP heater train consists of three HP feedwater heaters, FWH 6 through FWH 8, and an additional topping desuperheater using extraction from the IP turbine. Due to the relative high degree of superheating, FWH 6, FWH 7, and FWH 8 are equipped with desuperheaters to utilize the extracted steam before condensing. All three HP heaters are equipped with extraction condensate subcoolers. The first extraction on the IP turbine, shortly downstream of the hot reheat, contains a very high degree of superheat. The temperature of this steam is in excess of 611°C (1132°F), but its saturation temperature is only about 215°C (419°F). An additional desuperheater located above the top HP heater can remove most of this superheat before the steam flows to the normal feedwater heater. This configuration will increase the final feedwater temperature with a corresponding decrease in turbine cycle heat rate. This additional heater is simply a shell-and-tube heat exchanger, no different than the desuperheating zone of any other feedwater heater, but it does not contain the condensing and subcooling sections.

The overall unit heat rate decreases with increased final feedwater temperature. The selection of final feedwater temperature was based on steam cycle studies for the selected main steam conditions with consideration of boiler implications. Boiler (economizer) limits on final feedwater temperature were based on requirements on flue gas outlet temperature and the impacts on boiler efficiency.

Boiler Feedwater Pump

Two 50% capacity motor-driven boiler feedwater pumps with booster pumps and hydraulic couplings are selected for the project. Although turbine driven feedwater pumps are more conventional for large supercritical units in the United States, motor-driven feedwater pumps offer several advantages compared to turbine-driven pumps:

- No additional start-up pump required.
- Simplification of plant layout because of less scope and complexity of equipment.
- Smaller impact on the LP turbine design when a large quantity of low-pressure steam is extracted for future use in post-combustion CO_2 capture plants.
- Lower capital cost. This assumes the same size of electrical generators for both configurations. The turbine driven pump configuration may have a relatively smaller electrical generator and main transformer, which will offset the higher capital cost of the turbine.
- Lower operating and maintenance costs.

Because Case 2 incorporates a CO_2 capture plant into the design, it was decided to use a motor-driven pump.

Circulating Water System

The circulating water system consists of two 50% capacity vertical circulating water pumps, a multi-cell mechanical draft cooling tower, and carbon-steel cement-lined interconnecting piping.

Plant Performance

Based on the configuration described in this report, the performance of the power plant without CO_2 removal is presented in Table 4-3. The estimated auxiliary power loads of the plant without PCC are presented in Table 4-4.

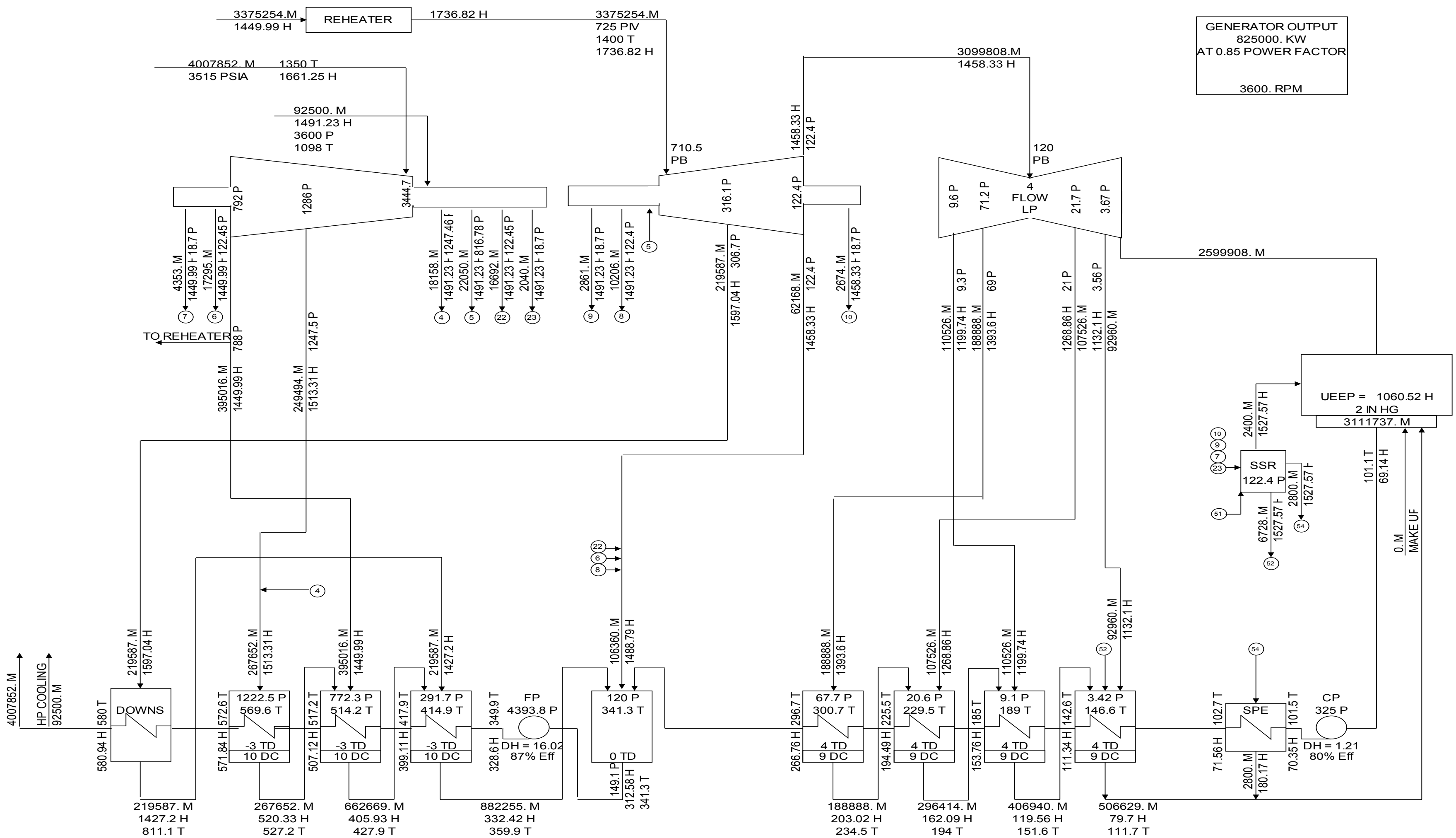

Figure 4-5
3500 psig Steam Cycle Diagram for Case 1

Table 4-3
Summary Performance of the Advanced Ultra-Supercritical Plants without PCC

Description	SI Units	U.S. Customary Units
Feedwater heaters	Five LP, deaerator, three HP, topping desuperheater	
Feed pump drive	Motor	
Cooling system	Mechanical draft cooling tower	
Emission controls	SCR, ESP, Wet FGD, and $CaBr_2$ injection into the furnace for NO_x, particulate, sulfur, and mercury control	
Throttle conditions	242 bar/732°C/760°C	3515 psia/1350°F/1400°F
Main steam flow	1,818,000 kg/h	4,008,000 lb/hr
Hot reheat flow	1,563,000 kg/h	3,375,000 lb/hr
Condenser flow	1,412,000 kg/h	3,112,000 lb/hr
Condenser pressure	6.7 kPa	2 inches mercury
Final feedwater temp	304°C	580°F
Gross plant output, kW	825,000	825,000
Auxiliary load, kW	70,801	70,801
Net plant output, kW	754,199	754,199
Net plant heat rate	8698 kJ/kWh	8244 Btu/kWh
Net plant efficiency, % (HHV)	41.4	41.4
Plant fuel consumption	338,000 kg/hr	746,000 lb/hr
CO_2 emission without PCC	596,000 kg/hr	1,313,000 lb/hr
CO_2 emission without PCC	722 kg/MWh, gross basis	1592 lb/MWh, gross basis

Table 4-4
Auxiliary Loads for the Advanced Ultra-Supercritical Plants without PCC

Equipment	**kW**
Condensate Pump	1188
Circulating Water Pumps	5262
Motor-Driven Feedwater Pumps	21,263
Pulverizers	3105
PA Fans	1740
SA/FD Fans	2587
Induced Draft Fans	10,783
Ash Handling System	1352
Coal Handling	436
Cooling Tower Fans	6364
Steam Turbine Auxiliaries	695
Electric Static Precipitator	639
Wet FGD System	7906
SCR	208
Water Treatment	496
Miscellaneous Loads (Controls, Lighting, etc.)	2980
Transfer and Cable Losses	2772
Allowance for unknowns	1024
Total Auxiliary Power Consumption	70,801

5

CASE 2: PERFORMANCE WITH CO_2 CAPTURE

Case 2 estimates the performance of the same A-USC plant described in Case 1 but incorporates the CO_2 removal necessary to meet EPA's proposed standard for CO_2 emissions of new coal-fired power plants (section 111(b) of the Clean Air Act). This standard of performance was proposed at 500 kg-CO_2/MWh (1100 lb-CO_2/MWh) on a gross power basis, and was subsequently modified in August 2015 to 630 kg-CO_2/MWh (1400 lb-CO_2/MWh) on a gross power basis. For a PC plant, EPA defines gross power as the steam turbine gross output minus the boiler feedwater pump power. Analyses were conducted using the Case 1 base plant performance values from Section 4 showing the gross steam turbine power is 825,000 kW and the boiler feedwater pump power is 21,263 kW. Thus, for Case 2, the gross power as defined by EPA would be 803,737 kW. To meet the EPA 111(b) limit as originally proposed, the unit would only be allowed to emit approximately 401,000 kg/hr (884,000 lb/hr) of CO_2. The CO_2 emitted from this plant in Case 1 was 596,000 kg/hr (1,313,000 lb/hr), so approximately one-third of the CO_2 would have to be removed from the flue gas. Under the modified EPA 111(b) limit, approximately 510,000 kg/hr (1,125,000 lb/hr) of CO_2 would be allowed, and preliminary analyses suggest only about 15% of the CO_2 would have to be removed from the flue gas.

The steam required to regenerate the CO_2 removal solvent is extracted from the LP turbine steam path. Extracting this steam at constant heat input decreases steam turbine gross output. By using significant thermal energy in the PCC system to heat the condensate instead of rejecting it to the cooling tower, less steam needs to be extracted for low pressure feedwater heating and more steam is available to generate power, tempering the output impact of solvent regeneration steam extraction. Both the steam used to regenerate the solvent and the "CO_2 removal waste heat" available are dependent on the amount of CO_2 removed and the CO_2 removed is dependent on the steam turbine gross power, resulting in an. iterative calculation. When heat integration was included, the amount of flue gas required to be treated was 34%.

The HMB for the PCC system was based on the system presented in EPRI report 1017515[20] with the exception being that the solvent used for this study was a "near-term" solvent that has been evaluated at the National Carbon Capture Center (NCCC). Due to proprietary agreements in place, the name and manufacturer of this solvent cannot be disclosed, and only minimal information about the solvent is available. Using this information, as well as some assumptions based on the performance of typical amine solvent systems, an approximate heat and material balance was developed.

Process Description

The PCC technology for this case was assumed to be a conventional two column absorber/regenerator scheme as shown in Figure 5-1. To meet the EPA's requirement, only a portion of the CO_2 had to be removed from the flue gas. For this study, this removal is

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[20] *An Engineering and Economic Assessment of Post-Combustion CO_2 Capture for 1100°F Ultra-Supercritical Pulverized Coal Power Plant Applications: Phase II Task 3 Final Report.* EPRI, Palo Alto, CA: 2010. 1017515.

accomplished by flowing a portion of the flue gas through the PCC system, while the remaining flue gas bypasses the system. For the flue gas entering the PCC system it was assumed that 90% of the CO_2 was removed. The CO_2 depleted flue gas exiting the absorber mixed with the bypassed flue gas before entering the stack. The bypass flow rate is adjusted so that the rate of CO_2 exiting the stack is 500 kg/MWh (1100 lb/MWh) on a gross output basis. In the design of an actual plant, it may be necessary to treat more of the flue gas to account for start-up, shut-down, part-load operation, etc., as the EPA emission regulation is based on an annual average.

Before entering the absorber, the flue gas stream from the boiler entered a flue gas scrubber where it was contacted with circulating, cooled water. This scrubber served several purposes:

- By cooling the flue gas, the volumetric flow of the flue gas decreased, which reduced the power requirement of the flue gas blower.
- Water was condensed from the flue gas. This minimized condensation of water in the absorber, which would dilute the solvent. The excess water would have to be removed during regeneration, which would increase the reboiler duty and decrease the steam turbine power.
- By adding a dilute caustic solution to the circulating water, sulfur and other contaminants that remain in the flue gas after the FGD could be removed.
- Any residual solids carried over from the FGD were reduced, minimizing the potential for foaming in the absorber.

The flue gas scrubber is a packed column. The flue gas enters the bottom of the column, where it is contacted with recirculated, cooled water. This water cools the flue gas to below the dew point, simultaneously condensing water from the flue gas and reducing the volumetric flow rate. As the water was condensed from the flue gas, the level in the column was controlled by purging the excess water back to the FGD or the cooling towers as makeup. The circulating water and the condensed water were warmed by the latent heat from the water condensation. This water was then cooled by the flue gas scrubber cooler before being recycled back to the top of the flue gas scrubber.

From the flue gas scrubber, the cooled gas entered the flue gas blower where the pressure of the flue gas was increased to overcome the pressure drop of the absorber and piping. The scrubbed flue gas was compressed to about 0.1 barg (1.5 psig) before it entered the bottom of the CO_2 absorber.

The flue gas was brought into contact with the advanced solvent in the absorber. The flue gas flowed upward through the column and the “lean” solvent flowed downward, removing 90% of the CO_2 from the flue gas. Good mass transfer occurred as the flue gas and solvent were brought into intimate contact in the column as they flowed through structured packing. The capture of CO_2 from the flue gas was assumed to be an exothermic process similar to amine solvents, so it was assumed that the temperature of the solvent increased as it flowed through the column. The flue gas exited the “lower” section of the column, before entering the upper “wash” section of the column, where it was brought into contact with circulating water. As the flue gas flowed through the bubble cap trays in this section of the column, the water cooled the flue gas and scrubbed it of any solvent that had escaped the lower section of the absorber. The flue gas then flowed to the stack and was combined with the bypassed flue gas flow before discharging to the atmosphere. The wash-water stream was heated by the flue gas and was cooled by the absorber wash cooler before being pumped back to the top of the CO_2 absorber. A small purge stream removed the recovered solvent and condensed water from the wash section and returned them to the absorber section of the column as makeup.

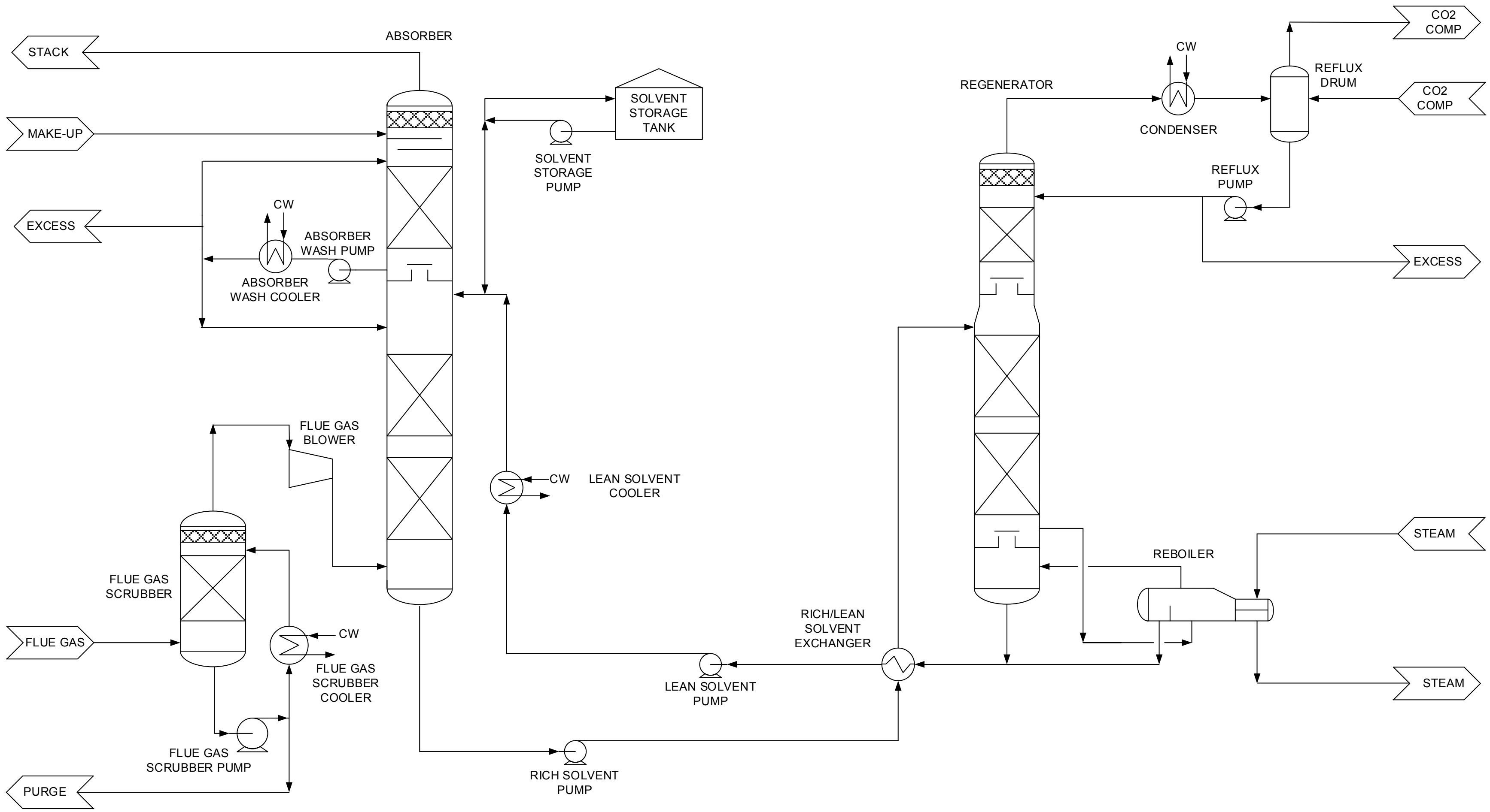

Figure 5-1
CO_2 Capture System

The CO_2 “rich” solvent exited the bottom of the absorber where it was pumped using the rich solvent pump through the plate and frame rich/lean solvent exchanger and into the regenerator. The rich solvent leaving the absorber was assumed to be 63°C (145°F), and the CO_2 “lean” solvent was assumed to exit the bottom of the regenerator at about 124°C (255°F). It was desirable to heat the rich solvent before it entered the regenerator to decrease the steam required for regeneration, and to cool the “lean” solvent as much as practical before it entered the absorber. Therefore, heat was transferred between the two streams in the rich/lean solvent exchanger.

The rich solvent entered the top of packed column regenerator where the CO_2 was removed from the solvent by the addition of heat. This heat breaks the chemical bonds, liberating the CO_2 and regenerating the solvent so that it can be returned to the absorber for further CO_2 capture. The heat was provided by the condensation of low pressure steam at approximately 2.5 bar (36 psia) in the regenerator’s kettle-type reboilers. This source of this steam was the first extraction port in the low pressure steam turbine.

The vapor leaving the regenerator was cooled to 38°C (100°F) before entering the reflux drum. The vapor exiting the reflux drum is the product CO_2, which is sent to the CO_2 compressor system, shown in Figure 5-2. Part of the condensed liquid was returned to the regenerator as reflux, and the remaining water was used as makeup for the absorber.

The “lean” solvent exited the bottom of the regenerator and flowed through the previously mentioned rich/lean solvent plate and frame exchanger where it was cooled. The cooled lean solvent was pumped using the lean solvent pumps, and then flowed through the lean solvent plate and frame cooler before returning to the absorber. The lean solvent cooler reduced the temperature of the lean solvent as much as practical to maximize the capture of CO_2 from the flue gas, minimizing the circulation rate of the solvent and decreasing the regeneration duty.

The cooled CO_2 flowed from the regenerator into the first stage CO_2 compressor knockout drum to remove any entrained water droplets before it entered the first stage of the centrifugal compressor. The CO_2 was then compressed, cooled in the intercooler, and entered another knockout drum to separate condensed water. The CO_2 then flowed through a second stage of compression, cooling, and water separation before entering the dehydration unit, where water was removed to meet CO_2 pipeline requirements. The water from the second and third stage CO_2 knockout drums flowed back to the first stage knockout drum where it was collected in a single stream and returned to the regenerator.

The dehydration unit was assumed to be a parallel-bed, molecular-sieve system. The CO_2 flowed through one bed until the molecular sieve was almost saturated with water, at which point, the beds were swapped. The saturated bed was regenerated with heated CO_2 to remove the adsorbed water. The hot CO_2/vapor stream was cooled (internal to the dehydration package) and flowed back to the first stage CO_2 compressor knockout drum where the water was separated.

The dried CO_2 leaving the dehydration unit entered the third stage of the centrifugal compressor where it was compressed and cooled to 91 bar (1315 psia). The supercritical fluid was collected in a drum and pumped and cooled to the final pipeline pressure of 153 bar (2215 psia). At this pressure, the supercritical CO_2 can enter an enhanced oil recovery (EOR) pipeline and be utilized for increased petroleum production, or be injected underground for geologic storage.

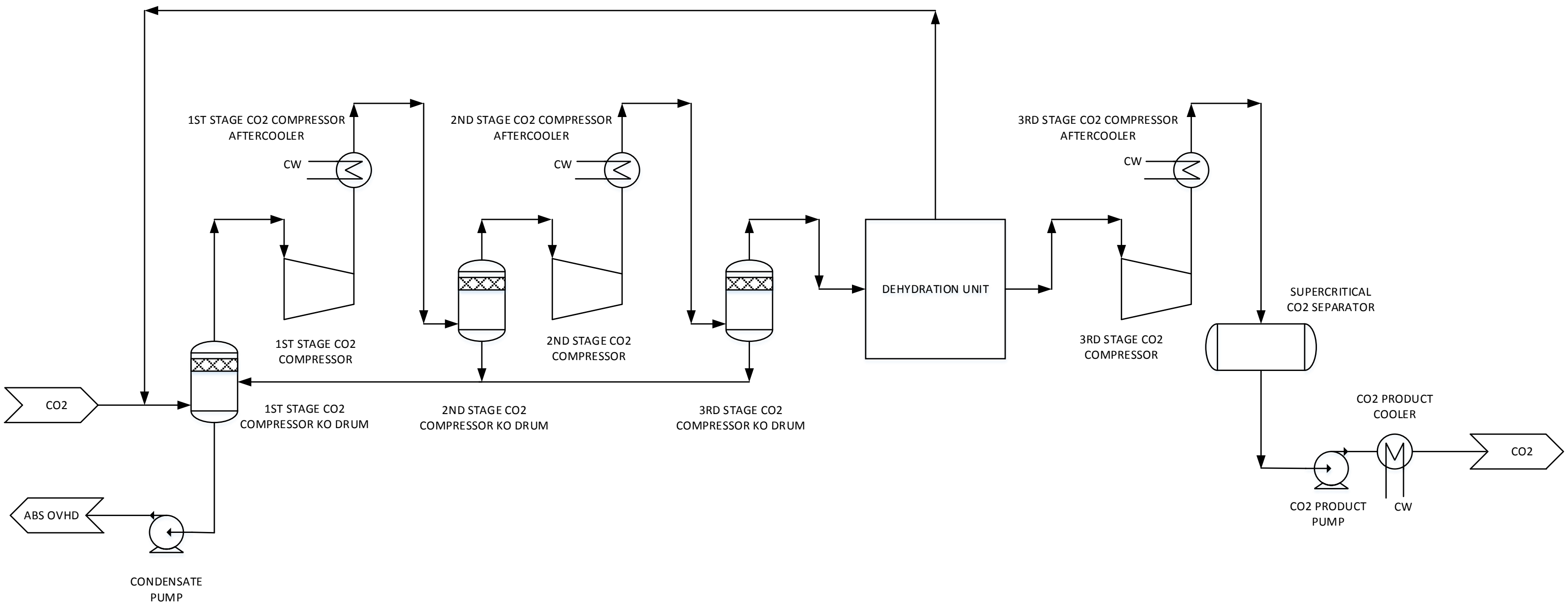

Figure 5-2
CO_2 Compression System

Design of the PCC System

Because of the proprietary nature of this solvent, a minimum amount of information about it was provided by the NCCC. Therefore, to produce a heat and material balance and to estimate the performance, the following information and assumptions were used:

- The steam to the solvent regenerator reboiler was 2080 kJ/kg (895 Btu/lb) of CO_2 removed.
- The CO_2 concentration in the lean solvent was 2.55 wt%.
- The CO_2 concentration in the rich solvent was 11.3 wt%.
- The flue gas scrubber and flue gas blower parameters were the same as in EPRI report 1017515. (These are shown in Appendix A).
- The lean solvent entered the absorber at 38°C (100°F).
- It was assumed that the reaction between the solvent and CO_2 is exothermic. Therefore, the temperature of the rich solvent leaving the absorber was assumed to be 63°C (145°F), and the flue gas exiting the absorber to the cooling portion of the column was 57°C (135°F).
- It was assumed that the lean solvent exiting the regenerator is at 124°C (255°F).
- Thermal properties are based on water.
- As in EPRI report 3002003740, the temperature of the CO_2/water stream leaving the regenerator overhead was 105°C (221°F.)
- The parameters for the CO_2 compression system were the same as EPRI report 1017515. These are also shown in Appendix A.

The steam for the regenerator reboiler was supplied from the first extraction of the low-pressure steam turbine section. At this point, the superheated steam was at 4.8 bar (69 psia), but at a temperature of 385°C (725°F). Desuperheating water from the condensate pump outlet was used to cool the steam to approximately 150°C (300°F) before entering the regenerator reboiler.

Heat Integration

In a typical utility Rankine cycle, once the steam flows through the turbine to produce electricity, it is condensed in the steam turbine condenser. For this case, it was assumed that the condenser operated at 6.7 kPa (2 in-Hg), and the condensate exited at 38°C (101°F). A conventional PC unit, has few opportunities to heat this water with process streams utilizing heat integration, outside of steam extracted from the steam turbine into a series of LP and HP feedwater heaters. The presence of a PCC thus presents a unique heat integration opportunity.

This study assumed that process streams for heating the condensate were available from two primary sources:

- Because the steam for regenerating the solvent was supplied from the LP turbine, the steam turbine gross output decreases. However, a significant portion of this heat could be recovered from the PCC system to heat the condensate. This reduced the extraction steam used for boiler feedwater heating and allowed the steam turbine to generate more power.
- The addition of a “combustion air preheater” to transfer heat between the flue gas and the secondary air stream was proposed in EPRI report 3002003740 and was included in this study.

CO_2 Regeneration and CO_2 Compression Heat Recovery

Within the CO_2 capture system, there are several sources of high temperature heat. For this study, it was assumed that the PCC system was located at a significant distance from the condensate system. Therefore a "tempered water" loop was added to transfer the heat between these two systems. The tempered water acted as cooling water for the exchangers in the PCC system; this hot water was then sent to the condensate system. The maximum temperature of the tempered water was approximately 104°C (220°F); therefore, the pressure of the tempered water loop could be lower than the condensate system. Even though the tempered water would be demineralized, this would act to protect the condensate in the event of a tube leak—condensate would flow into the tempered water system instead of vice versa.

The sources of heat in the PCC system, described below, are shown in Figure 5-3:

- The hot CO_2/water stream leaving the top of the regenerator was cooled before flowing to the CO_2 compressor. There was a significant quantity of water in the overhead stream that condensed as the stream was cooled. Thus the bulk of the energy transferred was latent heat. Because of the significant latent heat, the cooling curve was non-linear and AspenPlus™ was used to determine the amount of heat available for heat integration.
- The CO_2 compressor intercoolers cool the hot gas from each CO_2 compressor stage to increase the compressor efficiency and to control the gas temperature entering the next compression stage. Like the regenerator overhead stream, this stream contained water vapor, so the cooling curve was non-linear and AspenPlus™ was used to determine the amount of heat available for heat integration.
- The individual tempered water streams were combined into a single stream that flowed from the PCC system to the feedwater heater system and was used to heat a portion of the condensate.

Combustion Air Preheater

The combustion air preheater concept presented in EPRI report 3002003740 noted that the flue gas leaving the primary air heater was at a significantly higher temperature (approximately 127°C or 260°F) than that of the combustion air. Typically, the flue gas stream leaving the ID Fan would be quenched before entering the FGD, and this heat would not be recovered.

In the report, it was proposed that a heat exchanger between the ID Fan and the FGD would be added, as shown in Figure 5-4, to transfer the heat from the flue gas to the combustion air. Because the combustion air would now enter the "primary" air heater at a higher temperature, a portion of the hot flue gas exiting the SCR could be bypassed around the air heater. This high temperature (>370°C or >700°F) could then be used as heat source for a second tempered water loop and used to heat the condensate.

For this study, it was decided that because low-sulfur PRB coal was used as the fuel, the flue gas could be cooled to a minimum temperature of 82°C (180°F) in these heat exchangers to maximize heat recovery.

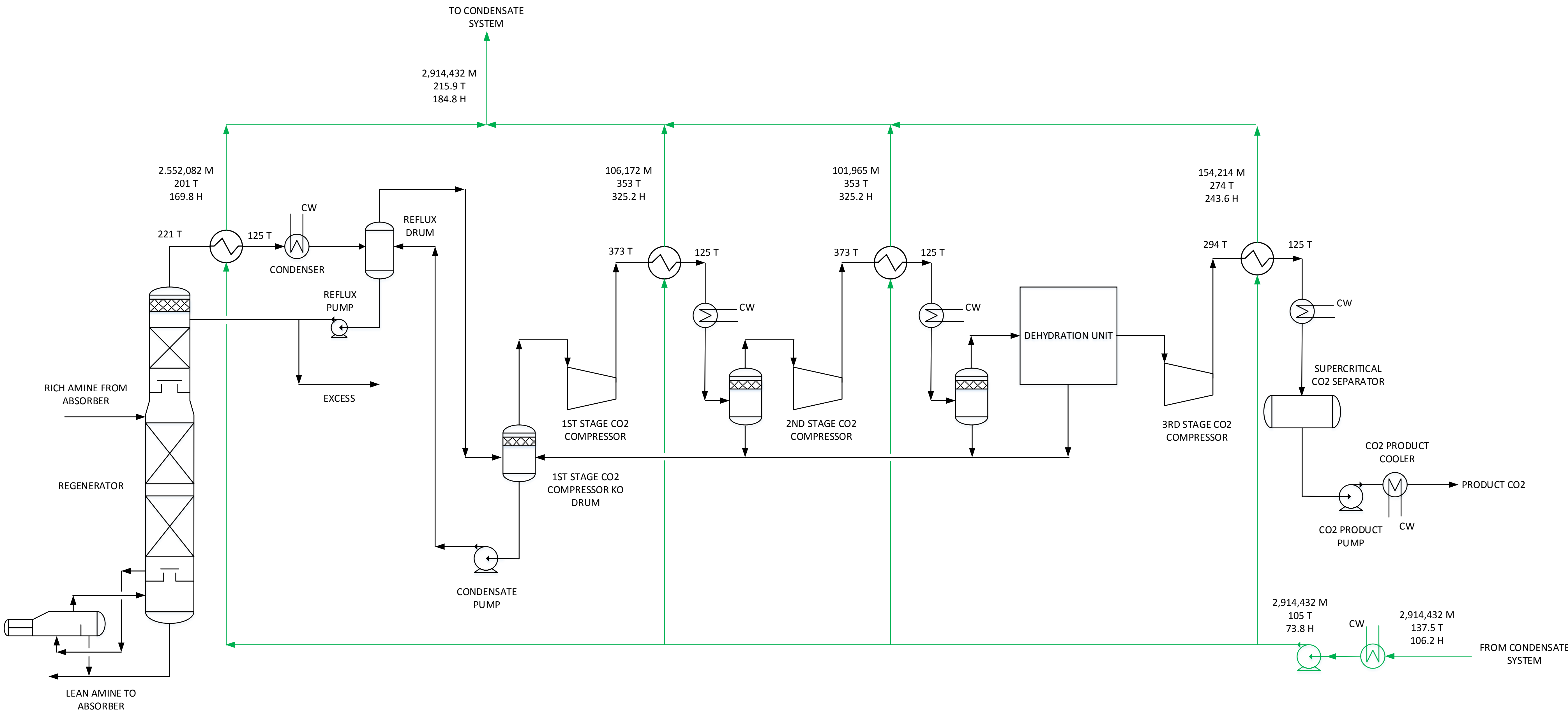

Figure 5-3
PCC Heat Integration Scheme

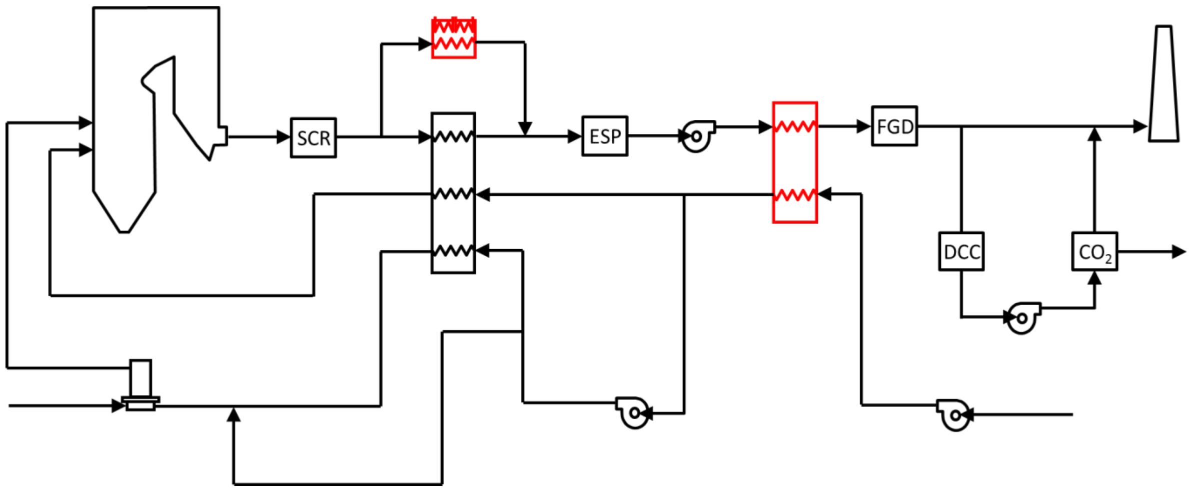

Figure 5-4
Combustion Air Preheater (from EPRI Report 3002003740)

The overall heat integration scheme for the plant is shown in the heat exchanger network diagram of Figure 5-5. The cold streams are indicated by the blue lines at the top, flowing from right to left, and the hot streams are at the bottom in red flowing from left to right. The two tempered water streams are indicated by the green lines at the bottom of the diagram. Each heat exchanger is identified by a “dumb-bell” shape that matches the hot and cold streams with each end a vertical line. Arrows have been added to the vertical “exchanger” line to indicate the direction of heat flow. The number below the lower circle in the “dumb-bell” shape is the duty for the heat exchanger in million Btu per hour.

The overall heat balance diagram for the plant, incorporating the heat integration, is shown in Figure 5-6.This diagram focuses only on the low pressure section of feedwater heating between the condenser and the deaerator. Using the heat integration scheme described eliminated feedwater heaters 1-3. The steam that would be been used by these exchangers flowed through the LP turbine and generated power, which partially offset the power lost by extracting steam to operate the PCC regenerator reboiler.

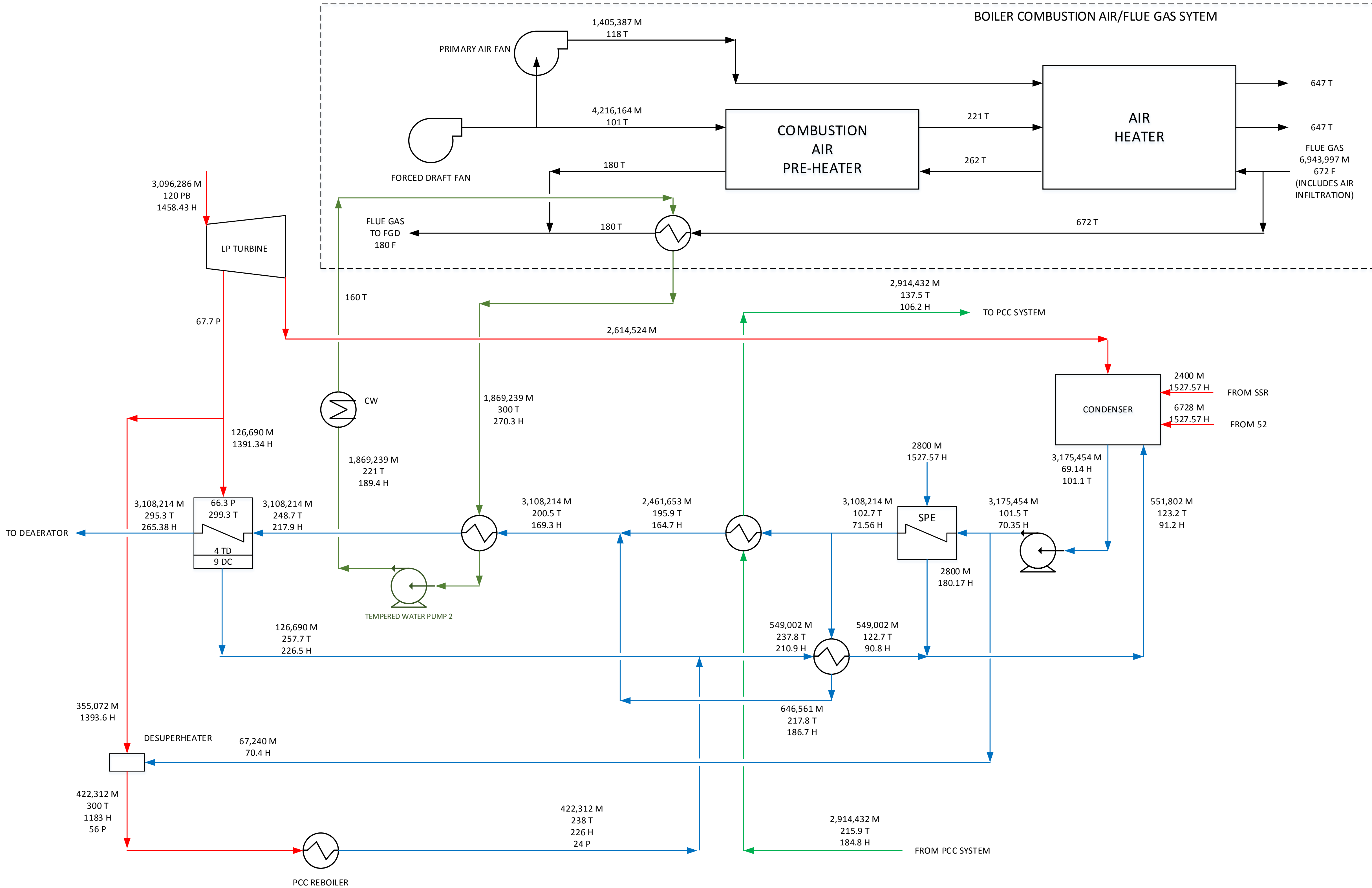

Figure 5-5
PCC Heat Integration Scheme

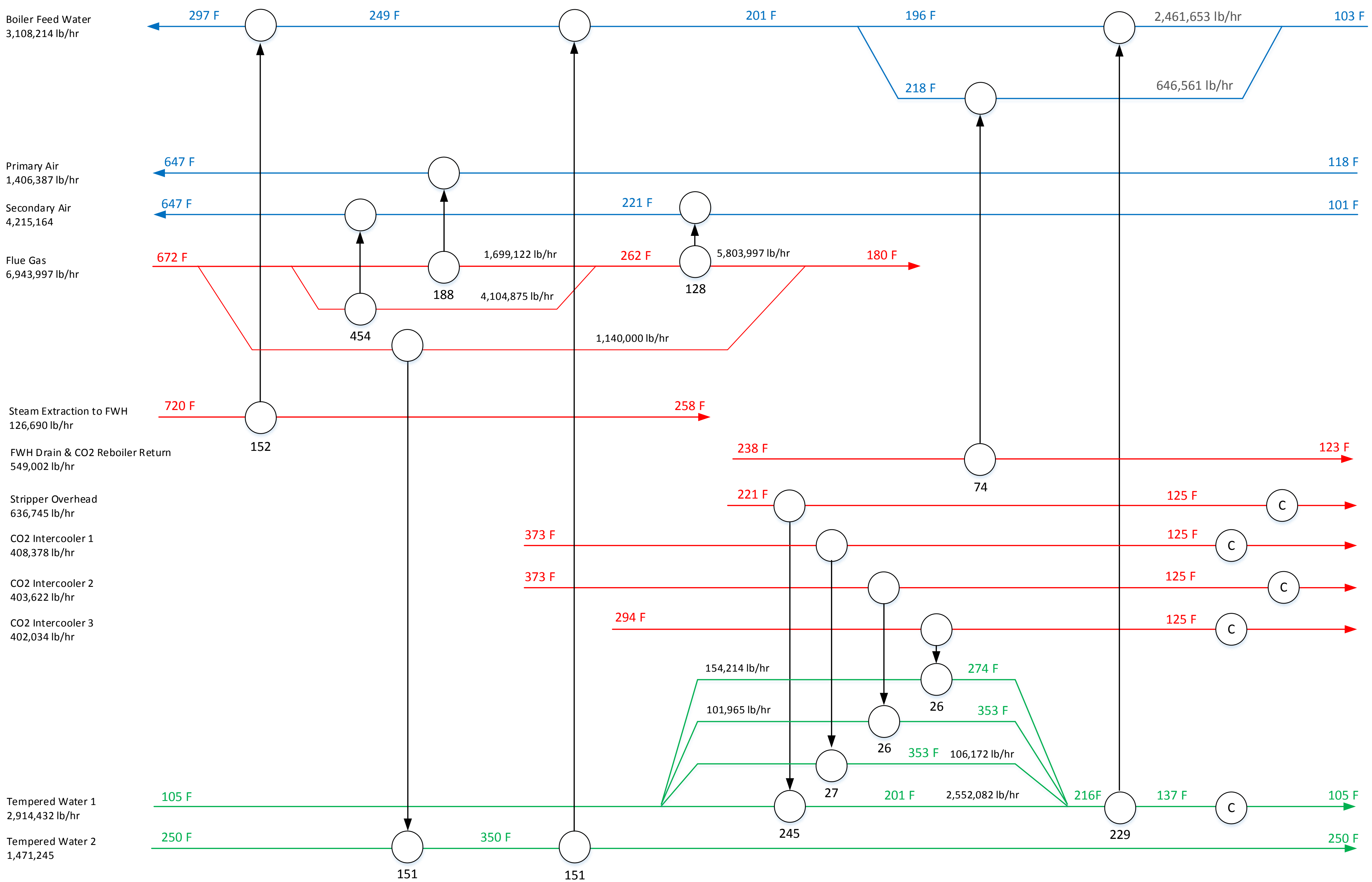

Figure 5-6
Heat Exchanger Network Diagram

Plant Performance

Based on the configuration described, the performance of the A-USC power plant utilizing the advanced solvent for CO_2 capture to meet EPA's 111(b) requirement for new coal plants is shown in Table 5-1. The auxiliary load associated with the PCC system is shown in Table 5-2.

Table 5-1
Summary Performance of the Advanced Ultra-Supercritical Plant with PCC

Description	SI Units	U.S. Customary Units
Throttle conditions	242 bar/732°C/760°C	3515 psia/1350°F/1400°F
Main steam flow	1,818,000 kg/h	4,008,000 lb/h
Hot reheat flow	1,563,000 kg/h	3,375,000 lb/h
Condenser flow	1,440,000 kg/h	3,175,000 lb/h
Condenser pressure	6.7 kPa	2 inches mercury
Final feedwater temp	304°C	580°F
Steam to PCC System	161,000 kg/h	355,000 lb/h
Gross plant output, kW	809,071	809,071
Auxiliary load, kW	100,742	100,742
Net plant output, kW	708,329	708,329
Net plant heat rate	9,261 kJ/kWh	8,778 Btu/kWh
Net plant efficiency, % (HHV)	38.9	38.9
Plant fuel consumption	338,000 kg/hr	746,000 lb/hr
CO_2 emission with PCC	393,000 kg/hr	867,000 lb/hr
CO_2 emission with PCC	500 kg/MWh, EPA gross basis	1100 lb/MWh, EPA gross basis

Table 5-2
Auxiliary Loads for the Advanced Ultra-Supercritical Plant with PCC

Equipment	kW
Original "Base" Plant Load (From Table 4-3)	70,801
PCC Plant Loads	
Wash Water Pump	496
Flue Gas Blower	5,261
Absorber Wash Cooler	163
Rich Solvent Pump	507
Lean Solvent Pump	331
Reflux Pump	15
CO_2 Compressor	21,106
Circulating Water Pump (PCC)	933
Cooling Tower Fan (PCC)	1,129
Total Auxiliary Power Consumption	100,742

Out of Service Operation

For a power plant that has a PCC system installed, there are several scenarios that could lead to the PCC system being taken out of service:

- A failure in any of the equipment in the PCC system.
- If the product CO_2 is supplied to an EOR pipeline, there could be a trip in the EOR system that prevents the "off-taker" of the CO_2 from taking delivery.
- The EPA regulations are written on an annual CO_2 emission basis. Therefore, it might be possible to oversize the PCC system to capture more CO_2 during non-peak hours and trip the PCC system during the utility's peak demand hours. As shown in Table 5-2, for Case 2 this would provide the utility with an additional 30 MW of electricity.

With the elimination of the first three LP feedwater heaters, the power plant in Case 2 must evaluate operation with the PCC system out of service, compare this operating case to the performance of Case 1, and determine how much steam extraction would be required from this first LP turbine nozzle to heat the condensate to the same temperature. For this analysis, the following assumptions were made regarding the heat integration:

- Heat from the LP feedwater heater drain would be recovered by heating condensate before the LP feedwater heater drain flow entered the condenser.
- There would be no steam extraction to the PCC system, and likewise, there would be no hot tempered water from the PCC system.
- The combustion air pre-heater system would be still be operational and would transfer heat from the flue gas to the condensate.

The low pressure condensate heat balance is shown in Figure 5-7. Notice that the steam extraction flow is approximately 158,000 kg/hr (349,000 lb/hr), almost three times as much as the extraction steam flow to the LP feedwater heater during normal operation with the PCC in service. But with the PCC reboiler out of service, the steam extracted from the LP turbine is about 60,000 kg/hr (133,000 lb/hr) less than the combined PCC reboiler steam plus the steam for the LP feedwater heater. A practical solution may be to have 2 or 3 LP feedwater heaters installed in parallel, and if the PCC system trips, split the condensate and extraction steam to multiple heaters to accommodate the increased duty. With the heat balance arrangement shown in Figure 5-7, the gross power of the steam turbine was estimated to be 821,131 kW, approximately 3.8 MW less than in Case 1.

Performance analyses for an A-USC plant with PCC designed to meet the modified EPA's 111(b) CO_2 emission standard of 635 kg/MWh (1400 lb/MWh), on a gross output basis, are under way. Preliminary findings suggest that the efficiency and heat rate would improve, respectively to 40.3% (HHV) and 8930 kJ/kWh (8470 BTU/kWh). Auxiliary loads are reduced to about 84,000 kW; gross power is about 818,000 kW and net power is about 734,000 kW.

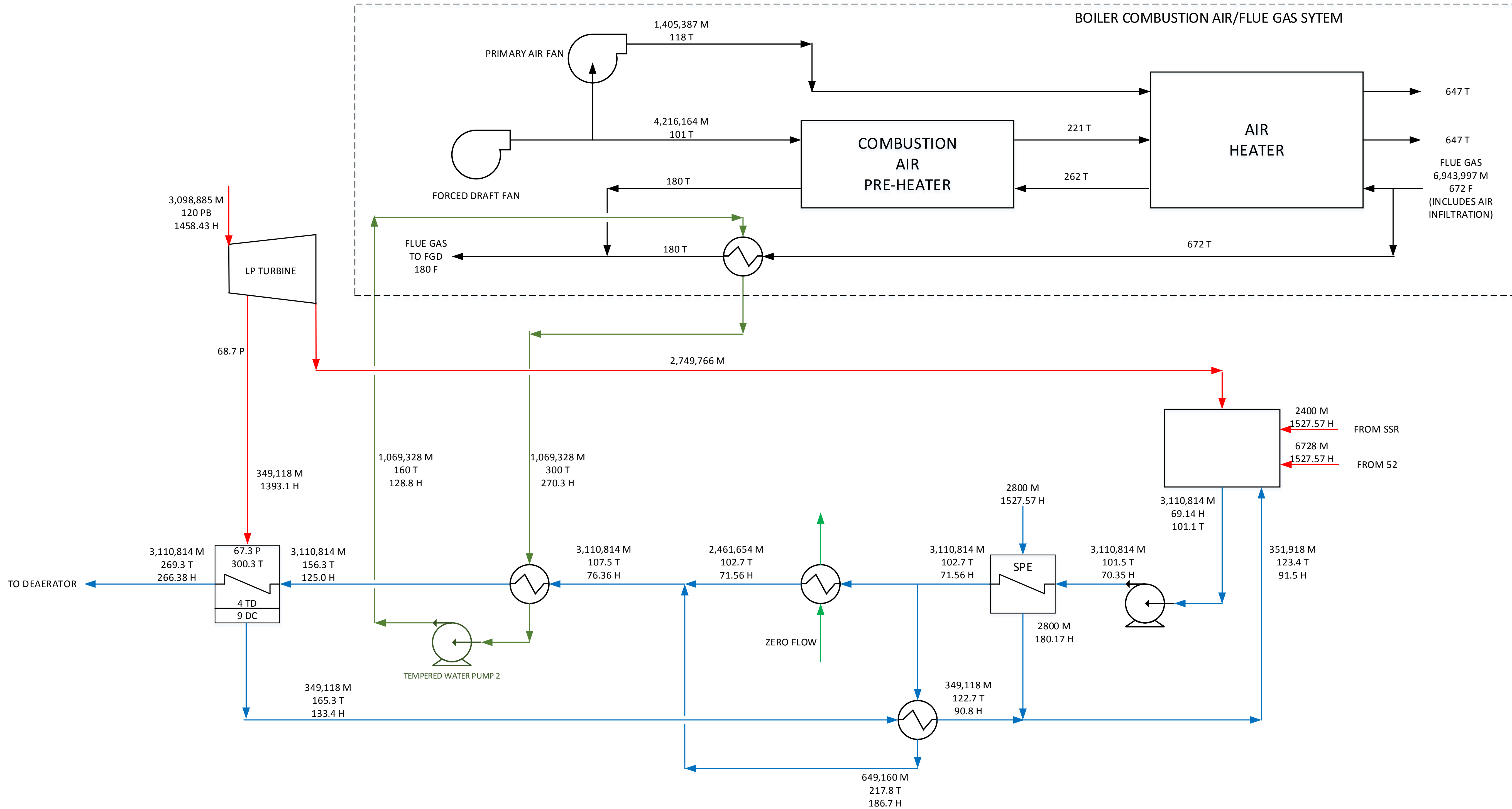

Figure 5-7
Low Pressure Condensate Heat Balance with PCC System Out of Service

6
ECONOMIC EVALUATION

Case 1 – A-USC PC without Carbon Capture

Total Plant Cost

The Total Plant Cost (TPC) for the A-USC without the PCC system is estimated using EPRI's PC Cost program. The program was developed primarily to accommodate subcritical and supercritical designs up to 593°C (1100°F) and therefore had to be modified for this study, as discussed in Section 3, to accommodate the higher cycle temperatures of an A-USC power plant.

PC Cost is not intended to be a design program and is used to provide a conceptual (±30%) cost estimate. The costs are presented on an n^{th}-of-a-kind basis; therefore, no process contingencies are applied. A contingency of 10% is applied to the project cost, and all costs are escalated to December 2014.

Steam Turbine Costs

The primary thermodynamic factors that influence the cost of a large steam turbine cycle are the steam inlet and exhaust pressure, inlet temperature and mass flow rate. Inlet pressure and temperature determine the specific volume of the steam entering the turbine. Combining this with the steam mass flow rate yields the volumetric flow rate. This determines the flow area. As the steam expands through the turbine, the pressure drops and the specific volume increases. The flow area and overall size of each turbine section increases. At the LP exhaust, the condenser pressure determines the specific volume of the steam. LP last stage buckets are designed for optimum efficiency at a fixed axial velocity. To maintain highest LP efficiency, the last stage bucket flow area should be matched to the condenser pressure and specific volume that yields the optimum axial velocity. In order to accomplish this across a range of possible condenser designs and pressures, a turbine supplier must have several last stage bucket sizes to choose from.

Once turbine section flow area and size is determined, the next major influence is steam temperature. For highest reliability, turbine materials are selected that provide adequate creep rupture and fatigue life, wear resistance, oxidation characteristics and resistance to erosion for 30 year life. As temperatures increase, the strength of turbine materials decreases. Turbine suppliers must find materials with higher and higher strength as steam temperatures increase. Higher strength alloys cost more and these costs must be exceeded by the value for the increased efficiency and output provided by the higher temperature thermodynamic cycle.

The bulk of the existing U.S. supercritical fleet operates at turbine inlet conditions of 230–242 bar (3320–3500 psig) and 538°C (1000°F). The materials applied at this temperature have generally been low alloy CrMoV steels in the HP and Reheat sections. In the LP sections, where inlet temperatures are below about 370°C (700°F), the LP casings are generally low carbon steel and the rotors are a NiCrMoV alloy. Many of these turbines have been in operation since the 1970s and are still operating today.

The advent of USC steam cycles have taken HP inlet temperatures as high as 610°C (1130°F) and reheat temperatures up to 621°C (1150°F). To achieve these temperatures, turbine suppliers have applied advanced 9–12Cr alloys in the HP and reheat sections. In some cases, steam cooling is used to lower the local temperature where stresses are highest. Inlet temperatures to the LP section have generally remained below 370°C (700°F), allowing the continued use of conventional supercritical plant materials in the LP section.

During the early years of USC technology, the advanced 9–12Cr materials commanded large cost premiums, as much 200–300% compared to the more common low-alloy steels. As of mid-2015, the cost difference between a low-alloy steel casting and a 10Cr casting is less than 100%.

The materials down-selected for steam turbine applications at A-USC temperatures are Haynes 282 and Nimonic 105. These nickel alloys are expensive to produce, process and machine. Based on the Haynes 282 castings and forgings procured during this program, the material cost of steam turbine castings and forgings, ready for machining, was estimated. The cost for machining Haynes 282 was based on experience machining other nickel alloys.

This A-USC cost estimate used an 800 MW USC reference design of the same tandem compound architecture described in Section 4. Steam conditions for the reference design are 242 bar (3514.7 psia), 600°C (1112°F) main steam, and 621°C (1150°F) reheat steam. The reference HP section is a single flow steam path with double shell construction. There are two inner shells contained within the outer shell. The inlet valves are mounted close by the HP section on each side. There are two bearings supporting the HP rotor. The HP rotor is a welded construction. The reference Reheat section is similar in all aspects to the HP section. The Reheat steam path flow direction is opposite the HP section. There are two double flow LP casings, each LP flow containing a 1016 mm (40 inch) last-stage bucket. Each LP rotor is supported by two bearings. The LP sections exhaust downward into two condensers.

This reference steam turbine design provided HP and Reheat components of similar size, quantity and weight to the A-USC design described by the heat balance. For the estimate of the stationary components, it was only necessary to adjust the material and labor costs for those components exposed to steam temperatures above the capability of advanced 9-12 Cr steels. For the HP and Reheat turbine rotating components, the total rotor weights are similar to the A-USC design. Cost adjustments were made to factor in the bolted rotor architecture with associated material changes. For the buckets and nozzles, once again, the critical sizes were similar to the reference design. Cost adjustments were made to account for material, method of attachment to the rotor and labor.

For the A-USC design, the LP inlet steam temperature is 460°C (860°F), which meant that the standard LP inner casing and the LP rotor, both designed for 370°C (700°F) inlet steam, had to be adjusted to account for the increased temperature. For the inner casing, it was assumed that the component would be made from a CrMo alloy typically used at that temperature. For the LP rotor, a three piece welded rotor was designed and the associated cost was applied to the estimate.

The main stop and control valves for the reference turbine were of the same type and size needed for the A-USC design. Material and labor cost adjustments were made to account for the required nickel alloy material needed for the higher temperature. The combined reheat valves for the reference design were larger than those required for the A-USC design and the next smaller valve design was applied to A-USC and material and labor cost adjustments were made to this valve.

The reference design uses piping to conduct the steam from the main stop and control valves to the HP turbine inlet. For A-USC this piping will be Inconel 740 and the reference piping cost has been adjusted to reflect the material change.

The A-USC Steam Turbine and Generator cost includes the flange-to-flange steam turbine-generator with electrical, controls and accessories as listed:

1. Main Stop & Control valves
2. Main steam piping
3. HP Section
4. Reheat Stop & Intercept Valve
5. IP Section
6. Crossover between IP Exhaust & LP Section
7. Two (2) Double Flow LP sections
8. Bearings, Bearing enclosures
9. Generator
10. Electrical System, Instruments & Control System
11. Accessories (to include):
    a. Lube Oil System
    b. Hydraulic Power Unit
    c. Steam Seal System
    d. Turbine Drain Valves

A-USC Steam Turbine & Generator cost does not include:

1. Installation Cost
2. Extraction Piping
3. Hot & Cold Reheat Piping

The results of the A-USC Steam Turbine-Generator cost study are summarized in Figure 6-1.

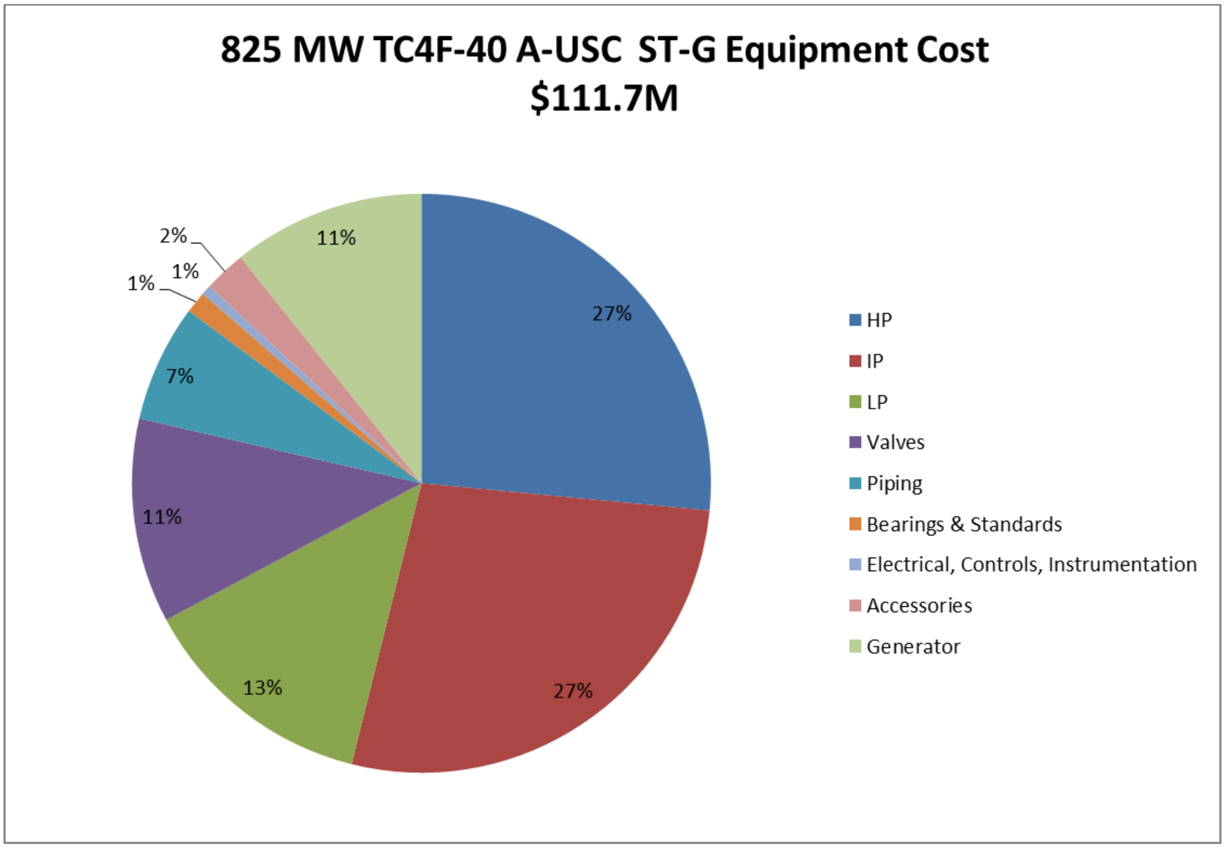

Figure 6-1
Steam Generator Cost Study Results

Compared to the USC reference design, the A-USC steam turbine-generator cost is nearly double the comparable USC cost. This is due mainly to cost increases of 3-4 times in the HP and IP sections, with the increase higher in the IP section due to its larger size. Valve and piping increases contribute significantly to the remaining cost increases.

These large increases are mainly the result of the high cost of the nickel alloys, which currently represent an increase in cost for a delivered casting or forging of close to 8 times compared to advanced 9–12Cr ferritic steels. Add to this the difficulty for machining super-alloys and the nickel alloy multiplier reaches 10 times.

Regarding the ST-G cost for Case 1 and Case 2 at 3500 psi inlet pressure, it is assumed to be the same. The ST-G cost increase for the 4250 psi case was estimated to be $1.8 million. This was based on calculations of wall thickness changes and the influence on component weight.

It remains to be seen how the cost for these nickel alloys changes in the future. This study made no attempt to predict the "nth unit" cost for these materials.

Levelized Cost of Electricity

The LCOE includes the fuel costs, variable and fixed operating costs, as well as the capital cost "carrying charge."

The costs are based on the following assumptions:

- Annual capacity factor: 80%
- Coal cost: $1.71/GJ ($1.80/MBtu)
- Annual Capital Carrying Charge: 12.1% of Total Plant Cost (TPC)
- Operator Cost: 18 operators per shift
- Administration/Overhead Cost: 22 personnel
- Maintenance labor: 2% of TPC/yr
- Maintenance materials: 1% of TPC/yr
- Ammonia for SCR: $573/tonne ($520/ton)
- FGD reagent cost: $19/tonne ($17/ton)
- FGD solids disposal costs: $13/tonne ($12/ton)
- Ash disposal costs: $19/tonne ($17/ton)
- Other miscellaneous variable costs: 1% of TPC/yr
- Select material costs ($/lb):[21]
    - Carbon steel – $1.18
    - T91 – $2.63
    - S304H – $7.03
    - HR3C – $8.11
    - HR6W – $16.23
    - HR230 – $21.64
    - IN617/IN740 – $40

Based on this information, the TPC and the LCOE for Case 1 is shown in Table 6-1.

[21] EPRI market analysis based on material purchases from 2012-2014, informal discussions with vendors, and raw alloy costs.

Table 6-1
Total Plant Cost and Cost of Electricity – without Post-Combustion CO_2 Capture

Total Plant Cost – 754,199 kW Net Power		
Cost Category	**Capital Cost, $(000)**	**$/kW**
Earthwork	$25,400	$34
Boiler Room Structure	$54,800	$73
Turbine/BOP/Yard Structures	$94,900	$126
Boiler Plant Equipment & Systems	$471,500	$625
Turbine & BOP	$548,300	$727
Flue Gas Desulfurization	$241,400	$320
Miscellaneous Directs	$152,800	$203
Indirect Costs	$421,700	$559
Contingency	$201,100	$267
Total	$2,212,000	$2,933
Levelized Cost of Electricity – 80% Capacity Factor		
Cost Category	**Annual Operating Cost, $(000)/yr**	**$/MWh**
Capital Cost	$267,652	$50.60
Fuel Costs	$78,433	$14.80
Fixed Operating Costs	$76,287	$14.40
Variable Operating Costs	$26,038	$4.90
Total	$448,410	$84.70

Case 2 – A-USC PC with Post-Combustion CO_2 Capture

Total Plant Cost

For Case 2, the design and cost of the PCC system is based on the system described in Appendix D of EPRI report 1017515.[22] Because only 34% of the flue gas is treated in the PCC system, only one CO_2 removal/compression "train" would be necessary for the plant to meet EPA's 111(b) requirement. For scaling purposes the following assumptions were made:

- The columns and fan cost was scaled based the flue gas flow rate
- The pump and vessel/storage costs were scaled based on the solvent flow rate
- The heat exchangers were scaled based on the calculated heat duties
- The CO_2 compressor was scaled based on the CO_2 product flow rate

[22] *An Engineering and Economic Assessment of Post-Combustion CO_2 Capture for 1100°F Ultra-Supercritical Pulverized Coal Power Plant Applications: Phase II Task 3 Final Report.* EPRI, Palo Alto, CA: 2010, 1017515.

The cost basis for the original report is October 2009, and Chemical Engineering's Plant Cost Index was used to escalate the costs to a December 2014 basis.

The estimate produced by PC Cost for the boiler island does not include the modifications to the ductwork that are required to incorporate the PCC system. Therefore a "retrofit" cost is added to incorporate the modifications made to accommodate the PCC system to the existing plant design ductwork, steam system, controls, etc. The costs necessary to incorporate the heat integration scheme from Section 5 were included as well.

Levelized Cost of Electricity

To adjust the LCOE to incorporate CO_2 capture, the following changes to the previously mentioned assumptions are made:

- The number of plant operators is increased from 18 to 22 per shift.
- The number of administration personnel is increased from 22 to 25 to accommodate additional lab personnel.
- A variable cost of $4.08/yr*(hr/lb) or $1.85/yr*(hr/kg) multiplied by the solvent circulation rate was added to account for the annual costs associated with the MEA solvent (replacement and disposal).
- A CO_2 transportation and storage cost of $10/tonne ($9.07/ton) is added.

The TPC and the LCOE for Case 2 are shown in Table 6-2.

Table 6-2
Total Plant Cost and Cost of Electricity – with Post-Combustion CO_2 Capture

Total Plant Cost – 708,362 kW Net Power		
Cost Category	**Capital Cost, $(000)**	**$/kW**
Base Plant	$2,212,000	$3,120
Modifications for PCC System	$178,000	$250
Total	$2,390,000	$3,370
Levelized Cost of Electricity – 80% Capacity Factor		
Cost Category	**Annual Operating Cost, $(000)/yr**	**$/MWh**
Capital Cost	$287,859	$58.00
Fuel Costs	$78,433	$15.80
Fixed Operating Costs	$83,632	$16.90
Variable Operating Costs	$34,032	$6.90
CO_2 Transportation & Storage	$12,800	$2.60
Total	$498,067	$100.20
Cost of CO_2 Avoided, $/tonne	$65.84	
Cost of CO_2 Captured, $/tonne	$45.01	

Economic analyses for an A-USC plant with PCC designed to meet the modified EPA's 111(b) CO_2 emission standard of 635 kg/MWh (1400 lb/MWh), on a gross output basis, are under way. Preliminary findings suggest that the capital cost would be reduced to about $2.34 million ($3,190/kW) and the levelized cost-of-electricity would drop to about $93.25/MWh.

The capital savings associated with the alternative A-USC boiler arrangements to reduce the nickel piping length and the reduction in materials would also apply to the capital cost of the system with a PCC system installed, lowering the overall plant capital cost and the cost of electricity.

A-USC Boiler Configurations to Reduce Capital Cost

Past EPRI studies (Reports 1015699, 1026644, 3002001788) estimate the capital cost of an A-USC PC unit at approximately 10% more than the cost of a "state-of-the-art" USC PC unit based on U.S. market prices and labor rates. A significant portion of this increase is due to the high cost of the main steam and reheat steam lines in a conventionally arranged boiler. These lines are approximately 140 m (450 ft) long, and would be constructed from IN740. IN740 is a new material for this application, and the price is very high—approximately $75–90/kg ($35–40/lb) at the time of the studies. Due to the long length of these piping runs and the thickness of the main steam line, the cost of these two piping runs can approach the cost of the boiler!

Two design approaches have emerged to address the cost of these piping leads. One approach, being pursued in China is to split the HP and IP steam turbine into two components. The HP turbine and a “high temperature” IP (HTIP) turbine section would be located at the top of the boiler near the main and reheat steam outlets, with a conventional IP/LP turbine located on the turbine floor several hundred feet away (i.e., short connecting piping runs where nickel alloy is required and longer runs where ferritic alloys are sufficient). The exhaust of the HTIP turbine would be reheated, but to a more “typical” reheat temperature before flowing to a conventional IP/LP turbine. This arrangement is illustrated in Figure 6-2 and Figure 6-3.

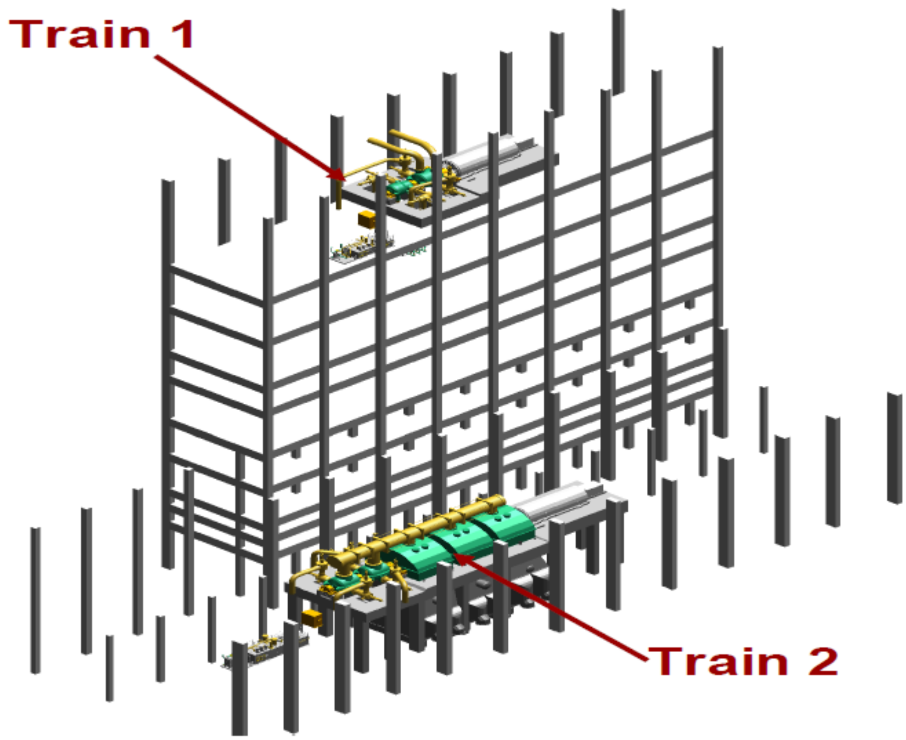

Figure 6-2
Layout of a High and Low Position Steam Turbine Arrangement

(Source: Shanghai Waigaoqiao)

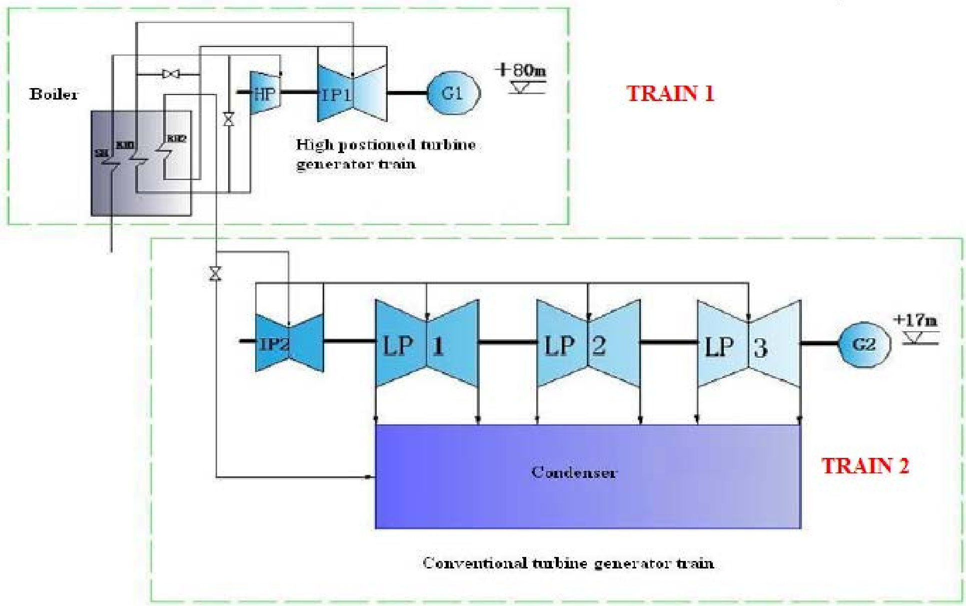

Figure 6-3
Turbine Train Schematic Showing Components

(Source: Shanghai Waigaoqiao)

A second approach is the "Inverted Tower" arrangement proposed by Babcock and Wilcox, shown in Figure 6-4.[23]

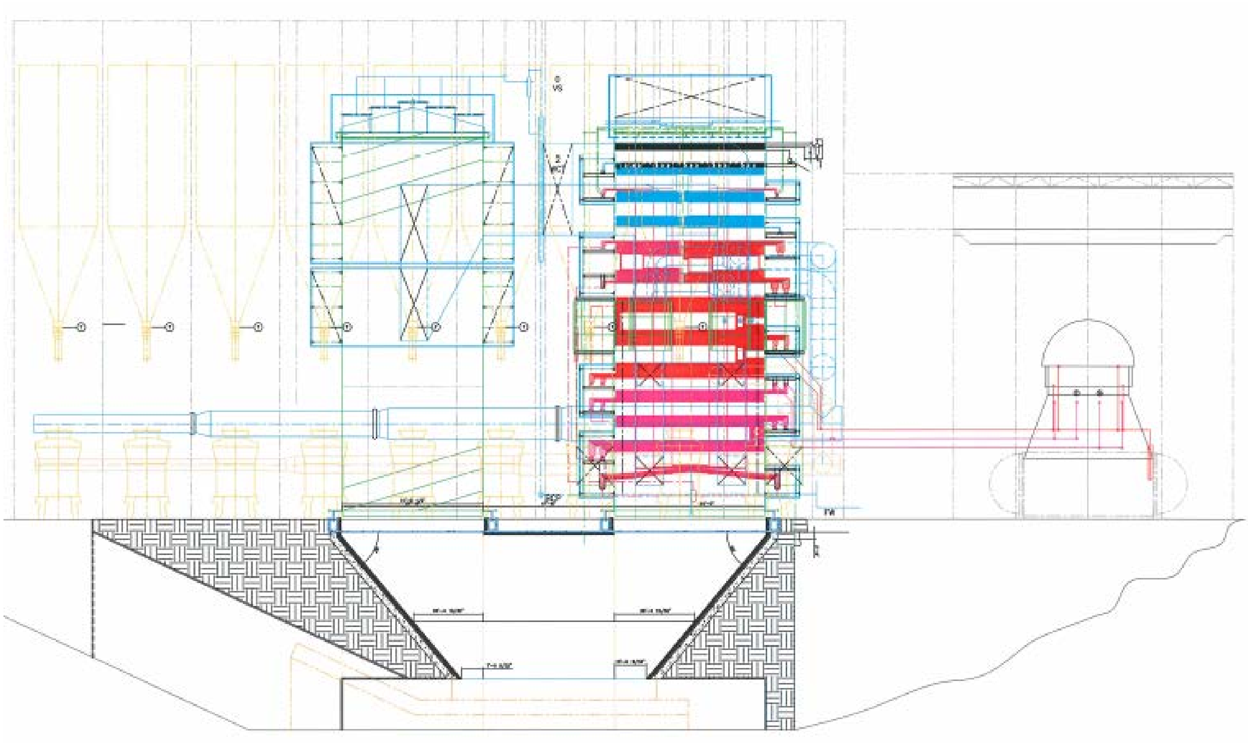

Figure 6-4
Babcock and Wilcox Downdraft Inverted Tower Design

In this design, coal is fed into the furnace enclosure (shown on the left in the drawing) and the hot gas exhausts vertically down into a refractory lined "adiabatic tunnel," where the flue gas turns 180 degrees and flows upward through the backpass. In this configuration, the finishing superheater and reheater are located at the turbine elevation, minimizing the length of the nickel alloy piping. According to a DOE study[24], the capital cost of an A-USC PC boiler with this configuration could be comparable to that of a USC PC boiler.

There was insufficient information to estimate the design and cost of these two arrangements. However, a "sensitivity" was performed assuming that main steam/reheat steam piping was one-half of the original length. This would decrease the capital cost by about $95 million or 4.3%.

Another likely decrease in capital cost would be due to a drop in the cost of the IN740 piping. IN740 is a relatively new material, and to date the supply for the material in boiler applications has been very limited. As A-USC boiler technology becomes more widespread, the demand for the material will create a greater supply base and the price may fall. A second sensitivity analysis was performed, which indicated that a 20% decrease in IN740 cost would decrease the capital

[23] Courtesy of Babcock and Wilcox.

[24] "Estimated Costs of the AUSC Boiler, Steam Turbine", and Steam Piping, U.S. Department of Energy/NETL Office of Fossil Energy, under DOE Contract Number DE-FE0004001, ESPA Task 341.03.01, September 5, 2014.

cost by approximately $41 million or 1.9%. The combination of the lower IN740 cost with the shorter main steam/reheat steam piping would decrease the cost by over $110 million or 5.1%.

Higher Pressure Steam Cycle

Increasing efficiency is the motivation behind increasing the temperature for A-USC steam conditions and, as discussed in Section 4, increasing the steam cycle pressure further increases the cycle efficiency. One of the sensitivities studied for this report included evaluating the performance and economics of a 242 bara (3500 psig) steam cycle compared with a 294 bara (4250 psig) steam cycle for Case 2 CO_2 removal. Section 4 concluded that higher inlet pressure was not economical for an A-USC PC unit without carbon capture. Following the same methodology laid out for the PCC system with heat integration as described in Section 5, a comparison of the two steam cycles is shown in Table 6-3 to determine if higher inlet pressures are economical for an A-USC PC unit with carbon capture.

Table 6-3
Performance Comparison of a 242 bara (3500 psig) steam cycle compared to a 294 bara (4250 psig) steam cycle with Carbon Capture

Description	241 bara (3500 psia)	293 bara (4250 psia)
Gross Power, kW	809,071	809,628
Auxiliary load, kW	100,742	101,918
Net Power, kW	708,329	707,710
Heat Rate, Btu/kWh	8,778	8,640
Net Efficiency, %	38.9	39.5
Portion of flue gas treated, %	34.0	33.1
TIC, $(000)	$2,390,000	$2,430,000
TIC, $/kW	$3,370	$3,430
Annual Operating Cost, $/yr,	$498,067,000	$503,382,000
LCOE, $/MWh	$100.20	$101.50
Heat Rate Improvement, Btu/kWh	138	
Delta Plant Cost, $(000)	$40,000	
Delta cost per Btu/kWh	$290,000	

Based on the assumptions made in the analysis, interesting observations include:

- The increased efficiency of the steam cycle decreased the coal being burned in the boiler by about 2.7%. This decreased the amount of flue gas required to be treated in the PCC system and reduced the auxiliary load of the PCC system approximately 1 MW.
- However, the increased discharge pressure of the boiler feedwater pump for the 4250 psi case was approximately 2 MW, so the 4250 psig cycle had a higher auxiliary load.

- Even though the cost of the 4250 psig system PCC system was marginally lower in cost, the difference was relatively small. What was more significant was the increased capital cost due to the thicker boiler components, especially the high-cost nickel alloys.

From this analysis, it appears that the addition of a PCC system decreases the cost per unit of heat rate improvement associated with moving from 3500 psig to 4250 psig when compared to the value presented in Section 4. However, a value of roughly $290,000 per Btu/kWh of heat rate is still higher than the U.S. utility market is likely to accept.

7
CONCLUSIONS AND FUTURE WORK

Summary of Findings

A-USC development over the past 15+ years has significantly advanced the technology to the point where it is on the cusp of commercial deployment. The increased efficiency of the technology has two significant business "drivers:"

1) In power markets where CO_2 regulations require the use of carbon capture on coal plants, increasing the efficiency of the power cycle decreases the CO_2 emitted per MW of electricity produced. In scenarios that require partial CO_2 capture, such as that proposed by the U.S. EPA 111(b) regulation, the higher efficiency decreases the penalty associated with adding carbon capture to the power plant.

2) In the U.S. power industry market, coal fired plants typically have low fuel prices and high capital/labor costs. The largest component of the LCOE is associated with the capital for building the plant. However, this is not the case in other parts of the world where fuel prices are much higher and labor is less expensive. In these cases the fuel cost savings associated with the improved efficiency offsets the higher capital cost.

Past EPRI studies have shown that when compared to USC, the cost of A-USC is approximately 10% higher in capital cost. For this study, the estimated capital cost was $2,933/kW, and the LCOE was $84.70 (Table 7-1 and Table 7-2). A significant contributor to the higher cost has been the long main and reheat steam piping lines, which would be fabricated from Inconel 740. This impact on capital cost has led the power industry to respond with several innovative boiler concepts, as discussed in Section 6. Capital cost estimates on some of these concepts indicate that they may result in reductions that would bring the cost of an A-USC plant into parity with an equivalently sized USC plant. As these concepts are further developed, A-USC technology will likely become more cost competitive. The A-USC programs in countries such as China and India have announced plans to build plants within the next decade. As experience is gained with the technology, the risks and cost associated with a new technology will diminish.

In addition to the research on A-USC, research continues around the world on CO_2 capture technologies for PC plants. Facilities like the DOE's National Carbon Capture Center are evaluating the performance of new solvents and CO_2 technologies that will dramatically decrease the cost of post-combustion carbon capture. Coupling the high efficiency of A-USC technology with these advanced solvents and other removal technologies should minimize the cost of carbon capture from pulverized coal plants.

In this study, an advanced solvent system was used to remove CO_2 down to a level of 500 kg CO_2/MWh (1100 lb CO_2/MWh). To meet this requirement, only 34% of the flue gas had to be treated. Steam from the turbine was required to regenerate the solvent, but due to the high efficiency and the advanced solvent the steam flow was dramatically reduced compared to past studies on USC power plants using MEA as the solvent for carbon capture. In Case 2 the gross power from the steam turbine decreased only 16 MW (about 2%) when this regeneration steam was extracted from the turbine. The auxiliary load increased nearly 30 MW, with 21 of the 30 MW associated with CO_2 compression.

The capital cost of the CO_2 removal equipment was estimated to be approximately $178 million, approximately 8% more than the base plant cost. But when divided by the net power the cost increased $440/kW (about 15%) due to a combination of the increased capital and lost generation. The LCOE increased about 19% due to the increased capital, lost generation, and increased operating labor and materials associated with the carbon capture system.

Table 7-1
Case 1 vs. Case 2 Comparison (SI Units)

Description	Case 1	Case 2
Feedwater heaters	Five LP, deaerator, three HP, topping desuperheater	
Feed pump drive	Motor	
Cooling system	Mechanical draft cooling tower	
Emission controls	SCR, ESP, Wet FGD, and $CaBr_2$ injection into the furnace for NO_x, particulate, sulfur, and mercury control	
Throttle conditions	242 bar/732°C/760°C	242 bar/732°C/760°C
Main steam flow	1,818,000 kg/h	1,818,000 kg/h
Hot reheat flow	1,563,000 kg/h	1,563,000 kg/h
Condenser flow	1,412,000 kg/h	1,440,000 kg/h
Condenser pressure	6.7 kPa	6.7 kPa
Final feedwater temp	304°C	304°C
Steam to PCC System	N/A	161,000 kg/h
Gross plant output, kW	825,000	809,071
Auxiliary load, kW	70,801	100,742
Net plant output, kW	754,199	708,329
Net plant heat rate	8,698 kJ/kWh	9,261 kJ/kWh
Net plant efficiency, % (HHV)	41.4	38.9
Plant fuel consumption	338,000 kg/hr	338,000 kg/hr
CO_2 emission without PCC	596,000 kg/hr	393,000 kg/hr
CO_2 emission without PCC	722 kg/MWh, gross basis	500 kg/MWh, EPA gross basis
Total Capital Cost, $(000)	$2,212,000	$2,379,000
Total Capital Cost, $/kW	$2,933	$3,370
Total Operating Costs, $(000)/yr	$448,410	$498,067
LCOE, $/MWh	$84.70	$100.20

Table 7-2
Case 1 vs. Case 2 Comparison (U.S. Customary Units)

Description	Case 1	Case 2
Feedwater heaters	Five LP, deaerator, three HP, topping desuperheater	
Feed pump drive	Motor	
Cooling system	Mechanical draft cooling tower	
Emission controls	SCR, ESP, Wet FGD, and $CaBr_2$ injection into the furnace for NO_x, particulate, sulfur, and mercury control	
Throttle conditions	3515 psia/1350°F/1400°F	3515 psia/1350°F/1400°F
Main steam flow	4,008,000 lb/hr	4,008,000 lb/hr
Hot reheat flow	3,375,000 lb/hr	3,375,000 lb/hr
Condenser flow	3,112,000 lb/hr	3,175,000 lb/h
Condenser pressure	2 inches mercury	2 inches mercury
Final feedwater temp	580°F	580°F
Steam to PCC System		355,000 lb/h
Gross plant output, kW	825,000	809,071
Auxiliary load, kW	70,801	100,742
Net plant output, kW	754,199	708,329
Net plant heat rate	8,244 Btu/kWh	8,778 Btu/kWh
Net plant efficiency, % (HHV)	41.4	38.9
Plant fuel consumption	746,000 lb/hr	746,000 lb/hr
CO_2 emissions (Case 2 with PCC)	1,313,000 lb/hr	867,000 lb/hr
CO_2 emissions (Case 2 with PCC)	1592 lb/MWh, gross basis	1100 lb/MWh, EPA gross basis
Total Capital Cost, $(000)	$2,212,000	$2,379,000
Total Capital Cost, $/kW	$2,933	$3,370
Total Operating Costs, $(000)/yr	$448,410	$498,067
LCOE, $/MWh	$84.70	$100.20

Economic analyses for an A-USC plant with PCC designed to meet the modified EPA's 111(b) CO_2 emission standard of 635 kg/MWh (1400 lb/MWh), on a gross output basis, are under way. Preliminary findings suggest that the capital cost would be reduced to about $2.34 million ($3,190/kW) and the levelized cost-of-electricity would drop to about $93.25/MWh.

The Next Step

As discussed in Section 1, many of the international programs, including the DOE/EIO lead program in the United States have tested or are planning to test steam loops at A-USC conditions. The next logical step is to move toward a commercial demonstration of the technology. With guidance from the U.S. Utility Industry, the Advanced Materials for Ultra-Supercritical Boiler and Steam Turbine Consortia developed the “Component Test (ComTest) Program Concept” as illustrated in Figure 7-1. The utility members working with the consortium identified these components, previously described in Section 1, as needing further development and demonstration: the Membrane Wall, Superheater, High-Pressure, High-Temperature Valve Test, Desuperheater/Thick-Wall Section and the Steam Turbine.

As originally conceived all of these components would be installed in and adjacent to a utility host’s USC power plant. Steam would be withdrawn from the power plant’s main steam line. A portion of the steam would be desuperheated and flow through the membrane wall, which would be installed inside the furnace. The second portion of the steam would flow through the in-furnace superheater where it would be heated to approximately 760°C (1400°F). From the exit of this superheater, the steam would flow through a gas-fired auxiliary superheater which would be used to control the temperature of the steam as the host unit changes load throughout the day. The high pressure, 760°C (1400°F) steam would next flow through the High Pressure, High Temperature valve test apparatus. Two steam turbine stop and control valves would be installed in parallel and cycle continuously throughout the test. Desuperheating the steam will be a requirement for temperature control in any utility boiler, so the steam leaving the valve test will next enter the desuperheating section. This component proved problematic in the European COMTES700 facility, so validation of this technology is crucial. A downstream thick wall section will simulate a superheater header, complete with flowing pipe penetrations. The desuperheater will be used thermally cycle this component to determine its susceptibility to cracking during boiler cycling. After the thick wall section, the steam from the membrane wall will combine with the steam flowing through the thick wall section and be depressurized to approximately 48 bar (700 psia) before flowing through the steam turbine. Depressurizing the steam will decrease its temperature, so the steam will flow through a gas-fired auxiliary reheater to increase the temperature back to 760°C (1400°F) before expanding through the steam turbine. The exhaust steam will then be desuperheated and depressurized (if necessary) before returning to the host’s steam/condensate system.

As the consortium searched for hosts for the ComTest, Youngstown Thermal in Youngstown, Ohio, expressed interest early in hosting the lower pressure components—primarily the A-USC steam turbine (Figure 7-2). Youngstown Thermal is a district heating facility that provides saturated steam at approximately 10 bar (150 psi) to its customers in the downtown area. To test the steam turbine, natural gas-fired package boilers would be rented and brought to the site (Figure 7-3). These boilers would provide steam at nominally 48 bar (700 psia) and 400°C (750°F). The steam from the package boilers would flow through a gas-fired superheater developed by Babcock and Wilcox with DOE funding (see Figure 7-4). The superheater is designed to heat the steam to 760°C (1400°F). The steam would then flow through the steam turbine and would be exhausted at approximately 12 bar (170 psia) and desuperheated to saturated temperature before flowing into the header system supplying Youngstown Thermal’s customers. This project would evaluate membrane walls (in the superheater), superheater coil, steam turbine, and high temperature (but lower pressure) steam control and stop valves at A-

USC temperatures. Proposals for this project have been submitted to the DOE and these proposals are under review as of mid-2015. At the same time, conversations continue between the Consortium and potential utility hosts about testing the components of the ComTest Program that require high-pressure steam.

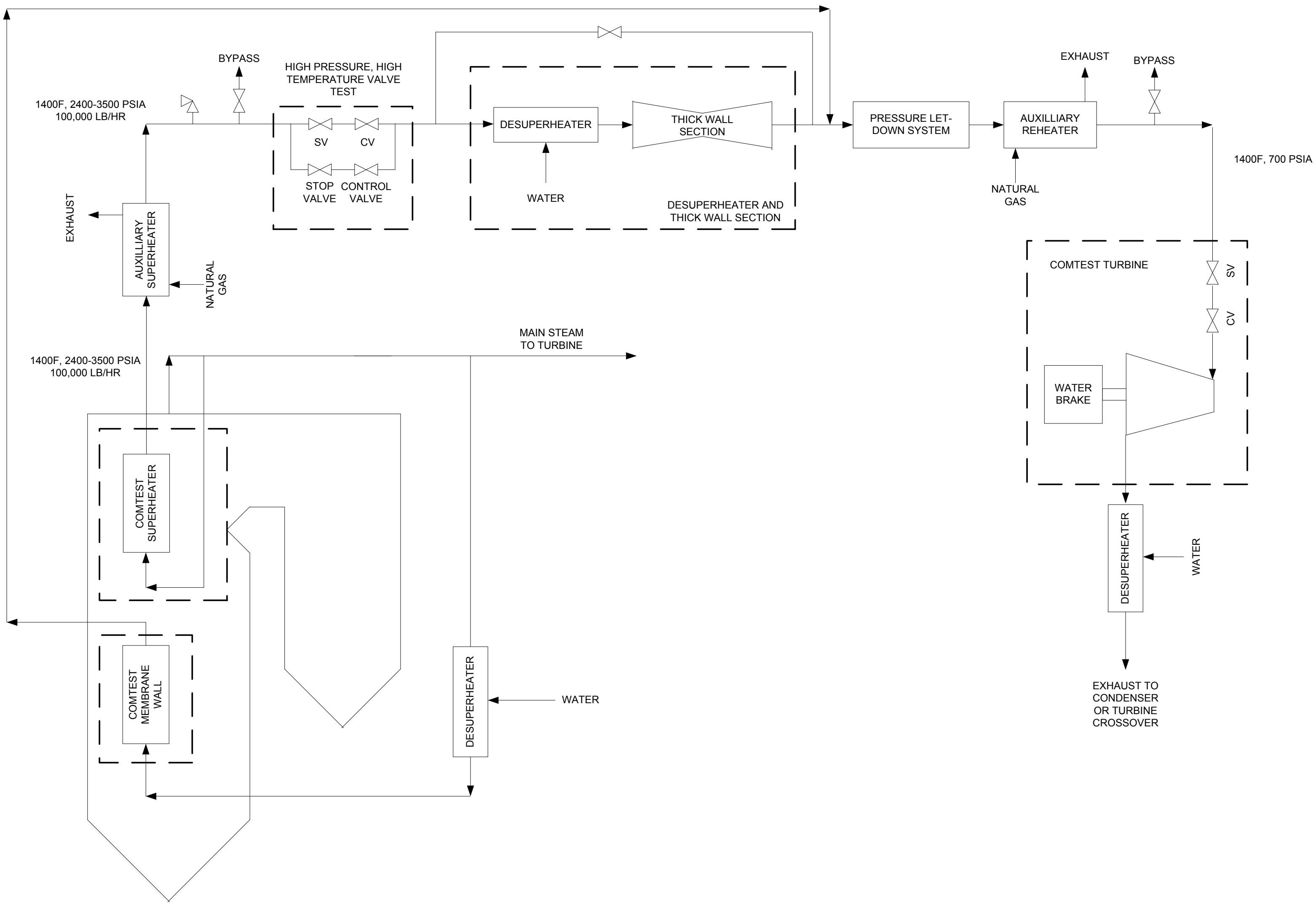

Figure 7-1
Overall A-USC ComTest Program Concept

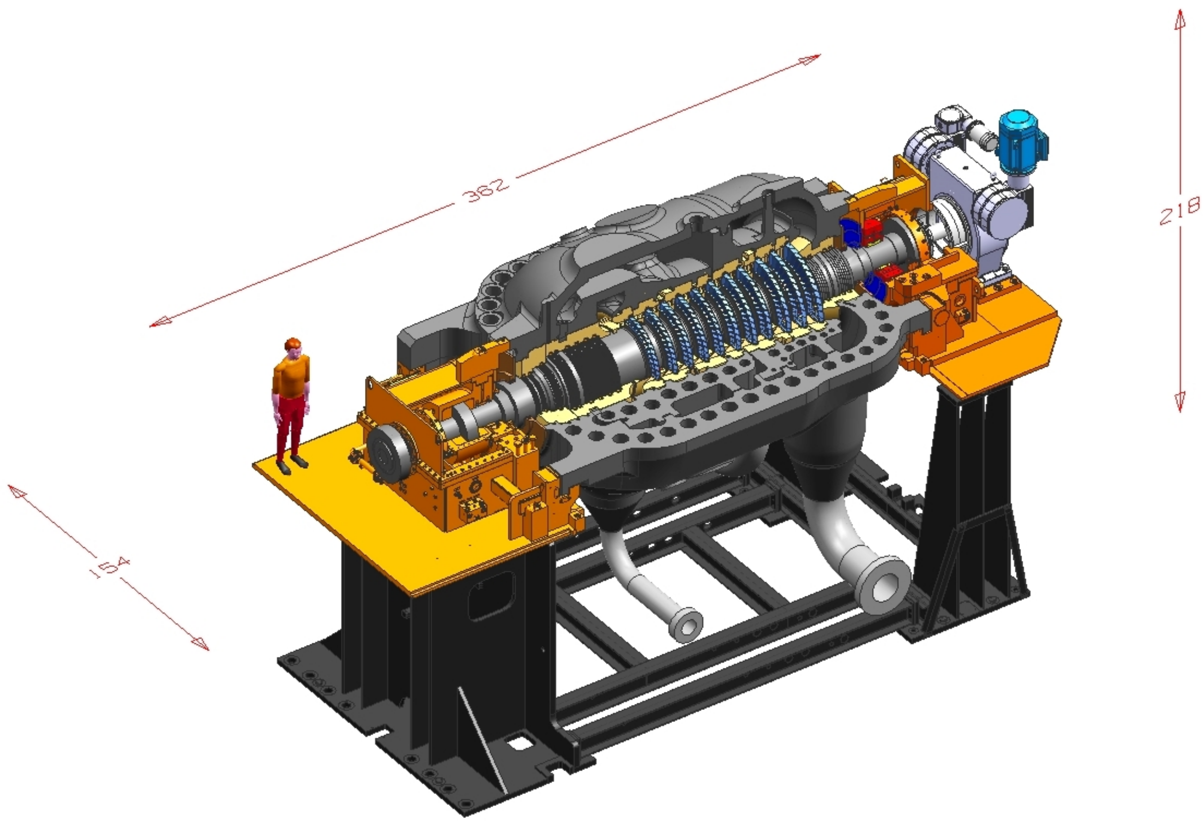

Figure 7-2
General Electric A-USC Steam Turbine Concept (Dimensions in inches)[25]

[25] Courtesy of General Electric

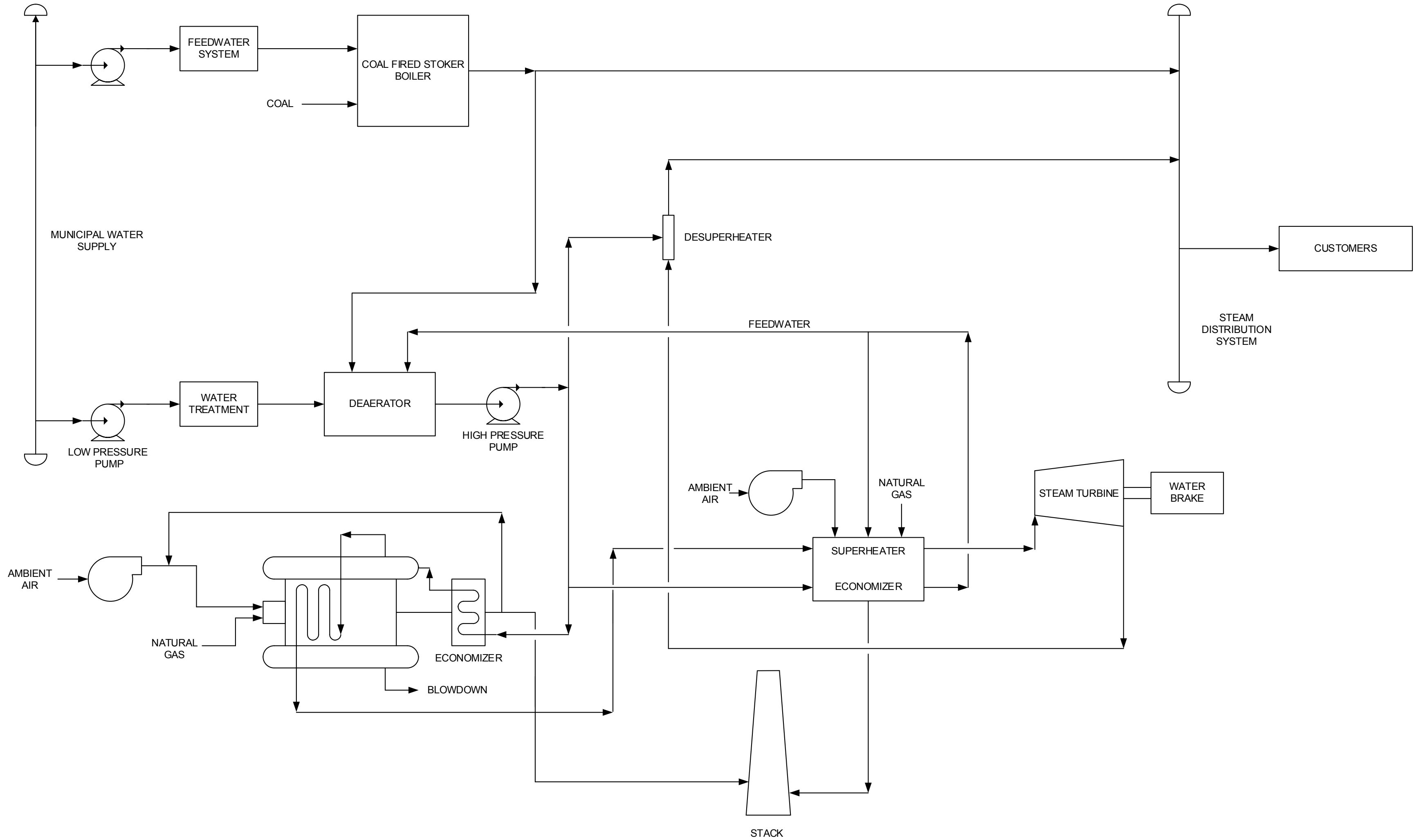

Figure 7-3
Proposed Youngstown Thermal Flowsheet

Figure 7-4
Babcock and Wilcox A-USC ComTest Superheater Tower[26]References

[26] Courtesy of Babcock and Wilcox

8 REFERENCES

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2. Engineering and Economic Analysis of 1300°F Series USC Plant with Post-Combustion Capture. EPRI, Palo Alto, CA, 2011. 1026645.
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A
PROCESS FLOW DIAGRAM/HEAT AND MATERIAL BALANCE

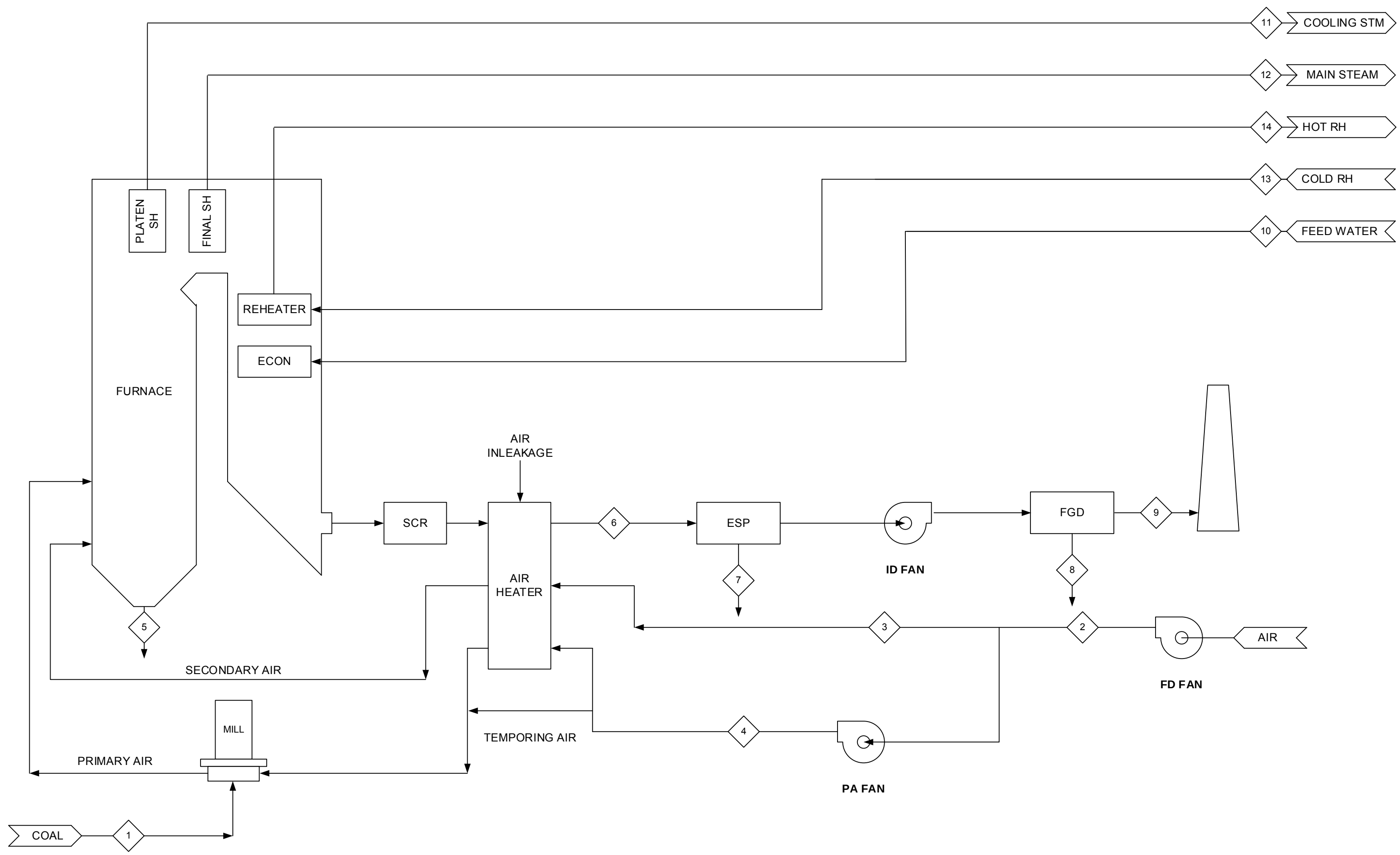

Figure A-1
Boiler Island

Table A-1
Boiler Island Material Balance for Case 1 & 2

Boiler/AQCS Heat and Material Balance											
PFD Stream		1	2	3	4	5	6	7	8	9	10
Aspen Stream		WET-COAL	TOTAIR	SATOAIRH	PATOAIRH	BTMASH	AIRHTOUT	FLYASH		0	BFW1
Description		As Received Coal	Air From FD Fan	Secondary Air to Air Heater	Primary Air To Air Heater	Bottom Ash	Flue Gas to ESP	Fly Ash	Gypsum	To Stack	Boiler Feed Water
Temperature	F	95	101	101	118	95	264	95	95	138	581
Pressure	psia	14.7	15.2	15.2	16.6	14.7	13.7	13.5	14.7	14.8	3865.3
Component											
N2	lb/hr	-	4,202,215	3,151,661	1,050,554	-	4,674,018	-	-	4,680,608	-
O2	lb/hr	-	1,275,960	956,970	318,990	-	354,011	-	-	354,679	-
CO2	lb/hr	-	-	-	-	-	1,309,556	-	-	1,313,228	-
CO	lb/hr	-	-	-	-	-	-	-	-	-	-
H2O	lb/hr	-	143,379	107,534	35,845	-	606,406	-	-	859,264	4,100,352
NO2	lb/hr	-	-	-	-	-	251	-	-	251	-
SO2	lb/hr	-	-	-	-	-	5,508	-	-	161	-
SO3	lb/hr	-	-	-	-	-	4	-	-	4	-
HCL	lb/hr	-	-	-	-	-	77	-	-	-	-
NO	lb/hr	-	-	-	-	-	-	-	-	-	-
S	lb/hr	-	-	-	-	-	-	-	-	-	-
H2	lb/hr	-	-	-	-	-	-	-	-	-	-
CL2	lb/hr	-	-	-	-	-	0	-	-	-	-
C	lb/hr	-	-	-	-	0	-	-	-	-	-
NH3	lb/hr	-	-	-	-	199	1,597	1,594	-	-	-
COAL	lb/hr	745,528	-	-	-	-	-	-	-	-	-
ASH	lb/hr	-	-	-	-	4,405	35,257	35,201	-	-	-
LIMESTONE	lb/hr	-	-	-	-	-	-	-	-	-	-
GYPSUM	lb/hr	-	-	-	-	-	-	-	18,645	-	-
Total Mass Flow	lb/hr	745,528	5,621,554	4,216,165	1,405,388	4,604	6,986,684	36,795	18,645	7,208,195	4,100,352

Table A-2
Boiler Island Material Balance for Case 1 & 2 (Continued)

Boiler/AQCS Heat and Material Balance											
PFD Stream		**11**	**12**	**13**	**14**						
Aspen Stream			STM7	CRH	HRHOUT						
Description		Cooling Steam	Main Steam	Cold Reheat	Hot Reheat						
Temperature	F	1100	1352	888	1400						
Pressure	psia	3600.0	3585.3	772.5	739.5						
Component											
Component	lb/hr	-	-	-	-						
N2	lb/hr	-	-	-	-						
O2	lb/hr	-	-	-	-						
CO2	lb/hr	-	-	-	-						
CO	lb/hr	-	4,007,852	3,375,254	3,375,254						
H2O	lb/hr	92,500	-	-	-						
NO2	lb/hr	-	-	-	-						
SO2	lb/hr	-	-	-	-						
SO3	lb/hr	-	-	-	-						
HCL	lb/hr	-	-	-	-						
NO	lb/hr	-	-	-	-						
S	lb/hr	-	-	-	-						
H2	lb/hr	-	-	-	-						
CL2	lb/hr	-	-	-	-						
C	lb/hr	-	-	-	-						
NH3	lb/hr	-	-	-	-						
COAL	lb/hr	-	-	-	-						
ASH	lb/hr	-	-	-	-						
LIMESTONE	lb/hr	-	-	-	-						
Total Mass Flow	lb/hr	92,500	4,007,852	3,375,254	3,375,254	-	-	-	-	-	-

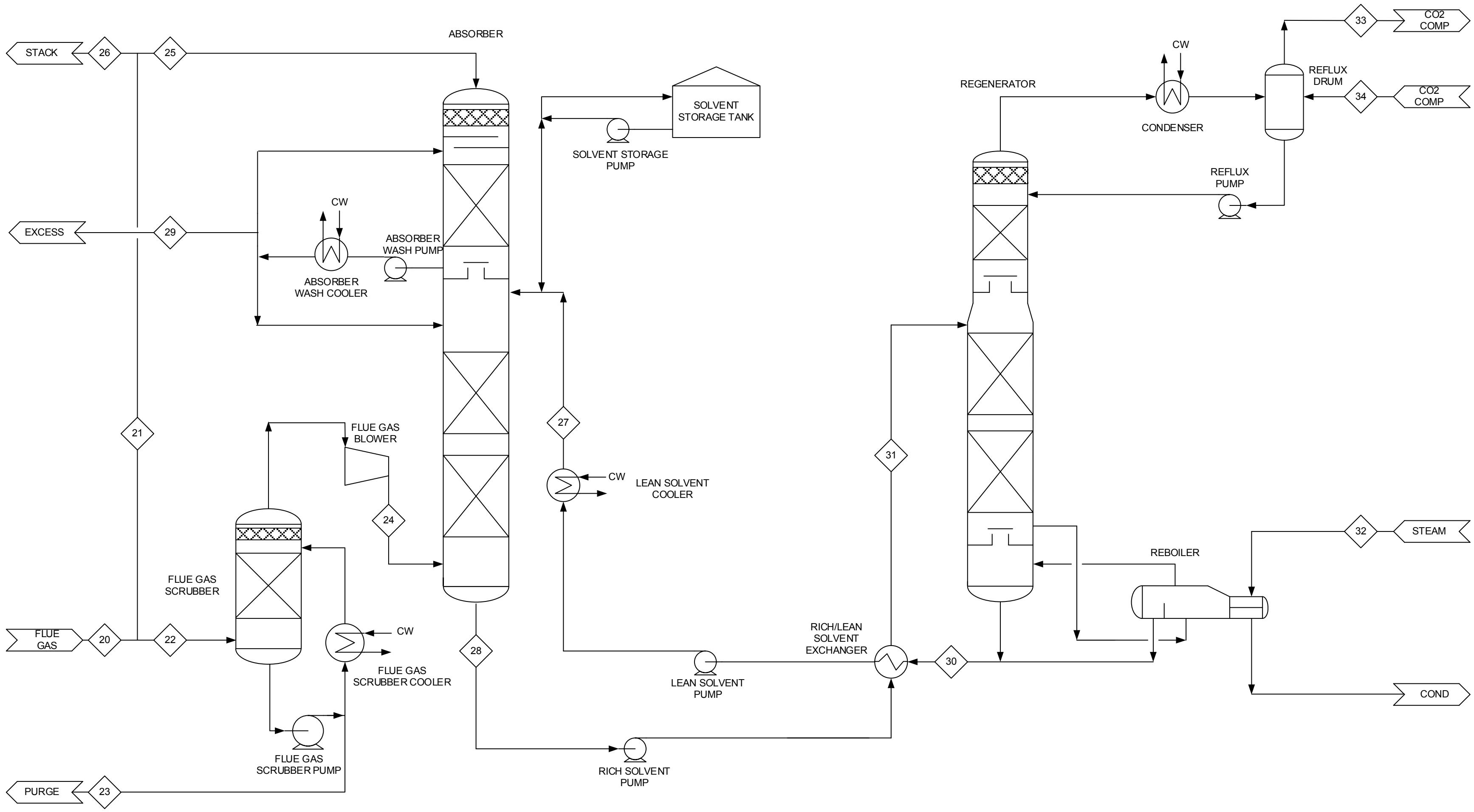

Figure A-2
Post-Combustion CO_2 Removal System

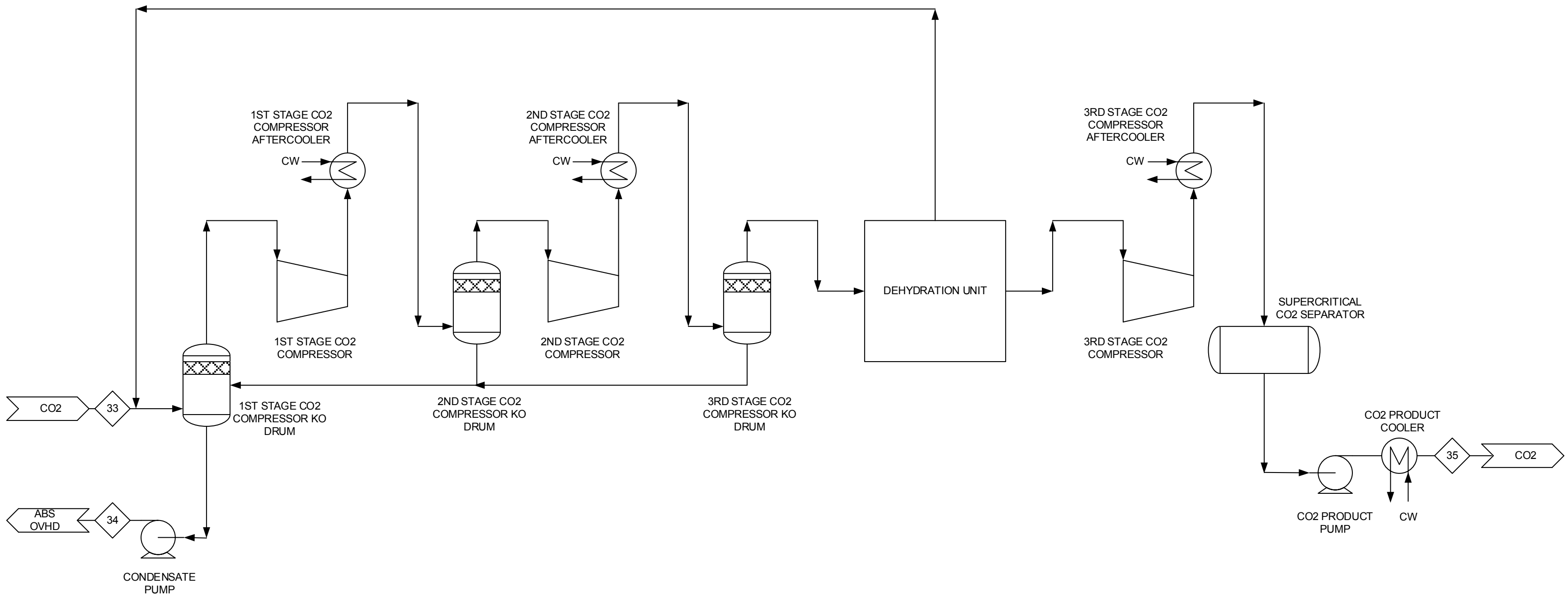

Figure A-3
CO_2 Compression System

Table A-3
Post-Combustion CO_2 Removal System Material Balance for Case 2

Stream Number	20	21	22	23	24	25	26	27	28	29	30
Stream Description	Flue Gas Downstream of ID Fan	Bypass Flow	Flue Gas to PCC System	Feed Gas Water Condensate	Flue Gas Entering Absorber	Flue Gas Exiting Absorber	Flue Gas Exiting Stack	Lean Solvent Entering Absorber	Rich Amine Exiting Absorber	Purge water from Absorber Cooler	Lean Amine Leaving Stripper Bottoms
Temperature (F)	138	138	138	123	136	130	136	100	145	130	255
Pressure (psia)	14.8	14.8	14.8	60.0	15.6	14.7	14.7	60.0	15.6	14.7	28.2
Mass Flow, lb/hr											
N2	4,680,608	3,088,465	1,592,144	-	1,592,144	1,592,144	4,680,608	-	-	-	-
O2	354,679	234,032	120,647	-	120,647	120,647	354,679	-	-	-	-
CO2	1,313,228	866,524	446,704	-	446,704	44,670	911,194	82,941	484,975	-	82,941
CO	-	-	-	-	-	-	-	-	-	-	-
H2O	859,447	567,100	292,347	161,416	130,931	198,012	765,112	-	-	28,284	-
NO2	251	165	85	-	85	85	251	-	-	-	-
SO2	161	106	55	-	55	55	161	-	-	-	-
SO3	4	3	1	-	1	1	4	-	-	-	-
HCL	-	-	-	-	-	-	-	-	-	-	-
NO	-	-	-	-	-	-	-	-	-	-	-
S	-	-	-	-	-	-	-	-	-	-	-
H2	-	-	-	-	-	-	-	-	-	-	-
CL2	-	-	-	-	-	-	-	-	-	-	-
NH3	-	-	-	-	-	-	-	-	-	-	-
C	-	-	-	-	-	-	-	-	-	-	-
COAL	-	-	-	-	-	-	-	-	-	-	-
ASH	-	-	-	-	-	-	-	-	-	-	-
SOLVENT FLOW	-	-	-	-	-	-	-	3,169,664	3,169,664	-	3,169,664
TOTAL, LB/HR	7,208,378	4,756,395	2,451,983	161,416	2,290,567	1,955,614	6,712,009	3,252,605	3,654,639	28,284	3,252,605

Table A-4
Post-Combustion CO_2 Removal System Material Balance for Case 2 (Continued)

Stream Number	31	32	33	34	35						
Stream Description	Rich Amine Entering Stripper	Steam Entering Reboiler	CO2 to Compressor	Water from CO2 Compressor	Product CO2						
Temperature (F)	235	271	100	100	100						
Pressure (psia)	60.0	36.0	21.3	30.0	2215.0						
Mass Flow, lb/hr											
N2	-	-	-	-	-						
O2	-	-	-	-	-						
CO2	484,975	-	402,034	-	402,034						
CO	-	-	-	-	-						
H2O	-	422,312	7,682	7,682	-						
NO2	-	-	-	-	-						
SO2	-	-	-	-	-						
SO3	-	-	-	-	-						
HCL	-	-	-	-	-						
NO	-	-	-	-	-						
S	-	-	-	-	-						
H2	-	-	-	-	-						
CL2	-	-	-	-	-						
NH3	-	-	-	-	-						
C	-	-	-	-	-						
COAL	-	-	-	-	-						
ASH	-	-	-	-	-						
SOLVENT FLOW	3,169,664	-	-	-	-						
TOTAL, LB/HR	3,654,639	422,312	409,715	7,682	402,034						

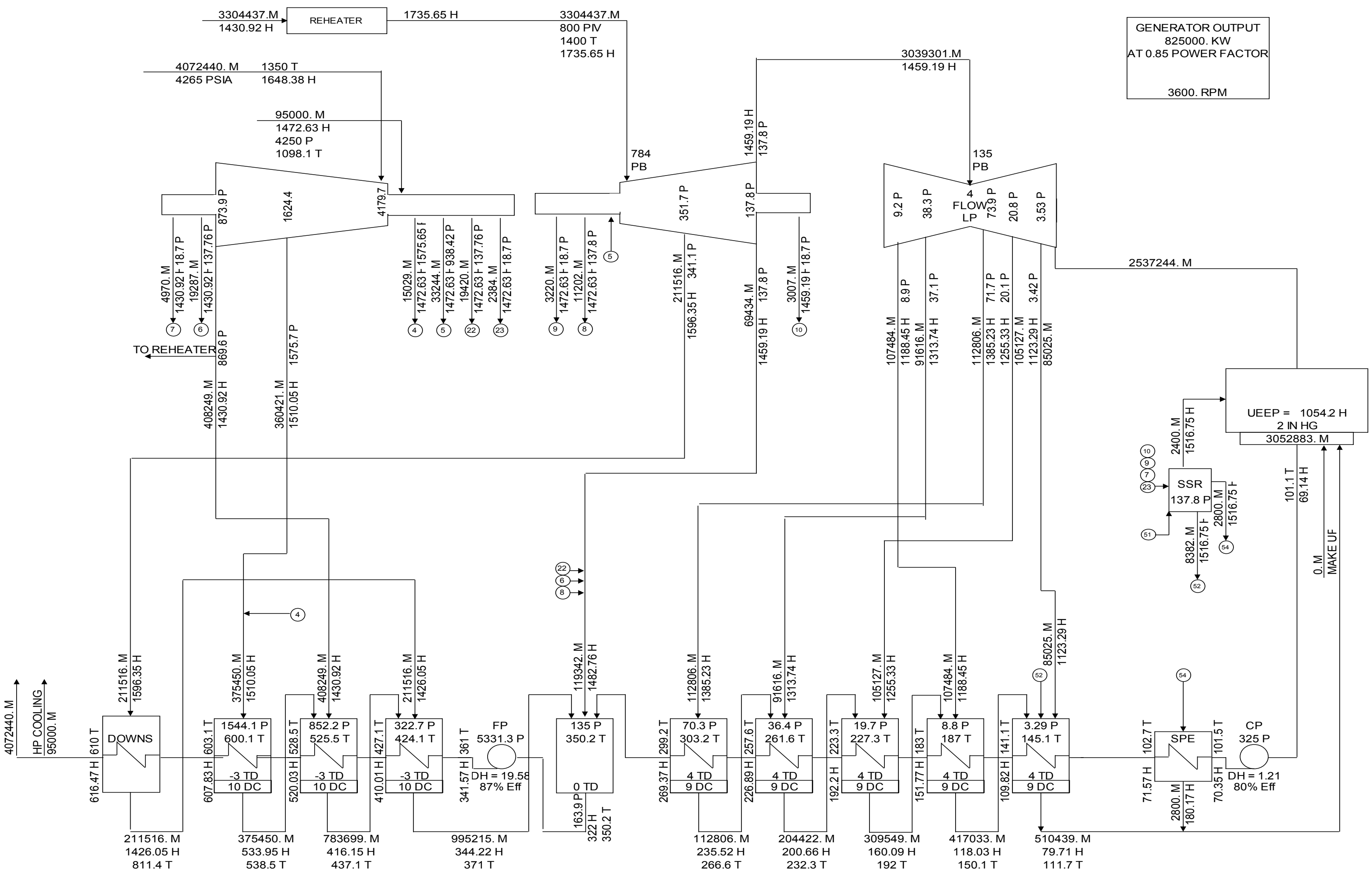

Figure A-4
4250 psig Inlet Pressure Steam Cycle Diagram, No PCC

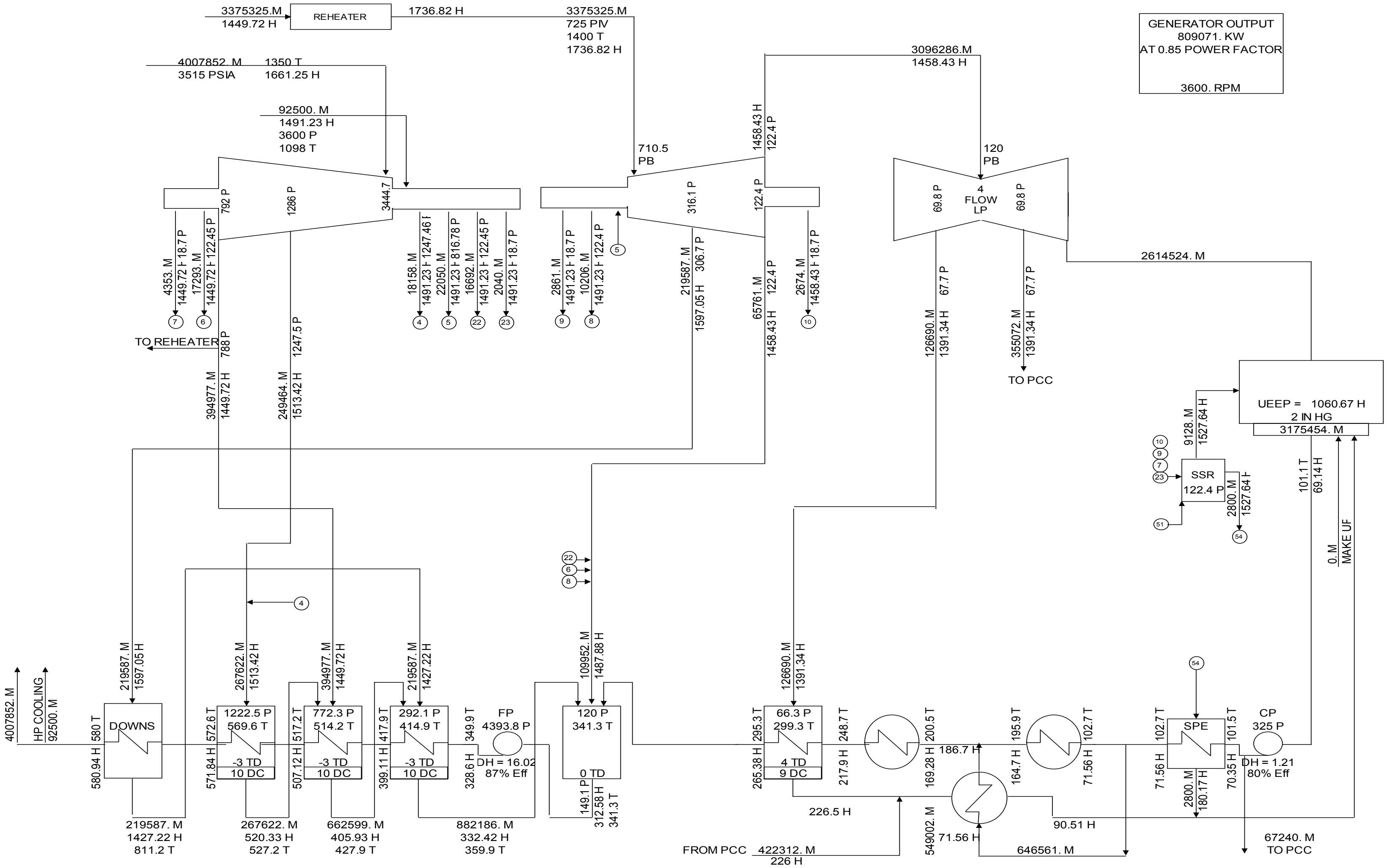

Figure A-5
3500 psig Inlet Pressure Steam Cycle Diagram, PCC System

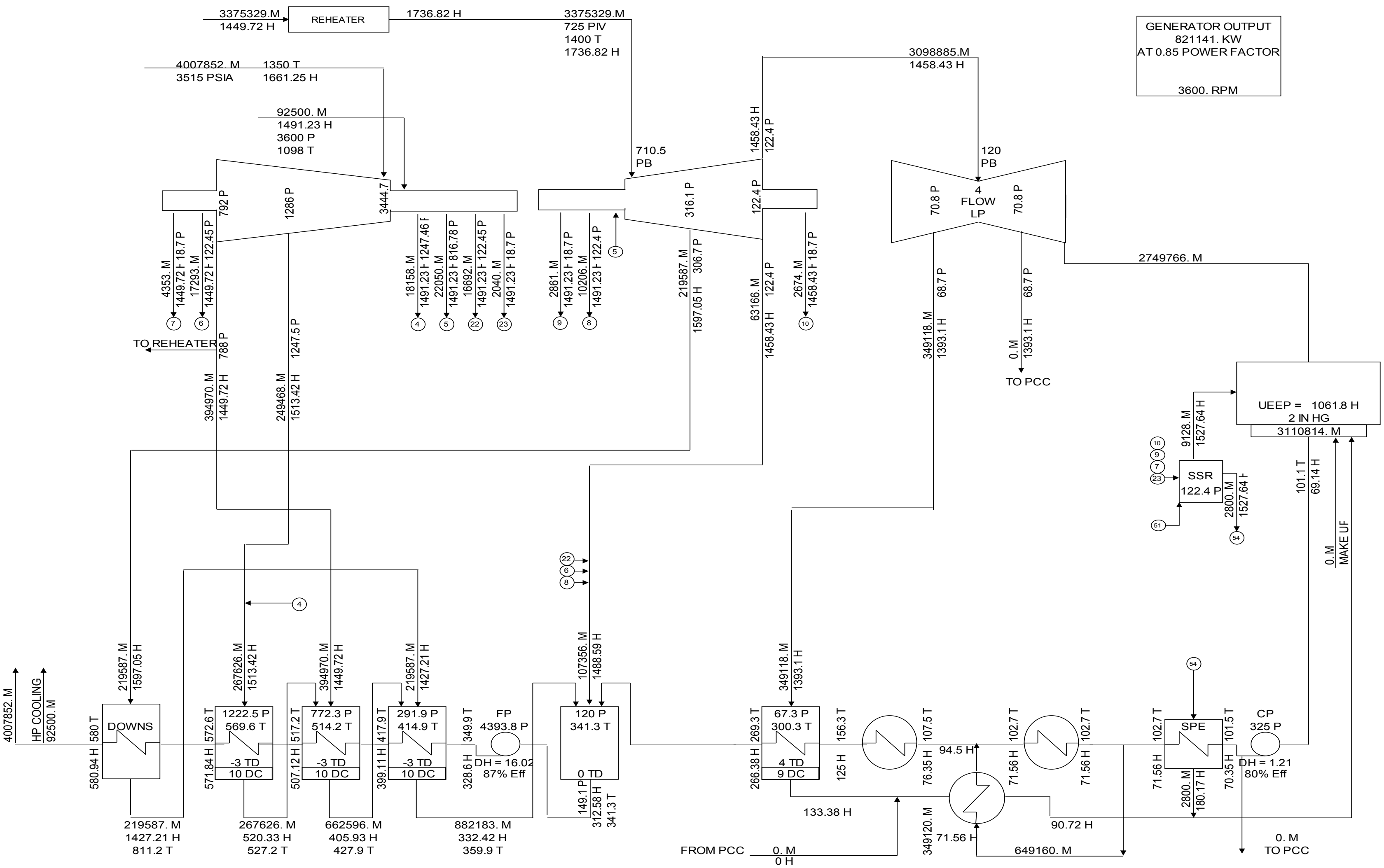

Figure A-6
3500 psig Inlet Pressure Steam Cycle Diagram, PCC System Off

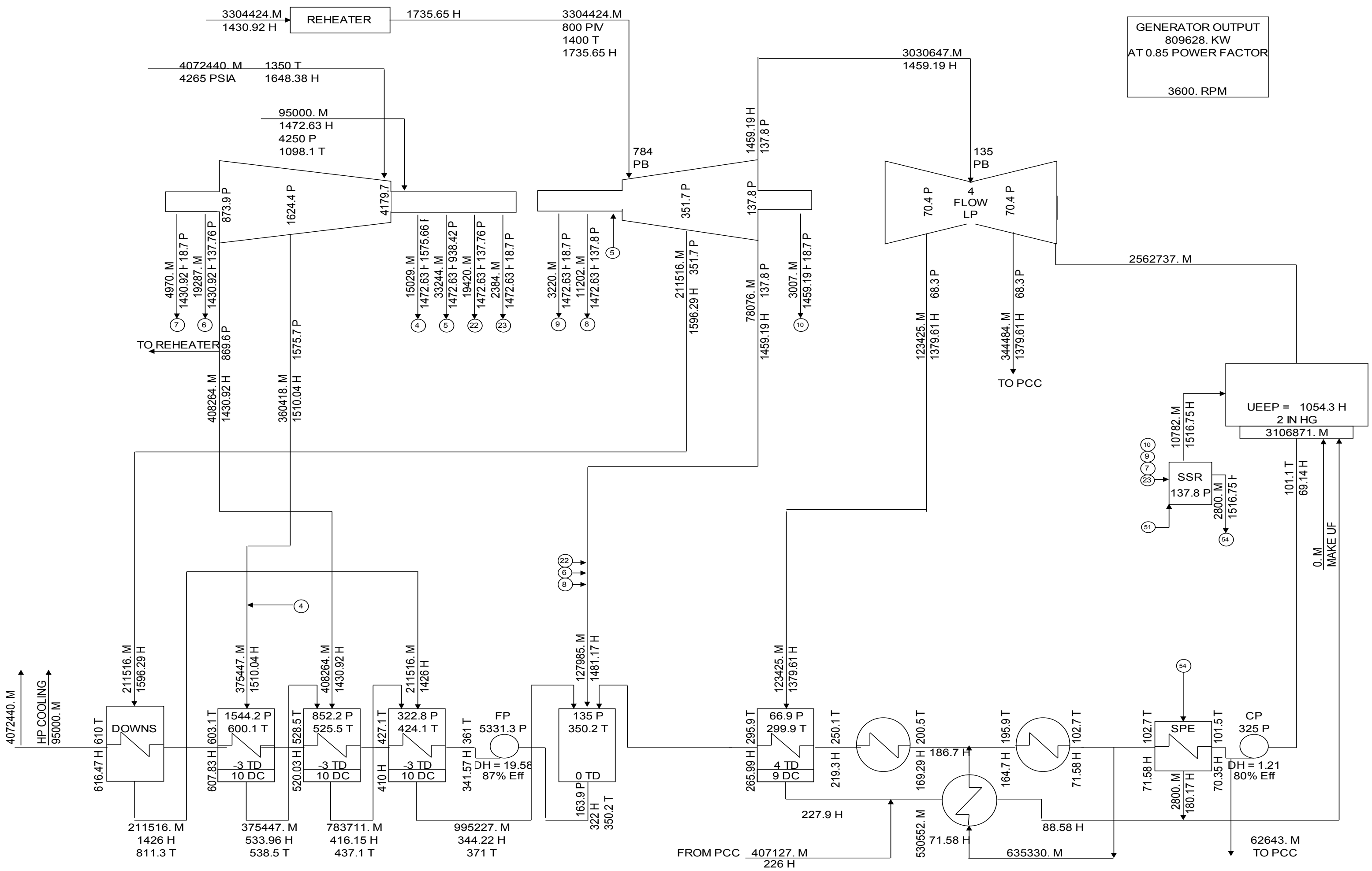

Figure A-7
4250 psig Inlet Pressure Steam Cycle Diagram, PCC

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