Results of the Technical and Economic Feasibility Analysis for a Novel Biomass Gasification-Based Power Generation System for the Forest Products Industry

Phase 1 Pre-Design Evaluation Final Report

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ABSTRACT

In 2001, the Gas Technology Institute (GTI) entered into Cooperative Agreement DE-FC26-01NT41108 with the U.S. Department of Energy (DOE) for an Agenda 2020 project to develop an advanced biomass gasification-based power generation system for near-term deployment in the Forest Products Industry (FPI). The advanced power system combines three advanced components, including biomass gasification, 3-stage stoker-fired combustion for biomass conversion, and externally recuperated gas turbines (ERGTs) for power generation.

The primary performance goals for the advanced power system are to provide increased self-generated power production for the mill and to increase wastewood utilization while decreasing fossil fuel use. Additional goals are to reduce boiler NOx and CO_2 emissions. The current study was conducted to determine the technical and economic feasibility of an Advanced Power Generation System capable of meeting these goals so that a capital investment decision can be made regarding its implementation at a paper mill demonstration site in DeRidder, LA.

Preliminary designs and cost estimates were developed for all major equipment, boiler modifications and balance of plant requirements including all utilities required for the project. A three-step implementation plan was developed to reduce technology risk. The plant design was found to meet the primary objectives of the project for increased bark utilization, decreased fossil fuel use, and increased self-generated power in the mill. Bark utilization for the modified plant is significantly higher (90-130%) than current operation compared to the 50% design goal. For equivalent steam production, the total gas usage for the fully implemented plant is 29 % lower than current operation. While the current average steam production from No.2 Boiler is about 213,000 lb/h, the total steam production from the modified plant is 379,000 lb/h. This steam production increase will be accomplished at a grate heat release rate (GHRR) equal to the original boiler design. Boiler efficiencies (cogeneration-steam plus air) is increased from the original design value of 70% to 78.9% due to a combination of improved burnout, operation with lower excess air, and drier fuel. For the fully implemented plant, the thermal efficiency of fuel to electricity conversion is 79.8% in the cogeneration mode, 5% above the design goal. Finally, self-generated electricity will be increased from the 10.8 MW currently attributable to No.2 Boiler to 46.7MW, an increase of 332%.

Environmental benefits derived from the system include a reduction in NOx emissions from the boiler of about 30 – 50% (90-130 tons/year) through syngas reburning, improved carbon burnout and lower excess air. This does not count NOx reduction that may be associated with replacement of purchased electricity. The project would reduce CO_2 emissions from the generation of electricity to meet the mill's power requirements, including 50,000 tons/yr from a net reduction in gas usage in the mill and an additional 410,000 tons/yr reduction in CO_2 emissions due to a 34 MW reduction of purchased electricity. The total CO_2 reduction amounts to about 33% of the CO_2 currently generated to meet the mills electricity requirement.

The overall conclusion of the study is that while significant engineering challenges are presented by the proposed system, they can be met with operationally acceptable and cost effective solutions. The benefits of the system can be realized in an economic manner, with a simple payback period on the order of 6 years. The results of the study are applicable to many paper mills in the U.S. firing woodwastes and other solid fuels for steam and power production.

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1.0 EXECUTIVE SUMMARY

1.1 INTRODUCTION

Paper mills use stoker boilers extensively to recover energy from wastewood, bark, and sludge. Because of the variability of feedstock moisture and ash content, the steam generation capacity of these boilers is generally limited by the ability to burn fuel on the grate. This limitation can be overcome by fuel and air staging, which improve combustion, reduce NOx emissions, and increase boiler efficiency through improved carbon burnout and operation at reduced excess air.

In 2001, the Gas Technology Institute (GTI) entered into Cooperative Agreement DE-FC26-01NT41108 with the U.S. Department of Energy (DOE) for an Agenda 2020 project to develop an advanced biomass gasification-based power generation system for near-term deployment in the Forest Products Industry (FPI). The advanced power system will be used in conjunction with existing wood waste fired boilers and flue gas cleanup systems. It combines three advanced components, including biomass gasification, 3-stage stoker-fired combustion for biomass conversion, and externally recuperated gas turbines (ERGTs) for power generation.

The primary performance goals for the advanced power system are to provide increased self-generated power production for the mill and to increase wastewood utilization while decreasing fossil fuel use. Additional goals are to reduce boiler NOx and CO_2 emissions. The objective of the current study is to determine the technical and economic feasibility of an Advanced Power Generation System capable of meeting these goals so that a capital investment decision can be made regarding its implementation at a paper mill demonstration site in DeRidder, LA.

1.2 PLANT DESIGN

The study revealed that the original system configuration, with all high-pressure air heated in the bark boiler, was not feasible due to space and heat limitations. An alternate design was developed using chemical and sensible heat in the gasifier syngas stream to heat air in a second, external heat exchanger between the gasifier and the boiler. Bark dryers were added to insure gasifier feed reliability and to make more high-level heat in the furnace available for air heating.

Preliminary designs and cost estimates were developed for all major equipment, boiler modifications and balance of plant requirements. All utilities are produced by the project. An implementation plan was developed to reduce technology risk. In implementation Step 1 the gasification island, the external air heater, one turbine, one HRSG and one bark dryer will be installed. In Step 2, the second bark dryer will be installed to dry bark for the boiler and in Step 3, the internal air heater will be installed in No. 2 Boiler with the second turbine and HRSG.

1.3 EXPERIMENTAL

A study was conducted with bark from the DeRidder mill to determine VOC emissions from drying bark in a low temperature dryer with flue gas exhausted through the boiler flue gas cleaning system. VOCs were found to be in the expected range and calculated emissions from the dryer are expected to be within permitted limits. An experimental study was conducted to test candidate air heater tube materials inside the furnace of No. 2 Boiler. Tube samples were placed in the boiler to test performance under both oxidizing and reducing conditions. Several of the materials tested have survived for several thousand hours and exposure is continuing.

1.4 RESULTS AND DISCUSSION

The modified plant design meets the primary objectives of the project for increased bark utilization, decreased fossil fuel use, and increased self-generated power in the mill. Bark utilization for the modified plant is significantly higher (90-130%) than current operation compared to the 50% design goal. For equivalent steam production, the total gas usage for the modified plant lower than current operation. For Implementation Steps 1 and 2, gas usage is reduced by 45-47% and for Step 3 by 29 %. While the current average steam production from No.2 Boiler is about 213,000 lb/h, the combined steam production from the boiler and HRSGs will be about 315,000 lb/h for Steps 1 and 2 and 379,000 lb/h in Step 3. In Step 3 this steam production increase will be accomplished at a grate heat release rate (GHRR) equal to the original boiler design. Boiler efficiencies for Steps 1 and 2 (steam) and Step 3 (cogeneration-steam plus air) are increased from the original design value of 70% to 74.2, 79.4 and 78.9, respectively, due to a combination of improved burnout, operation with lower excess air, and drier fuel. For the fully implemented case in Step 3, the thermal efficiency of fuel to electricity conversion is 79.8% in the cogeneration mode, 5% above the design goal. Finally, self-generated electricity will be increased from the 10.8 MW currently attributable to No.2 Boiler to 29.7MW in Steps 1 and 2 and 46.7MW in Step 3, increases of 175% and 332%, respectively.

Environmental benefits derived from the system include a reduction in NOx emissions from the boiler of about 30 – 50% (90-130 tons/year) through syngas reburning, improved carbon burnout and lower excess air. This does not count NOx reduction that may be associated with replacement of purchased electricity. The project would reduce CO_2 emissions from the generation of electricity to meet the mill's power requirements, including 50,000 tons/yr from a net reduction in gas usage in the mill and an additional 410,000 tons/yr reduction in CO_2 emissions due to a 34 MW reduction of purchased electricity. The total CO_2 reduction amounts to about 33% of the CO_2 currently generated to meet the mills electricity requirement.

1.5 CONCLUSIONS

The overall conclusion of the study is that while significant engineering challenges are presented by the proposed system, they can be met with operationally acceptable and cost effective solutions. The benefits of increased wastewood utilization, reduced fossil fuel usage and increased self-generated electric power can be realized in an economic manner, with a simple payback period on the order of 6 years. Environmental benefits will also be realized in the form of reduced emissions of NOx and CO_2.

The results of the study for the DeRidder site are applicable to many paper mills in the U.S. firing woodwastes for steam and power production. These materials can all be gasified and utilized for electric power generation with less technology risk and with equipment more consistent with current pulp and paper mill powerhouse operations than typical IGCC. If the advanced power system were applied to20 % of the FPI woodwaste-fired boiler capacity in a similar manner to the current study, the increase in self-generated power would be over 8,000 million kWh/yr, or about 13% of the total power purchased by the industry. Assuming purchased electricity is generated from coal and considering woodwaste fuel as CO_2 neutral, substituting self-generated biomass power for coal-based purchased power in 20% of the industry's woodwaste-fired boiler capacity would reduce CO_2 emissions to the environment by over 10 million tons/yr. With an average NOx production of about 0.25lb NOx/MMBtu from woodwaste combustion, application of the technology to 20% of the industry's woodwaste boiler capacity has the potential to reduce NOx emissions by over 10,000 tons/yr.

2.0 INTRODUCTION

2.1 BACKGROUND

Paper mills use stoker boilers extensively to recover energy from wastewood, bark, and sludge. Because of the variability of feedstock moisture and ash, the steam generation capacity of these boilers is generally limited by the ability to burn fuel on the grate, or, in other words, the boiler capacity is underutilized because of combustion limitations. It has been demonstrated conclusively that this limitation can be overcome by the application of fuel staging (reburning) and air staging, which improves combustion on the grate, reduces NOx emissions by up to 50%, and increases boiler efficiency through operation at reduced excess air and improved carbon burnout. The successful application of a 3-stage reburning combustion system was demonstrated in 1999 on a bark- and sludge-fired boiler using natural gas as the reburn fuel in an Agenda 2020 project sponsored by the U.S. Department of Energy (DOE)[1].

Agenda 2020 is the joint federal-industry program executed by DOE and the AF&PA in 1994. The overall objective of this project is to demonstrate the commercial applicability of a technology that advances the goals of Agenda 2020, providing new technology that will enable industry to achieve several important goals of Agenda 2020, namely to:

- Build technology leadership to advance United States global competitiveness
- Build energy self-sufficiency by taking advantage of biomass
- Meet demanding new environmental standards and requirements without incurring the predicted increases in cost for additional capital equipment, operations and energy consumption
- Protect the industry's most valuable resource (the forest) through sustainable management
- Continue providing high quality product
- Improve safety (an overarching goal of both industry and government)

In 2001, the Gas Technology Institute (GTI) entered into a Cooperative Agreement (DE-FC26-01NT41108) with DOE for another Agenda 2020 project to develop, demonstrate, and place in continuous operation an advanced biomass gasification-based power generation system suitable for near-term commercial deployment in the Forest Products Industry. The advanced power system is to be used in conjunction with, rather than in place of, existing wood waste fired boilers and flue gas cleanup systems and combines three advanced technological components including biomass gasification and 3-stage stoker-fired combustion for biomass conversion, and externally recuperated gas turbines for power generation. The system concept is intended to avoid the major hurdles of high-pressure gasification, i.e., high-pressure fuel feeding and ash removal, and hot gas cleaning that are typical for conventional IGCC power generation. It aims to also minimize capital intensity and technology risks in the initial demonstration and is intended to meet the immediate needs of the Forest Products Industry for highly efficient and environmentally friendly electricity and steam generation systems utilizing existing wood waste as the fuel resource.

2.2 PROJECT OUTLINE AND OBJECTIVES

The project is being conducted in four phases with the technical and economic feasibility of the proposed biomass gasification-based power generation approach verified in Phase 1 and detailed design, construction and demonstration operations to be completed in subsequent phases, contingent on a decision to proceed by the demonstration host site.

The overall objectives of this development project are:

- Development and field testing of the advanced system to increase utilization of the available wood waste fuels in the forest products industry
- Demonstration of stable and reliable operation along with the economic and environmental benefits of the biomass gasification-based power and steam generation system at full commercial scale
- Plant acceptance in continuous use within 6 years of project start
- Promotion and acceleration of the near-term acceptance of the developed and demonstrated technology in the Forest Products Industry

The major activities of Phase 1 include:

- Determining the information necessary to describe and quantify all anticipated environmental impacts of the project.
- Evaluating and selecting a paper mill as host site for the demonstration.
- Developing the preliminary design for the plant's Gasification Island
- Developing the preliminary design for the Indirect Air Heaters
- Evaluating the requirements for modifications to the existing bark-fired stoker boiler to accommodate an internal high pressure high temperature air heater (HTHP AH) and the addition of syngas reburn injection nozzles
- Evaluating and selecting an externally recuperated gas turbine (ERGT) and heat recovery steam generator (HRSG) for the plant's power island
- Evaluating the requirements for successful integration of the new power system into existing mill steam and electricity production and distribution systems
- Determining the balance-of-plant equipment requirements and the overall plant capital and operating costs

Specific energy and environmental performance goals for the advanced power system include:

- 50% or greater increase in wood waste usage for electricity and steam generation with a corresponding decrease in fossil fuel usage and/or purchased electricity
- Increased thermal efficiency of fuel to electricity conversion: up to 75% in cogeneration mode with 70% of energy utilized for electric power production, and up to 58% for electricity production only
- 50% or greater reduction in NO_X emissions (NO_X below 70 ppmvd at 3% O_2)
- 40% reduction in CO_2 emissions

2.3 PROJECT PARTICIPANTS

The Phase 1 evaluation effort was led by the GTI. The project team for the evaluation included Babcock Power Inc. (BPI) for the air heater and boiler modifications studies, Solar Turbines Inc for the gas turbine portion of the power island study, Carbona Corporation for the gasification island study, Nexant LLC, a Bechtel technology and consulting company, for the plant integration and economic evaluations, and Boise Cascade Corporation as the demonstration plant host site.

The Cooperative Agreement for the project was executed by DOE in August 2001. In September 2001, Boise Cascade Corporation agreed to provide a host site for the demonstration at their pulp and paper mill in DeRidder, Louisiana. This report discusses the results of the Phase 1 technical and economic evaluation study for the DeRidder host site.

3.0 PLANT DESIGN STUDY & ECONOMIC ANALYSIS

3.1 STUDY BASIS

3.1.1 Plant Design Basis

The following energy and environmental goals provide the basis for plant design:

- **Goal 1:** Increase wood waste usage for electricity and steam generation resulting in a corresponding decrease in fossil fuel usage and/or purchased electricity. This is accomplished by converting a portion of the wood waste normally burned in the stoker into syngas in a biomass gasifier, then using the syngas as reburning fuel in the stoker.
- **Goal 2:** Increase self-generation of electrical power thus reducing dependence on grid power. This is accomplished by employing an advanced gas turbine in a recuperative manner. The combustion air for the gas turbine is preheated by utilizing the excess sensible heat from the wood waste gasification and stoker combustion processes mentioned in Goal 1 above.
- **Goal 3:** Reduce plant NOx emission by using syngas as a reburning fuel in the stoker boiler as indicated in Goal 1 above.
- **Goal 4:** Reduce CO_2 emissions by increasing the utilization of biomass fuels and reducing the use of fossil fuel used for electricity and steam production.

3.1.2 Site Characteristics

The proposed project is envisaged to be a part of and located within the premises of the Pulp and Paper Mill owned by Boise Cascade Corporation Southern Operations and located at DeRidder, Louisiana. The project equipment is proposed to be located near #2 Bark Boiler. The characteristics of the site are shown in Table 3.1.2-1.

Table 3.1.2-1: Site Characteristics

Elevation:	207 ft. above mean sea level
Minimum average daily temperature:	45.8 ^{o}F (occurs in January)
Maximum average daily temperature:	81.2 ^{o}F (occurs in July)
Relative humidity:	Between 50 and 95; annual average 71.5%
Annual rainfall:	78.6 in. (2001 total)
Maximum wind velocity:	--
Seismic zone:	Zone III

3.1.3 Biomass Fuel Characteristics

The average characteristics of the biomass fuel are presented in Table 3.1.3-1.

Table 3.1.3-1: Biomass Fuel Characteristics

Composition	wt % (dry)
Carbon	54.0
Hydrogen	6.4
Oxygen	21.5
Nitrogen	0.2
Sulfur	0.0
Ash	5.5
Moisture, wt %	52.5
Fuel Consumption	
On-site generated	34.0% (by energy content)
Purchased	66.0% (by energy content)
Size Consistency	
Fines	17.7%
1/8 in.	20.2%
1/4 in.	38.6%
5/8 in.	11.0%
7/8 in.	4.7%
1 1/2 in.	7.8%
HHV (wet), Btu/lb	4526

3.1.4 Emissions Standards

EPA guidelines for a site that has the potential to emit more than 100 tons per year of regulated criteria air pollutants, more than 10 tons per year of a single hazardous air pollutant (HAP), and more than 25 tons per year of combined HAPs are presented below. The State of Louisiana is required to follow these guidelines or have separate but equivalent or more stringent regulations. The thrust for these guidelines and regulations is derived from prevention of significant deterioration (PSD) of the environment, that is, a site or a facility has a potential to emit more than 100 tons of a regulated criteria air pollutant.

Even though a facility may have been grand fathered and follow the pre 1979 regulations, future modification and upgrades will have to meet or exceed the above requirements. Under new guidelines EPA is requiring that facilities constructed before the Clean Air regulations of 1971, should implement a continuous emission monitoring system for accurate emission data.

Table 3.1.4-1 lists the most recent permitted utility boilers and their permit levels (boilers constructed between 1995-and 2000). The primary fuel is coal, but these regulations are also applied to coke, lignite and wood fired boilers. MSW and other hazardous waste burning facilities fall under separate category and are not addressed here.

Table 3.1.4-1: Permit Levels of Recent Utility Boilers

Boiler Sizes (Minimum Heat Input Capacity)	Fuel Types	Pollutants and Guidelines Limits (Lbs/MMBtu)
>250 MMBtu/hr	Primary – Coal or other solid fuel	SO_2 = 0.10
		PM = 0.025
		NO_X = 0.1~0.2
Monitoring Equipment		CEM

Details on emission regulations for stationary gas turbines are provided in 10CFR60 sub part GG.

Since primary fuel for the gas turbine system is natural gas, SO_2 and PM10 are not an issue and normally not specified.

3.2 PLANT DESCRIPTION

3.2.1 Overall Plant Description

A summary of the systems and equipment required to meet the objectives of the project is described in this section. A process flow diagram of the fully implemented gasification-based power generation system is presented in Figure 3.1.1-1. In the fully implemented system, an air-blown low-pressure fluidized bed gasifier is used to generate syngas from wood waste for use as reburn fuel in the existing bark-fired No. 2 Power Boiler. About one-third of the stoker's design fuel input is converted to syngas in the gasifier while the remaining two thirds is fed to the stoker.

Some of the air from the gasification air compressor is used into burn a portion of the syngas as it leaves the gasifier to increase its temperature from 850ºC (1562ºF) to 1204ºC (2200ºF) prior to entering the high-temperature high-pressure air heater AH-1. Heat is exchanged with a portion of a pressurized air stream from the compressor of an externally recuperated gas turbine (ERGT) generator, GT-1, preheating the air to about 760ºC (1400ºF) prior to the turbine combustor and cooling the syngas to 344ºC (650ºF) prior to entering the stoker boiler. The preheated air is further heated in the combustor to 852ºC (1565ºF) by combustion of natural gas prior to entering GT-1.

As the hot high-pressure air expands and cools through GT-1, 17 MWe of self-generated electric power is produced to displace power currently purchased from the grid. Vitiated air at about 18% O_2 is exhausted from GT-1 through a gasification air heater AH-2 and then to a heat recovery steam generator HRSG-1, where about 50,000 lb/h of additional 250 psig steam is generated for process use in the mill. A portion of the vitiated exhaust air from the HRSG is routed through a booster fan for use as undergrate and overfire (staged) combustion air in the boiler and the balance is discharged to the atmosphere through a stack.

The gasifier syngas is introduced through reburners into the stoker's primary combustion zone, creating a reducing zone immediately above the grate that destroys NO_x precursors and significantly reduces NO_x formation. The added gas flow also increases both heat release and mixing in the area immediately above the grate to greatly improve combustion stability and the ability to maintain boiler load through periods of increased fuel moisture. The syngas reburn staged-combustion arrangement eliminates the need to cofire fossil fuels continuously through auxiliary burners for this purpose. This allows low cost or negative cost CO_2-neutral biomass waste fuel to replace higher cost, CO_2-producing fossil fuels. Fuel- and air-staged combustion will also improve boiler efficiency through increased carbon burnout and reduced excess air at the boiler exit.

A second high-temperature high-pressure air heater, AH-3, is located in the upper furnace of the stoker boiler just below the furnace arch tip. Heat is exchanged with a portion of the pressurized air from a second externally recuperated gas turbine generator, GT-2, preheating the air prior to the turbine combustor. Air is further heated in the combustor in the same manner as GT-1, and an additional 17 MWe of self-generated electric power is produced. Vitiated air is exhausted from GT-2 to a second heat recovery steam generator, HRSG-2, where an additional 50,000 lb/h of 250 psig steam is generated for use in the mill. The exhaust air from HRSG-2 combines with the exhaust air from HRSG-1 prior to the combustion air booster fan.

Heat in the flue gas leaving the boiler economizer is used to dry the wet bark fuel from 52.5 % to 20% moisture upstream of the gasifier and stoker feeding systems. The boiler flue gas bypasses the existing tubular air heater and enters the bark dryers at about 318ºC (605ºF). The flue gas and evaporated moisture from the bark leave the dryer at a relatively low temperature of about 80ºC (175ºF), which limits volatile organic compounds (VOC) in the dryer exhaust. The dryer exhaust is returned to the boiler flue gas duct ahead of the dust collector for discharge through the existing ID fan, scrubber and stack.

Wet, hogged bark from the woodyard is diverted from the boiler bark conveyor to the dryers by a junction box and about two thirds of the dried bark is returned to the boiler conveyor in the same manner. The remaining bark is conveyed to a screening/sizing machine to reduce the fuel size sufficient for feeding through the gasifier feed lockhoppers.

3.2.2 Systems and Components of the Plant

The following systems and components are provided to accomplish the above functions:

- Biomass drying system with two dryers
- Gasification system
    - Fuel (dried bark) feeding
    - Limestone feeding
    - Gasifier
    - Gas feeding
    - Ash removal
    - Gasification process air supply
    - Product gas heating and cooling
    - Flaring

  - Product gas ducting
  - Nitrogen generation
- Gas turbine air preheating with external heat exchangers
- Boiler modification
  - Gas turbine air pre-heating in furnace with internal heat exchanger
  - Syngas injection to boiler
- Power generation with two externally recuperated gas turbine generators
- Heat recovery with two heat recovery steam generators (HRSG)
- Balance of plant
  - Cooling water supply
  - Compressed air supply
  - Fire protection
  - Electrical distribution
  - Bark conveying, screening/sizing and delivery
  - Natural gas supply

3.2.3 Implementation Steps

The implementation of the plant is envisaged to be in three successive steps, the basic elements and main features of which are summarized below:

Step 1: Installation of the gasification section, one of the two bark dryers and the first of two recuperated gas turbines. The process flow diagram for this step is shown in Figure 3.1.1-1. In this step:

- About one-third of the total wet wood waste is dried in a bark dryer and fed to the gasifier. The un-dried (wet) two-thirds is fed directly to the boiler.
- Heat for drying is provided by burning natural gas in the bark dryer.
- Exhaust gas from the dryer is routed to the boiler flue gas exhaust at the inlet to the dust collector.
- One externally recuperated 17 MWe gas turbine generator system (GT-1) is used to generate electricity.
- A portion of the combustion air for GT-1 is preheated in a syngas-to-air heat exchanger (AH-1).
- The air temperature to GT-1 is boosted in the turbine combustor by combustion of natural gas.
- The vitiated air (about 17.8% oxygen) exhaust from GT-1 is fed into a heat recovery steam generator (HRSG-1).
- Gasification air is preheated in an air-to-air heat exchanger (AH-2) between the GT-1 exhaust and HRSG-1
- About 65,000 lb/h of additional steam is generated in HRSG-1 for process use in the mill.
- Vitiated air exhaust from HRSG-1 is discharged to the atmosphere through a stack.

The Heat and Material Balance for the plant after Step 1 implementation is shown in Table 3.2.3-1

Step 2: Bypassing the tubular air heater to use boiler flue gas for bark drying. The process flow diagram for this step is shown in Figure 3.1.1-2. In this step:

- All wet wood waste to the gasifier and boiler are now dried. A second dryer is added to handle the full drying load. About one-third of the dried wood waste is fed to the gasifier, and two-thirds fed to the boiler.
- 17 MWe of electrical power continues to be generated via AH-1 and GT-1 along with about 65,000 lb/h of steam from HRSG-1.
- The boiler's tubular air heater is bypassed such that most or all of the flue gas from the economizer is used in the bark dryer.
- Exhaust gas from the dryer is routed into the boiler flue gas exhaust at the outlet of the economizer (at the inlet to the dust collector).
- Part of the vitiated air exhaust from HRSG-1 is used in the boiler as undergrate and overfire combustion air. The rest is discharged into the atmosphere through a stack.

The Heat and Material Balance for the plant after Step 2 implementation is shown in Table 3.2.3-2

Step 3: Installation of the second bark dryer, a second air heat exchanger, and the second recuperated gas turbine. The process flow diagram for this step is shown in Figure 3.1.1-3. In this step:

- A second externally recuperated 17 MWe gas turbine generator system (GT-2) is added for additional electricity generation.
- A portion of the combustion air for GT-2 is preheated in a platen heat exchanger inside the boiler (AH-3).
- The air temperature to GT-2 is boosted in the turbine combustor by combustion of natural gas.
- The vitiated air (about 18.1% oxygen) exhaust from GT-2 is fed into a heat recovery steam generator (HRSG-2).
- Part of the flue gas exhaust from HRSG-2 is used in the boiler as combustion air and over-fire air. The rest is discharged into the atmosphere through a stack.

The Heat and Material Balance for the plant after Step 3 implementation is shown in Table 3.2.3-3

Table 3.2.3-1 Heat and Material Balance for Implementation Step 1

English Units

		1	2	3	4	5	6	7	8	9	10	11	12	13
		Woodwaste to Gasifier	Air to Gasifier	Limestone to Gasifier	Purge Nitrogen	Ash from Gasifier	Raw Prod. Gas	Combustor Air	Heated Prod. Gas	Cooled Prod. Gas	Reburn Prod. Gas	Gasification Air	GT-1 Air to HE-1	Hot Air to Combustor
mass flow	lb/h	23621	40460	238	389	810	63422	14238	77660	77660	77660	54698	234389	234389
pressure	psia	14.7	43.5	14.7	43.5	14.7	29.0	177.9	26.1	23.2	15.2	43.5	194.0	178.0
temperature	°F	80	515	80	104	1472	1562	515	2199	651	651	285	661	1400
heat flow (HHV)	MMBtu/h	180.5	4.8	0.0	0.0	0.5	183.9	1.7	186.1	150.1	144.4	3.3	33.8	79.8
electric power	Mwe													
C	%w (dry)	54.00				1.8								
H	%w (dry)	6.40												
N	%w (dry)	0.21												
O	%w (dry)	33.85												
S	%w (dry)	0.00												
ash	%w (dry)	5.54				98.2								
moisture	%w	20.00												
HHV	Btu/lb (dry)	9528												
LHV	Btu/lb (dry)	8923												
		4724												
CaCO3	%w			77.0										
MgCO3	%w			14.0										
SiO2	%w			8.0										
Other inert material	%w			1.0										
CO	% v						16.94		14.66	14.66	14.66			
CO2	% v						10.75		9.56	9.56	9.56			
H2	% v						14.74		6.55	6.55	6.55			
H2O	% v						10.33		15.46	15.46	15.46			
CH4	% v						4.06		3.52	3.52	3.52			
N2	% v		79.0		98.0		42.93	79.00	50.08	50.08	50.08	79.0	79.0	79.0
C2H4	% v						0.04		0.021	0.021	0.021			
C6H6	% v						0.11		0.054	0.054	0.054			
H2S+COS	% v						0		0	0	0			
NH3+HCN	% v						0.11		0.09	0.09	0.09			
O2	%v		21.0		2.0			21				21.0	21.0	21.0
HHV	Btu/scf (wet)						149		107	107	107			
LHV	Btu/scf (wet)						137		100	100	100			
density	lb/scf						0.068		0.072	0.072	0.072			

SI Units

		1	2	3	4	5	6	7	8	9	10	11	12	13
		Woodwaste to Gasifier	Air to Gasifier	Limestone to Gasifier	Purge Nitrogen	Ash from Gasifier	Raw Prod. Gas	Combustor Air	Heated Prod. Gas	Cooled Prod. Gas	Reburn Prod. Gas	Gasification Air	GT-1 Air to HE-1	Hot Air to Combustor
mass flow	kg/s	2.98	5.10	0.03	0.05	0.10	7.99	1.79	9.79	9.79	9.79	6.89	29.53	29.53
pressure	bara	1.013	3	1.013	3	1.013	2	12.28	1.8	1.6	1.05	3	13.38	12.28
temperature	°C	27	268	27	40	800	850	268	1204	472	344	140.6	349	760
heat flow (HHV)	MJ/s	52.9	1.4	0.0	0.0	0.1	53.9	0.5	54.6	44.0	42.3	1.0	9.9	23.4
electric power	Mwe													
C	%w (dry)	54.00				1.8								
H	%w (dry)	6.40												
N	%w (dry)	0.21												
O	%w (dry)	33.85												
S	%w (dry)	0.00												
ash	%w (dry)	5.54				98.2								
moisture (as fed)	%w	20												
HHV	kJ/kg (dry)	22163												
LHV dry base	kJ/kg (dry)	20754												
CaCO3	%w			77.0										
MgCO3	%w			14.0										
SiO2	%w			8.0										
Other inert material	%w			1.0										
CO	% v						16.94		14.66	14.66	14.66			
CO2	% v						10.75		9.56	9.56	9.56			
H2	% v						14.74		6.55	6.55	6.55			
H2O	% v						10.33		15.46	15.46	15.46			
CH4	% v						4.06		3.52	3.52	3.52			
N2	% v		79.0		98.0		42.93	79.0	50.08	50.08	50.08	79.0	79.0	79.0
C2H4	% v						0.039		0.021	0.021	0.021			
C6H6	% v						0.106		0.054	0.054	0.054			
H2S+COS	% v						0		0	0.00	0.00			
NH3+HCN	% v						0.1091		0.094	0.09	0.09			
O2	%v		21.0		2.0			21.0				21.0	21.0	21.0
HHV wet base	kJ/m^{3}n (wet)						5833		4193	4193	4193			
LHV wet base	kJ/m^{3}n (wet)						5369		3920	3920	3920			
density	kg/m^{3}n						1.090		1.156	1.156	1.156			

[1] Includes cooling air

Table 3.2.3-1 Heat and Material Balance for Implementation Step 1 Continued

English Units

		14	15	16	17	18	19	20	21	22	24	25	26	37	40
		GT-1 Bypass Air	Gas to GT-1 Combustor	Total Air to GT-11	GT-1 Output	HSRG-1 Exhaust Air	Stoker Overfire Air	Undergrate Air	Woodwaste to Stoker	Stoker Bottom Ash	Wet Woodwaste	Gas to Dryer	Flue Gas to DC	Export Steam	Superheated Steam
mass flow	lb/h	456260	5997	732996	732996	732996	126824	167133	51750	641	91532	950	485759	57210	250000
pressure	psia	194.0	194.0	178.0	14.9	14.7	15.6	14.8	14.7	14.7	14.7	14.7	14.3	274.7	865
temperature	°F	661	80	1520	707	382	375	375	80	1004	80	80	336	409	825
heat flow (HHV)	MMBtu/h	65.7	143.3	294	130.4	69.5	11.1	14.6	234.3	1.5	414.3	22.7	117.2	68.8	351.1
electric power	Mwe				17.0										
C	%w (dry)								54.00	15.00	54.00				
H	%w (dry)								6.40		6.40				
N	%w (dry)								0.21		0.21				
O	%w (dry)								33.85		33.85				
S	%w (dry)								0.00		0.00				
ash	%w (dry)								5.54	85.00	5.54				
moisture	%w								52.50		52.5				
HHV	Btu/lb (dry)								9528		9528				
LHV	Btu/lb (dry)								8923		8923				
CaCO3	%w														
MgCO3	%w														
SiO2	%w														
Other inert material	%w														
CO	% v														
CO2	% v			1.46	1.46	1.46	0.03	0.03					11.22		
H2	% v														
H2O	% v			2.92	2.92	2.92	2.05	2.05					24.85	100	100
CH4	% v		100									100			
N2	% v	79.0		77.85	77.85	77.85	77.40	77.40					60.69		
C2H4	% v														
C6H6	% v														
H2S+COS	% v														
NH3+HCN	% v														
O2	%v	21.0		17.77	17.77	17.77	20.52	20.52					3.25		
HHV	Btu/scf (wet)														
LHV	Btu/scf (wet)														
density	lb/scf														

SI Units

		14	15	16	17	18	19	20	21	22	24	25	26	37	40
		GT-1 Bypass Air	Gas to GT-1 Combustor	Total Air to GT-1[1]	GT-1 Output	HSRG-1 Exhaust Air	Stoker Overfire Air	Undergrate Air	Woodwaste to Stoker	Stoker Bottom Ash	Wet Woodwaste	Gas to Dryer	Flue Gas to DC	Export Steam	Superheated Steam
mass flow	kg/s	57.49	0.76	92.36	92.36	92.36	15.98	21.06	6.52	0.08	11.53	0.12	61.21	7.21	31.50
pressure	bara	13.38	13.38	12.28	1.03	1.01	1.08	1.02	1.01	1.01	1.01	1.01	0.99	18.94	59.66
temperature	°C	349	27	827	375	194	191	191	27	540	27	27	169	209	441
heat flow (HHV)	MJ/s	19.3	42.0	86.2	38.2	20.4	3.2	4.3	68.7	0.5	121.4	6.7	34.4	20.2	101.6
electric power	Mwe				17.0										
											0.00				
C	%w (dry)								54.00	15.00	54.00				
H	%w (dry)								6.40		6.40				
N	%w (dry)								0.21		0.21				
O	%w (dry)								33.85		33.85				
S	%w (dry)								0.00		0.00				
ash	%w (dry)								5.54	85.00	5.54				
moisture (as fed)	%w								52.50		52.50				
HHV	kJ/kg (dry)								22163		22163				
LHV dry base	kJ/kg (dry)								20754		20754				
CaCO3	%w														
MgCO3	%w														
SiO2	%w														
Other inert material	%w														
CO	% v														
CO2	% v			1.46	1.46	1.46	0.03	0.03					11.22		
H2	% v														
H2O	% v			2.92	2.92	2.92	2.05	2.05					24.85	100.00	100.00
CH4	% v		100									100.00			
N2	% v	79.0		77.85	77.85	77.85	77.40	77.40					60.69		
C2H4	% v														
C6H6	% v														
H2S+COS	% v														
NH3+HCN	% v														
O2	%v	21.0		17.77	17.77	17.77	20.52	20.52					3.25		
HHV wet base	kJ/m^3n (wet)														
LHV wet base	kJ/m^3n (wet)														
density	kg/m^3n														

[1] Includes cooling air

Table 3.2.3-2 Heat and Material Balance for Implementation Step 2

English Units

		1	2	3	4	5	6	7	8	9	10	11	12	13	14
		Woodwaste to Gasifier	Air to Gasifier	Limestone to Gasifier	Purge Nitrogen	Ash from Gasifier	Raw Prod. Gas	Combustor Air	Heated Prod. Gas	Cooled Prod. Gas	Reburn Prod. Gas	Gasification Air	GT-1 Air to HE-1	Hot Air to Combustor	GT-1 Bypass Air
mass flow	lb/h	23621	40460	238	389	810	63422	14238	77660	77660	77660	54698	234389	234389	456260
pressure	psia	14.7	43.5	14.7	43.5	14.7	29.0	177.9	26.1	23.2	15.2	43.5	194.0	178.0	194.0
temperature	°F	80	515	80	104	1472	1562	515	2199	882	651	285	661	1400	661
heat flow (HHV)	MMBtu/h	180.5	4.8	0.0	0.0	0.5	183.9	1.7	186.1	150.1	144.4	3.3	33.8	79.8	65.7
electric power	MWe														
C	%w (dry)	54.00				1.8									
H	%w (dry)	6.40													
N	%w (dry)	0.21													
O	%w (dry)	33.85													
S	%w (dry)	0.00													
ash	%w (dry)	5.54				98.2									
moisture	%w	20.00													
HHV	Btu/lb (dry)	9528													
LHV	Btu/lb (dry)	8923													
		4724													
CaCO3	%w			77.0											
MgCO3	%w			14.0											
SiO2	%w			8.0											
Other inert material	%w			1.0											
CO	% v						16.94		14.66	14.66	14.66				
CO2	% v						10.75		9.56	9.56	9.56				
H2	% v						14.74		6.55	6.55	6.55				
H2O	% v						10.33		15.46	15.46	15.46				
CH4	% v						4.06		3.52	3.52	3.52				
N2	% v		79.0		98.0		42.93	79.00	50.08	50.08	50.08	79.0	79.0	79.0	79.0
C2H4	% v						0.04		0.021	0.021	0.021				
C6H6	% v						0.11		0.054	0.054	0.054				
H2S+COS	% v						0		0	0	0				
NH3+HCN	% v						0.11		0.09	0.09	0.09				
O2	%v		21.0		2.0			21				21.0	21.0	21.0	21.0
HHV	Btu/scf (wet)						149		107	107	107				
LHV	Btu/scf (wet)						137		100	100	100				
density	lb/scf						0.068		0.072	0.072	0.072				

SI Units

		1	2	3	4	5	6	7	8	9	10	11	12	13	14
		Woodwaste to Gasifier	Air to Gasifier	Limestone to Gasifier	Purge Nitrogen	Ash from Gasifier	Raw Prod. Gas	Combustor Air	Heated Prod. Gas	Cooled Prod. Gas	Reburn Prod. Gas	Gasification Air	GT-1 Air to HE-1	Hot Air to Combustor	GT-1 Bypass Air
mass flow	kg/s	2.98	5.10	0.03	0.05	0.10	7.99	1.79	9.79	9.79	9.79	6.89	29.53	29.53	57.49
pressure	bara	1.013	3	1.013	3	1.013	2	12.28	1.8	1.6	1.05	3	13.38	12.28	13.38
temperature	°C	27	268	27	40	800	850	268	1204	472	344	140.6	349	760	349
heat flow (HHV)	MJ/s	52.9	1.4	0.0	0.0	0.1	53.9	0.5	54.6	44.0	42.3	1.0	9.9	23.4	19.3
electric power	MWe														
C	%w (dry)	54.00				1.8									
H	%w (dry)	6.40													
N	%w (dry)	0.21													
O	%w (dry)	33.85													
S	%w (dry)	0.00													
ash	%w (dry)	5.54				98.2									
moisture (as fed)	%w	20													
HHV	kJ/kg (dry)	22163													
LHV dry base	kJ/kg (dry)	20754													
CaCO3	%w			77.0											
MgCO3	%w			14.0											
SiO2	%w			8.0											
Other inert material	%w			1.0											
CO	% v						16.94		14.66	14.66	14.66				
CO2	% v						10.75		9.56	9.56	9.56				
H2	% v						14.74		6.55	6.55	6.55				
H2O	% v						10.33		15.46	15.46	15.46				
CH4	% v						4.06		3.52	3.52	3.52				
N2	% v		79.0		98.0		42.93	79.0	50.08	50.08	50.08	79.0	79.0	79.0	79.0
C2H4	% v						0.039		0.021	0.021	0.021				
C6H6	% v						0.106		0.054	0.054	0.054				
H2S+COS	% v						0		0	0.00	0.00				
NH3+HCN	% v						0.1091		0.094	0.09	0.09				
O2	%v		21.0		2.0			21.0				21.0	21.0	21.0	21.0
HHV wet base	kJ/m^{3}n (wet)						5833		4193	4193	4193				
LHV wet base	kJ/m^{3}n (wet)						5369		3920	3920	3920				
density	kg/m^{3}n						1.090		1.156	1.156	1.156				

[1] Includes cooling air

Table 3.2.3-2 Heat and Material Balance for Implementation Step 2 Continued

English Units

		15	16	17	18	19	20	21	22	23	24	25	26	34	37	40
		Gas to GT-1 Combustor	Total Air to GT-11	GT-1 Output	HSRG-1 Exhaust Air	Stoker Overfire Air	Undergrate Air	Woodwaste to Stoker	Stoker Bottom Ash	Flue Gas from Stoker	Wet Woodwaste	Gas to Dryer	Flue Gas to DC	HRSG Export Air	Export Steam	Superheated Steam
mass flow	lb/h	5997	732996	732996	732996	165671	149582	27500	574	418926	86098	0	481321	417743	57210	250000
pressure	psia	194.0	178.0	14.9	14.7	15.6	14.8	14.7	14.7	14.5	14.7	14.7	14.3	0.0	274.7	865
temperature	°F	80	1520	707	382	375	375	80	1004	572	80	80	209.5	375	409	825
heat flow (HHV)	MMBtu/h	143.3	294	130.4	69.5	15.4	13.9	209.6	1.4	96.8	389.7	0.0	95.8	38.8	68.8	351.1
electric power	MWe			17.0												
C	%w (dry)							54.00	15.00		54.00					
H	%w (dry)							6.40			6.40					
N	%w (dry)							0.21			0.21					
O	%w (dry)							33.85			33.85					
S	%w (dry)							0.00			0.00					
ash	%w (dry)							5.54	85.00		5.54					
moisture	%w							20.00			52.5					
HHV	Btu/lb (dry)							9528			9528					
LHV	Btu/lb (dry)							8923			8923					
CaCO3	%w							5500								
MgCO3	%w															
SiO2	%w															
Other inert material	%w															
CO	% v															
CO2	% v		1.46	1.46	1.46	1.46	1.46			13.52			11.28	1.46		
H2	% v															
H2O	% v		2.92	2.92	2.92	2.92	2.92			15.15			23.76	2.92	100	100
CH4	% v	100										100				
N2	% v		77.85	77.85	77.85	77.85	77.85			69.09			61.95	77.85		
C2H4	% v															
C6H6	% v															
H2S+COS	% v															
NH3+HCN	% v															
O2	%v		17.77	17.77	17.77	17.77	17.77			2.24			3.01	17.77		
HHV	Btu/scf (wet)															
LHV	Btu/scf (wet)															
density	lb/scf															

SI Units

		15	16	17	18	19	20	21	22	23	24	25	26	34	37	40
		Gas to GT-1 Combustor	Total Air to GT-1[1]	GT-1 Output	HSRG-1 Exhaust Air	Stoker Overfire Air	Undergrate Air	Woodwaste to Stoker	Stoker Bottom Ash	Flue Gas from Stoker	Wet Woodwaste	Gas to Dryer	Flue Gas to DC	HRSG Export Air	Export Steam	Superheated Steam
mass flow	kg/s	0.76	92.36	92.36	92.36	20.87	18.85	3.47	0.07	52.78	10.85	0.00	60.65	52.64	7.21	31.50
pressure	bara	13.38	12.28	1.03	1.01	1.08	1.02	1.01	1.01	1.00	1.01	1.01	0.99	0.00	18.94	59.66
temperature	°C	27	827	375	194	191	191	27	540	300	27	27	99	191	209	441
heat flow (HHV)	MJ/s	42.0	86.2	38.2	20.4	4.5	4.1	61.4	0.4	28.4	114.2	0.0	28.1	11.4	20.2	101.6
electric power	MWe			17.0												
											0.00					
C	%w (dry)							54.00	15.00		54.00					
H	%w (dry)							6.40			6.40					
N	%w (dry)							0.21			0.21					
O	%w (dry)							33.85			33.85					
S	%w (dry)							0.00			0.00					
ash	%w (dry)							5.54	85.00		5.54					
moisture (as fed)	%w							20.00			52.50					
HHV	kJ/kg (dry)							22163			22163					
LHV dry base	kJ/kg (dry)							20754			20754					
CaCO3	%w															
MgCO3	%w															
SiO2	%w															
Other inert material	%w															
CO	% v															
CO2	% v		1.46	1.46	1.46	1.46	1.46			13.52			11.28	1.46		
H2	% v															
H2O	% v		2.92	2.92	2.92	2.92	2.92			15.15			23.76	2.92	100.00	100.00
CH4	% v	100										100.00				
N2	% v		77.85	77.85	77.85	77.85	77.85			69.09			61.95	77.85		
C2H4	% v															
C6H6	% v															
H2S+COS	% v															
NH3+HCN	% v															
O2	%v		17.77	17.77	17.77	17.77	17.77			2.24			3.01	17.77		
HHV wet base	kJ/m^{3}n (wet)															
LHV wet base	kJ/m^{3}n (wet)															
density	kg/m^{3}n															

[1] Includes cooling air

Table 3.2.3-3 Heat and Material Balance for Implementation Step 3

English Units

		1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
		Woodwaste to Gasifier	Air to Gasifier	Limestone to Gasifier	Purge Nitrogen	Ash from Gasifier	Raw Prod. Gas	Combustor Air	Heated Prod. Gas	Cooled Prod. Gas	Reburn Prod. Gas	Gasification Air	GT-1 Air to HE-1	Hot Air to Combustor	GT-1 Bypass Air	Gas to GT-1 Combustor	Total Air to GT-11	GT-1 Output	HSRG-1 Exhaust Air	Stoker Overfire Air	Undergrate Air
mass flow	lb/h	23621	40460	238	389	810	63422	14238	77660	77660	77660	54698	234389	234389	456260	5997	732996	732996	732996	184422	208055
pressure	psia	14.7	43.5	14.7	43.5	14.7	29.0	177.9	26.1	23.2	15.2	43.5	194.0	178.0	194.0	194.0	178.0	14.9	14.7	15.6	14.8
temperature	°F	80	515	80	104	1472	1562	515	2199	882	651	285	661	1400	661	80	1520	707	382	375	375
heat flow (HHV)	MMBtu/h	180.5	4.8	0.0	0.0	0.5	183.9	1.7	186.1	150.1	144.4	3.3	33.8	79.8	65.7	143.3	294	130.4	69.5	16.9	19.1
electric power	MWe																	17.0			
C	%w (dry)	54.00				1.8															
H	%w (dry)	6.40																			
N	%w (dry)	0.21																			
O	%w (dry)	33.85																			
S	%w (dry)	0.00																			
ash	%w (dry)	5.54				98.2															
moisture	%w	20.00																			
HHV	Btu/lb (dry)	9528																			
LHV	Btu/lb (dry)	8923																			
		4724																			
CaCO3	%w			77.0																	
MgCO3	%w			14.0																	
SiO2	%w			8.0																	
Other inert material	%w			1.0																	
CO	% v						16.94		14.66	14.66	14.66										
CO2	% v						10.75		9.56	9.56	9.56							1.46	1.46	1.36	1.36
H2	% v						14.74		6.55	6.55	6.55										
H2O	% v						10.33		15.46	15.46	15.46							2.92	2.92	2.72	2.72
CH4	% v						4.06		3.52	3.52	3.52					100					
N2	% v		79.0		98.0		42.93	79.00	50.08	50.08	50.08	79.0	79.0	79.0	79.0			77.85	77.85	77.92	77.92
C2H4	% v						0.04		0.021	0.021	0.021										
C6H6	% v						0.11		0.054	0.054	0.054										
H2S+COS	% v						0		0	0	0										
NH3+HCN	% v						0.11		0.09	0.09	0.09										
O2	%v		21.0		2.0			21				21.0	21.0	21.0	21.0			17.77	17.77	17.99	17.99
HHV	Btu/scf (wet)						149		107	107	107										
LHV	Btu/scf (wet)						137		100	100	100										
density	lb/scf						0.068		0.072	0.072	0.072										

SI Units

		1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
		Woodwaste to Gasifier	Air to Gasifier	Limestone to Gasifier	Purge Nitrogen	Ash from Gasifier	Raw Prod. Gas	Combustor Air	Heated Prod. Gas	Cooled Prod. Gas	Reburn Prod. Gas	Gasification Air	GT-1 Air to HE-1	Hot Air to Combustor	GT-1 Bypass Air	Gas to GT-1 Combustor	Total Air to GT-1[1]	GT-1 Output	HSRG-1 Exhaust Air	Stoker Overfire Air	Undergrate Air
mass flow	kg/s	2.98	5.10	0.03	0.05	0.10	7.99	1.79	9.79	9.79	9.79	6.89	29.53	29.53	57.49	0.76	92.36	92.36	92.36	23.24	26.22
pressure	bara	1.013	3	1.013	3	1.013	2	12.28	1.8	1.6	1.05	3	13.38	12.28	13.38	13.38	12.28	1.03	1.01	1.08	1.02
temperature	°C	27	268	27	40	800	850	268	1204	472	344	140.6	349	760	349	27	827	375	194	191	191
heat flow (HHV)	MJ/s	52.9	1.4	0.0	0.0	0.1	53.9	0.5	54.6	44.0	42.3	1.0	9.9	23.4	19.3	42.0	86.2	38.2	20.4	5.0	5.6
electric power	MWe																	17.0			
C	%w (dry)	54.00				1.8															
H	%w (dry)	6.40																			
N	%w (dry)	0.21																			
O	%w (dry)	33.85																			
S	%w (dry)	0.00																			
ash	%w (dry)	5.54				98.2															
moisture (as fed)	%w	20																			
HHV	kJ/kg (dry)	22163																			
LHV dry base	kJ/kg (dry)	20754																			
CaCO3	%w			77.0																	
MgCO3	%w			14.0																	
SiO2	%w			8.0																	
Other inert material	%w			1.0																	
CO	% v						16.94		14.66	14.66	14.66										
CO2	% v						10.75		9.56	9.56	9.56						0.00	1.46	1.46	1.36	1.36
H2	% v						14.74		6.55	6.55	6.55										
H2O	% v						10.33		15.46	15.46	15.46						0.00	2.92	2.92	2.72	2.72
CH4	% v						4.06		3.52	3.52	3.52					100					
N2	% v		79.0		98.0		42.93	79.0	50.08	50.08	50.08	79.0	79.0	79.0	79.0		0.00	77.85	77.85	77.92	77.92
C2H4	% v						0.039		0.021	0.021	0.021										
C6H6	% v						0.106		0.054	0.054	0.054										
H2S+COS	% v						0		0	0.00	0.00										
NH3+HCN	% v						0.1091		0.094	0.09	0.09										
O2	%v		21.0		2.0			21.0				21.0	21.0	21.0	21.0		0.00	17.77	17.77	17.99	17.99
HHV wet base	kJ/m^{3}n (wet)						5833		4193	4193	4193										
LHV wet base	kJ/m^{3}n (wet)						5369		3920	3920	3920										
density	kg/m^{3}n						1.090		1.156	1.156	1.156										

[1] Includes cooling air

Table 3.2.3-1 Heat and Material Balance for Implementation Step 3 Continued

English Units		21	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	37	38	39	40
		Woodwaste to Stoker	Stoker Bottom Ash	Flue Gas from Stoker	Wet Woodwaste	Gas to Dryer	Flue Gas to DC	GT-2 Air to HE-2	Hot Air to Combustor	GT-2 Bypass Air	Gas to GT-2 Combustor	Total Air to GT-21	GT-2 Output	HSRG-2 Exhaust Air	HRSG Export Air	HRSG-1 Steam	HRSG-2 Steam	Export Steam	Boiler Feedwater	Blowdown	Superheated Steam
mass flow	lb/h	38250	798	506318	104203	0	576069	320000	320000	370496	5158	732004	732004	732004	1072523	57210	58960	116170	252000	2000	250000
pressure	psia	14.7	14.7	14.5	14.7	14.7	14.3	208.0	178.0	194.0	194.0	178.0	14.9	14.7	14.7	274.7	274.7	274.7	1011.0	865.0	865
temperature	°F	80	1004	605	80	80	238	661	1400	661	80	1520	707	391	375	409	409	409	350	540	825
heat flow (HHV)	MMBtu/h	291.6	1.9	121.1	471.7	0.0	119.8	46.1	108.9	53.4	123.3	291	128.0	69.0	98.3	68.8	70.9	139.8	70.0	1.0	351.1
electric power	MWe												17.0								
C	%w (dry)	54.00	15.00		54.00																
H	%w (dry)	6.40			6.40																
N	%w (dry)	0.21			0.21																
O	%w (dry)	33.85			33.85																
S	%w (dry)	0.00			0.00																
ash	%w (dry)	5.54	85.00		5.54																
moisture	%w	20.00			52.5																
HHV	Btu/lb (dry)	9528			9528																
LHV	Btu/lb (dry)	8923			8923																
CaCO3	%w	7650																			
MgCO3	%w																				
SiO2	%w																				
Other inert material	%w																				
CO	% v																				
CO2	% v			13.48			11.36					1.26	1.26	1.26	1.36						
H2	% v																				
H2O	% v			15.05			23.92					2.52	2.52	2.52	2.72	100	100	100	100	100	100
CH4	% v					100					100										
N2	% v			69.13			61.81	79.0	79.0	79.0		78.00	78.00	78.00	77.92						
C2H4	% v																				
C6H6	% v																				
H2S+COS	% v																				
NH3+HCN	% v																				
O2	%v			2.33			2.92	21.0	21.0	21.0		18.22	18.22	18.22	17.99						
HHV	Btu/scf (wet)																				
LHV	Btu/scf (wet)																				
density	lb/scf																				

SI Units		21	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	37	38	39	40
		Woodwaste to Stoker	Stoker Bottom Ash	Flue Gas from Stoker	Wet Woodwaste	Gas to Dryer	Flue Gas to DC	GT-2 Air to HE-2	Hot Air to Combustor	GT-2 Bypass Air	Gas to GT-2 Combustor	Total Air to GT-2[1]	GT-2 Output	HSRG-2 Exhaust Air	HRSG Export Air	HRSG-1 Steam	HRSG-2 Steam	Export Steam	Boiler Feedwater	Blowdown	Superheated Steam
mass flow	kg/s	4.82	0.10	63.80	13.13	0.00	72.58	40.32	40.32	46.68	0.65	92.23	92.23	92.23	135.14	7.21	7.43	14.64	31.75	0.25	31.50
pressure	bara	1.01	1.01	1.00	1.01	1.01	0.99	14.34	12.28	13.38	13.38	12.28	1.03	1.01	1.01	18.94	18.94	18.94	69.72	59.66	59.66
temperature	°C	27	540	318	27	27	114	349	760	349	27	827	375	199	191	209	209	209	177	282	441
heat flow (HHV)	MJ/s	85.5	0.6	35.5	138.2	0.0	35.1	13.5	31.9	15.6	36.1	85.2	37.5	20.2	28.8	20.2	20.8	41.0	20.4	0.3	101.6
electric power	MWe												17.0								
					0.00																
C	%w (dry)	54.00	15.00		54.00																
H	%w (dry)	6.40			6.40																
N	%w (dry)	0.21			0.21																
O	%w (dry)	33.85			33.85																
S	%w (dry)	0.00			0.00																
ash	%w (dry)	5.54	85.00		5.54																
moisture (as fed)	%w	20.00			52.50																
HHV	kJ/kg (dry)	22163			22163																
LHV dry base	kJ/kg (dry)	20754			20754																
CaCO3	%w																				
MgCO3	%w																				
SiO2	%w																				
Other inert material	%w																				
CO	% v																				
CO2	% v			13.48			11.36					1.26	1.26	1.26	1.36						
H2	% v																				
H2O	% v			15.05			23.92					2.52	2.52	2.52	2.72	100.00	100.00	100.00	100.00	100.00	100.00
CH4	% v					100.00					100										
N2	% v			69.13			61.81	79.00	79.00	79.00		78.00	78.00	78.00	77.92						
C2H4	% v																				
C6H6	% v																				
H2S+COS	% v																				
NH3+HCN	% v																				
O2	%v			2.33			2.92	21.00	21.00	21.00		18.22	18.22	18.22	17.99						
HHV wet base	kJ/m^{3}n (wet)																				
LHV wet base	kJ/m^{3}n (wet)																				
density	kg/m^{3}n																				

[1] Includes cooling air

3.3 MAJOR PLANT AREAS

Descriptions of the major plant areas, which include biomass drying, biomass gasification, modification of the existing biomass boiler, power generation, and preheating of gas turbine combustion air are given in this section.

3.3.1 Biomass Drying System

System Description

Preparation of fuel for gasifier includes the drying of bark fuel. The fuel drying system reduces fuel moisture content from 52.5% to 20%. The dryer system includes two directly heated drum dryers. The drum dryers are of same size and capacity and will be installed in two steps. The dryers are integrated with the boiler flue gas system utilizing flue gas for the drying process and returning the exhaust gas to the dust removal system of the boiler. The dryer system is at negative pressure since it is upstream of the boiler's ID fan.

In the first implementation step the heat for drying is generated by burning natural gas and utilizing a small amount of recirculated flue gas from the boiler. Wet fuel is fed through an air lock (rotary feeder) at the inlet of the rotary drum. The drum is operated at negative pressure maintained by the ID fan of the boiler. Big particles of dried fuel drop out from the drum into a Drop-Out Box, which is equipped with a reversible screw conveyor to discharge the dried product. The small particles are conveyed with the gas flow to twin cyclones where the rest of dried fuel is separated from the gas stream. The cyclones are equipped with screw conveyors and airlock to remove the rest of dried fuel. The exhaust flange of the cyclone is connected to the dust removal system of the boiler, where the fine dust is removed from the gas flow.

In the second implementation step a second dryer will be installed. In this phase the two parallel dryers would provide dry feedstock to the gasifier and to the boiler. In this case both dryers would utilize flue gas from the boiler economizer as drying medium. Both dryers' exhausts are directed to the dust removal system of the boiler. The second dryer process is as described for the first dryer.

Design Basis

The design basis of the fuel dryers is summarized in Table 3.3.1-1.

Process Parameters

Technical and pricing information was received in quotations for the bark dryers from a U.S. (MEC) and a Swedish (Torkapparater) vendor. Process parameters for one (1) dryer are shown in the Table 3.3.1-2 below, based on MEC proposal data. Data shown for Step 1 is for base load operation of the Gasification Plant. Step 2 is 50% of the total requirement of the Gasification Plant and Stoker boiler at base load.

Table 3.3.1-1 Fuel Dryer Design Basis

Fuel Dryers Design Basis for Steps 1& 2
Material to be dried: bark

<u>Bark size Fraction</u>

Size,	Wt %
1-1/2in (38mm)	7.8
7/8in (22.2mm)	4.7
5/8in (15.9mm)	11.0
1/4in (6.35mm)	38.6
1/8in (3.2mm)	20.2
Fines	17.7

Product moisture into dryer, wet basis	52.5 %w	
Product moisture out of dryer, wet basis	20 %w	
Ambient air temperature at dryer air intake*	59 °F (15 °C)	
Product temperature at dryer air intake*	59 °F (15 °C)	
Bulk density for volumetric designs		
A. Oven dry basis	10.0 lb/ft^3 (160 kg/m^3)	
B. Wet as fed basis	21.1 lb/ft^3 (338 kg/m^3)	
	Step 1	Step 2
Wet fuel input, lb/h / kg/s	39685 /5.0	54015 / 6.81
Available flue gas from boiler, lb/h / kg/s	83351 /10.5	555670 / 70.0
Available boiler flue gas temperature, °F /°C	400 / 204	640/ 338
Available boiler flue gas composition		

CO_2	17.6	%v
O_2	4.3	%v
H_2O	16.3	%v
N_2	61.8	%v

* Based on vendor quotation, process ambient temperature is 66 °F

Table 3.3.1-2 Fuel Dryer Process Parameters

Dryer Process Parameters

	Step 1	Step 2
Wet fuel input lb/h / kg/s	39685 / 5.0	54015 / 6.81
Water evaporation rate, lb/h / kg/s	16121 / 2.03	21944 / 2.77
Dryer overall heat demand, MMBtu/h / MJ/s	2762 / 8.1	3475 / 10.18
Dried product mass flow rate, lb/h / kg/s	3563 / 2.97	32071 / 4.04
Dried product temperature, °F / °C	154 / 68	154 / 68
Exhaust gas flow, lb/h / kg/s	271 064 / 34.15	261490 / 32.95
Estimated cool gas temp. at dryer outlet, °F / °C	175 / 79	175 / 79
Dust in exhaust gas, lb/scf / g/m^{3}n	0.0028 / 45	0.0028 / 45
Expected uncontrolled VOC emission, lb/h / kg/h	41.0 / 18.6	63.5 / 28.8
Flue gas flow rate from boiler, lb/h / kg/s	82431/ 10.39	225 700 / 28.44
Boiler flue gas temperature, °F / °C	399 / 204	640 / 338
Natural gas consumption, lb/h / kg/s	1074 / 0.135	0 / 0
Total air supply to dryer, lb/h / kg/s	171437 / 21.6	13846 / 1.74
Estimated electric power demand, kW / kW	86 / 86	82 / 82
Pressure drop on dryer system, in. wg / mbar	20 / 51	20 / 51

3.3.2 Gasification Plant

3.3.2.1 Fuel Feeding System

The description of the Fuel Feeding System refers to the process flow diagram of the Gasification Plant (Figure 3.3.2-1). The two Fuel Feeding Systems A and B are identical, and note that limestone feeding occurs only in conjunction of Fuel Feeding System A.

System Description

The function of the Fuel Feeding System is to feed woodwaste fuel from atmospheric pressure to system pressure. The fuel is mainly bark of particle size smaller then 1.5 in/38 mm. The mean particle size is about 0.25 in/ 6.5 mm. The moisture content of biomass as fed is about 20%.

The fuel feeding system is a rotary valve / surge hopper system. The feeding system includes two identical feeding lines of equal capacity. Each feeding line has a capacity of 80% of total fuel feed, i.e. providing a total feeding capacity of 2x80%. Each of the fuel feeding lines includes one Weigh Silo, one rotary valve feeding screw, one rotary valve, one surge hopper and one metering screw.

Fuel is conveyed from the weigh silo of the fuel feeding lines. The weigh silos are equipped with vent-filters to prevent dust emission. The fuel is discharged from the weigh silos with discharge devices (live bottom) and fed with the rotary valve feeding screw to surge hoppers through rotary valves and filling valves. The rotary valve feeding screw is equipped with variable speed drive and it can be operated in reverse direction when the weigh silo has to be emptied. The function of the rotary valve is to keep the pressure in the surge hopper.

The fuel filling sequence based on level measurement in the weigh silo and surge hopper operates the filling valves. The isolation valve operation is interlocked with gasifier/surge hopper pressure difference and feeding screw temperature.

From the surge hopper the fuel is discharged through a screw type live bottom to the metering screws and are fed to the gasifier through the feeding screws using inert gas purging. The metering screws are equipped with variable speed drive (gasifier fuel feed rate control). The feeding screw is equipped with constant speed drive and water-cooled shaft. The Fuel Feeding System can be isolated from the gasifier by an isolating valve upstream of the feeding screws.

Design Basis

The gasifier fuel specification at as fed conditions is shown in Table 3.3.2-1.

3.3.2.2 Limestone Feeding System

The description of the Limestone Feeding System refers to the process Flow Diagram of the Gasification Plant (Figure 3.3.2-1)

Table 3.3.2-1. Fuel Specification

Fuel type:	bark						
Bark Size:	Fraction						
7.8	%w	1-1/2	in	38	mm		
4.7	%w	7/8	in	22.2	mm		
11.0	%w	5/8	in	15.9	mm		
38.6	%w	1/4	in	6.35	mm		
20.2	%w	1/8	in	3.2	mm		
17.7	%w	Fines		Fines			
As fed moisture content			20	%w			
Bulk density for volumetric designs				17.4-lb/ft^3		280	kg/m^3
Fuel temperature		154	°F	68	°C		
Fuel mass flow rate		23620	lb/h	2.98	kg/s		

Limestone System Description

Limestone is used as bed material in the fluidized bed gasifier. Limestone is fed by a separate feeding line into the gasifier, through the feeding screw of fuel feeding line A. The function of the Limestone Feeding System is to pressurize limestone from atmospheric pressure to system pressure at low temperature and feed it into the fluidized bed of the gasifier.

The Limestone Feeding System includes limestone weigh silo, lock hopper, surge hopper and metering screw. Limestone is transported pneumatically to the weigh silo (DFS-SILO1), which is equipped with vent-filter (DFS-EXHHD1). The weigh silo is equipped with a bottom discharge system (DFS-CNV2, variable elevation cone, hydraulically operated, vendor specific), which controls the limestone feed to the lock-hopper (DFS-LH1). When the lock-hopper is filled, the filling valve will be closed. The lock-hopper will be pressurized up to surge hopper pressure using the nitrogen from the process nitrogen network. When the pressure of the lock-hopper is about the same as the pressure of the surge hopper (DFS-LH2), the pressure equalizing valve/line opens between these two hoppers and pressure will be equalized. The closing valve in the bottom of the lock-hopper will be opened and limestone drops by gravity into the surge hopper. When all the limestone is transferred from the lock-hopper into the surge hopper, the lock-hopper closing valve will close and the empty lock-hopper will be depressurized. The valves are operated automatically according the limestone filling sequence based on level measurements in the lock-hopper and the surge hopper.

The Limestone is fed by the variable speed limestone metering screw (DFS-CNV1) from the surge hopper to the fuel feeding screw A, where the limestone will be mixed with fuel and fed into the gasifier.

Design Basis

Limestone feed specification is shown in Table 3.3.2-2.

Table 3.3.2-2. Limestone Feed Properties

Composition:							
$CaCO_3$		77 %w					
$MgCO_3$		14 %w					
SiO_2		8 %w					
Moisture		air dry					
Other inert material		1 %w					
Bulk density		87 lb/cf		$(1400 kg/m^{3})$			
Particle size distribution							
Mesh	-18	-30	-40	-50	-70	-100	-140
%W	100.0	88.6	62.0	37.0	19.5	9.1	3.6
Limestone temperature		66°F	(19°C)				
Limestone feed rate		238lb/h (0.03kg/s)					

The Limestone Feeding System is designed for 15 min feeding time in an hour (no continuous feeding required due to small limestone mass flow) at the base load operation of the gasifier.

3.3.2.3 Gasification System

The description of the Gasification System refers to the process Flow Diagram of the Gasification Plant (Figure 3.3.2-1).

System Description

The Gasification System includes Gasifier reactor, Cyclone, Start-up Heater and the Gasifier Gas Feeding System. The function of the gasifier is to convert solid feedstock (bark and wood waste) to Low-Calorific Value (LCV) product gas. The gasifier is a low pressure, bubbling fluidized bed gasification system.

The feedstock fed into the gasifier will dry and de-volatilize in the fluidized bed, the remainder char is gasified and partly burnt to maintain sufficient gasification temperature. The product gas contains

- Combustible components including CO, H_2 and CH_4
- Inert components including N_2, CO_2 and H_2O
- Trace contaminants including H_2S, COS, NH_3, HCN, HCl, vapor phase alkalis.

The bulk (about 40%) of the gas is nitrogen due to air blown gasification. The raw gas leaving the gasifier cyclone also contains entrained solid particles.

The Gasifier Reactor includes the following parts:

- Gasifier reactor pressure vessel with ports for gas and solids feed and removal, for measuring systems and manholes
- Gasifier reactor multi-component refractory lining of varying thickness along the height of the reactor
- Gas distributor system including the grid and ash removal pipe.

The Cyclone includes the following parts:

- Cyclone pressure vessel with connection to gasifier
- Refractory lining inside the pressure vessel
- Dipleg from the cyclone bottom to the gasifier (external)

The Start-up Heater includes the following parts:

- Heater pressure vessel with connections to the gasifier and ports for gas feed and measuring systems
- Refractory lining of the heater
- Heater burner

The Gasifier Gas Feeding System includes the following parts:

- Air piping from the Process Air System to the gasifier
- Air distribution manifold
- Air control valve systems
- Steam piping from pressure reduction to gasifier
- Steam control valve system
- Nitrogen piping to from nitrogen manifold to gasifier
- Nitrogen valves.

Gasifier Reactor

The Gasifier Reactor accommodates the fluidized bed and the freeboard area. The fluidized bed area is a bubbling fluidized bed of inert bed material (limestone), char and ash. The disengaging/freeboard area is a suspension of char and ash elutriated from the fluidized bed. The gasifier operates at 1560°F / 850 °C temperature and at 29.0 psia / 2.0 Bara pressure.

The reactor pressure vessel is of dual diameter. The refractory lining reduces the inside diameter of the reactor determining bed and freeboard diameters. The fluidized bed operates at between 3.0-4.3 ft/s / 1.0-1.3 m/s superficial velocity while the gas velocity decreases in the freeboard, returning the bulk of the elutriated fine particles to the fluidized bed and providing longer residence time for gases and solids. The height (volume) of the fluidized bed is determined mainly by the reactivity of the fuel.

Biomass is fed by water-cooled feeding screws to the lower area of the fluidized bed from the surge hoppers of the feeding lines. The elutriated particles are separated from the product gas stream by the cyclone and are returned to the fluidized bed area. The limestone bed material is fed into the gasifier bed area through fuel feeding screw A.

The gasification air (and steam, if any) is fed by the Gasifier Gas Feeding system to the gasifier reactor through the gas distributor. The gas distributor includes the grid and the bed material discharge system. The grid is of conical design equipped with horizontal nozzles. The bottom of the conical grid is connected to the ash removal pipe/classifier (air flows in counter flow with ash in the ash removal pipe carrying back particles of smaller than a certain particle size to the fluidized bed) where the gasifier ash discharge takes place. The ash removal pipe/classifier connects the gasifier bed area to the ash discharge system. The bed material (limestone and ash) is removed from the gasifier through the ash removal pipe and classifier via the cooling screw. Product gas exits the gasifier through the top of the pressure vessel and enters the cyclone.

Cyclone

The bulk of the elutriated particulate matter from the gasifier is separated from the raw gas flow in a cyclone separator. The separated dust is returned through the cyclone dipleg to the fluidized bed where the returned carbon will further gasify.

The cyclone is a refractory lined pressure vessel. The cyclone is connected to the top of the gasifier pressure vessel. The exit of the cyclone is connected through a refractory lined gas duct and air injection system to the gas cooler heat exchanger. The cyclone dipleg is arranged externally (outside gasifier pressure vessel) and connected to the gasifier pressure vessel at bottom of the bed area.

Start-up Heater

At startup the gasification system is heated-up by a start-up heater to sufficient temperature for the combustion of startup fuel and switchover gasification.

The start-up heater is a horizontal refractory lined pressure vessel equipped with the start-up burner. The start-up heater is connected to the ash removal pipe/classifier. The air introduced through the ash removal pipe is supplied through the start-up heater. The start-up burner is operated so that the heater exit temperature is controlled between 1470-1650 °F / 800-900 °C. Once gasification temperature is achieved the startup burner is shut down and only air flows through the start-up heater at 635 °F / 335 °C temperature.

The start-up burner is located at one end of the horizontal pressure vessel. The other end is the gas exit connected to the ash removal pipe/classifier joint.

Start-up Heater Burner

The description of the Start-up Heater Burner refers to the process Flow Diagram of the Gasification Plant (Figure 3.3.2-1).

The start-up heater burner is a gas burner used for gasification system heat-up during plant start-up. The start-up burner is connected to one end of the start-up burner chamber, which is a refractory lined, horizontal pressure vessel. The other end of the burner chamber is connected to the reactor ash discharge pipe of the gasifier.

The start-up burner is operated so that the heater exit temperature is controlled to 1470-1650°F / 800-900 °C with mixing air. The burner operates normally at atmospheric pressure, but it can be ignited and operated up to 29 psia / 2 Bara pressure. Since the heater vessel is connected to the gasifier reactor, ash removal air flows through the start-up heater at 635 °F / 335 °C temperature serving also as purge gas for the burner and the burner chamber. The ash removal air is fed through the mixing air nozzle.

3.3.2.4 Gas Feeding System

Gasification air is supplied at 43.5 psia / 3 Bara and 635 °F / 335 °C. The airflow is divided in the following streams through the air distribution manifold:

- Grid air, which maintains fluidization in the bed. Grid air enters the fluidized bed through nozzles of the grid.
- Ash removal air which controls bed material and ash removal from the bed. The ash removal air is fed via the start-up heater, which is connected to the mid section of the ash

removal pipe.

- Air for the partial combustion of product gas to maintain 2200 °F / 1204 °C temperature before GT air heater. This airflow is fed through nozzles to the combustions section down stream gasifier cyclone.
- Air for the startup heater burner. This airflow is fed separately to the startup burner as combustion air when it is operated and purge air when it is out of operation.

All air feeding lines are equipped with flow control valves and check valves.

Steam of (at least) 58 psia / 4 bara pressure is fed to the grid air line in the case of excess grid temperature or fluidized bed temperature or during emergency shutdown of the gasifier. Steam is fed in mixture of the grid airflow. Steam flow is controlled in accordance with the grid airflow. The steam line is equipped with flow control valve, check valve and blow down line for condensate removal.

Nitrogen is fed to the gasifier during emergency shutdown or excess temperature in the ash removal pipe. Nitrogen is supplied to the air distribution manifold or directly to ash removal air line. The nitrogen lines are equipped with flow control valves and check valves.

Design Basis

The process design data for the Gasifier System is summarized in Table 3.3.2-3.

Table 3.3.2-3 Gasifier System Design Basis

Ambient temperature	66°F	19°C
Fuel feed rate (AF, 20% moisture)	23621 lb/h	2.98kg/s
Fuel heat input	180.4 MMBtu/h	52.8 MJ/s
Product gas generation	63422 lb/h	7.99 kg/s
Gasification temperature	1560 °F	850 °C
Gasification pressure	29.0 psia	2.0 bara
Fluidization velocity	3.9.ft/s	1.2 m/s
Gasification air temperature	635 °F	335 °C

3.3.2.5 Ash Removal System

The description of the Gasifier Ash Removal System refers to the Flow Diagram of the Gasification Plant (Figure 3.3.2-1).

System Description

The function of the Gasifier Ash Discharge System is to remove and cool the discharged solids (bed material consisting spent limestone and fuel ash) from the gasifier at system pressure and high temperature. The solids are removed through a water-cooled screw and lock-hoppers. The bed material (ash and spent limestone) is removed from the gasifier system through the ash discharge pipe at the bottom of the gasifier pressure vessel. Reliable and continuous gasifier ash removal is essential to ensure the stable operation of the gasifier. The gasifier ash removal system consists of one cooling screw, one surge-hopper and one lock-hopper system for depressurizing the ash. The cooling screw is cooled by cooling water. The lock hopper system is pressurized and made inert by using nitrogen.

The bed material drops out from the gasifier fluidized bed through the ash removal pipe, in counter flow with a controlled airflow, into the ash cooling screw. The ash cooling screw is water-cooled, both shaft and jacket. In the cooling screw the bed material cools down from 1562 °F / 850 °C to 482 °F / 250 °C. The cooling screw is followed by the ash surge hopper, which enables the continuous operation of the cooling screw during ash depressurization. The isolation valves between the surge and lock-hopper opens after the lock-hopper is pressurized and the pressure is equalized. The ash drops from the buffer hopper into the lock-hopper. Then the isolation valve will close and the pressure of the lock-hopper will be let down to atmospheric pressure. After opening the valves between the lock-hopper and weigh silo, the ash drops into the weigh silo. After closing the valve between the lock-hopper and the weigh silo, the lock-hopper will be immediately pressurized and connected to the surge hopper. The ash removal sequence is operated based on level measurement in the lock-hopper.

Design Basis

The properties of reactor ash (mixture of bed material, ash and char) are shown in Table 3.3.2-4.

Table 3.3.2-4 Ash Properties

Density	62-75 lb/cf	1000-1200 kg/m3
Moisture (less than)	1 %w	1 %w
Material temperature	1562 °F	850 °C
Particle size	0.004-0.079 in	0.1-2 mm
Maximum particle size	1.0 in	25 mm
Ash discharge rate	810 lb/h	0.1 kg/s

The gasifier ash removal system includes one removal line of 120 % base load capacity.

3.3.2.6 Process Air System

The description of the Process Air System refers to the process Flow Diagram of the Gasification Plant (Figure 3.3.2-1).

System Description

The function of the Process Air System is to provide air for the gasifier at required pressure.

The gasification air is taken from atmosphere and compressed in the process air compressor to 43.5 psia / 3 bara pressure to overcome the pressure drop of the gas feeding system, air preheater and the entire gasification system down to the gas injection system. A two stage intercooled centrifugal air compressor is applied. The compressor is equipped with air intake filter; blow down valve and silencer at discharge. The compressor is equipped with variable speed drive for part load control and surge protection performed by the control system based on the measurement of motor current, air flow and discharge pressure. The compressor has no after cooler therefore the supply temperature of air is about 217 °F / 103°C. The air is further heated to 635 °F / 335°C in the last gas cooler AH-2 heat exchanger by turbine exhaust gas. After the gas cooler the airflow is partly directed to the gasifier gas feeding system including air distribution and control valves and partly fed to the partial gas combustion nozzles.

Design Basis

The process air compressor is electric motor driven, two-stage, intercooled centrifugal compressor. The compressor has no aftercooling. The design parameters of the process air compressor (PAS-C1) are summarized in the Table 3.3.2-5.

Table 3.3.2-5 Process Air Compressor Design Parameters

Process medium	air	
Air relative humidity	60%RH	
Air mass flow rate		
@ 66°F / 19 °C	57100 lb/h	7.2 kg/s
Air inlet pressure	14.5 psia	1.0 bara
Max. air inlet temp	77 °F	25 °C
Air temp after inter cooler	104 °F	40 °C
Air outlet pressure	43.5 psia	3.0 bara

The compressor is equipped with an air intake filter blow down valve and a silencer at discharge.

3.3.2.7 Product Gas Heating and Cooling

The description of the product gas heating and cooling system refers to the Process Flow Diagram of the Gasification Plant (Figure 3.3.2-1).

System Description

The function of the Partial Gas Combustion Nozzle is to burn product gas to increase the gas temperature high enough (2200 °F/1200 °C) for heating of gas turbine air. The high gas temperatures also cause the tar content of the gas to decompose.

The function of the gas coolers is to heat high-pressure air for expansion through the gas turbine and to cool the product gas prior to injection to the boiler.

The product gas heating and cooling system includes the following process components:

- Partial gas combustion nozzle
- Syngas cooler heat exchanger which cools the syngas by heating gas turbine air
- Emergency Spraying Nozzle

The product gas leaves the cyclone of the gasifier at 1562 °F / 850 °C temperature and 29 psia / 2.0 bara pressure. In the partial combustion nozzles swirled airflow is injected in the gas stream maintaining good mixing and partial combustion of the product gas. The temperature of gas increases to 2200 °F/1204 °C. After partial combustion the product gas flows through the gas cooler heat exchanger AH-1, which cools the syngas stream from 2200 °F/1200 °C temperature to about 670 °F / 343 °C.

The product gas is cooled in AH-1 by gas turbine air. The gas turbine air is extracted after the last stage of the compressor at 194 psia / 13.4 bara and 661 °F / 349 °C temperature and heated up to 1400°F / 760°C in AH-1.

The gas cooler system is also equipped with Emergency Spraying Nozzle before the partial combustion nozzle. In an emergency situation when the gas coolers' capacity is not enough to cool the product gas to the required temperature, water is injected into the product gas flow through the spraying nozzles to cool the gas.

The Emergency Spraying Nozzle is used to cool partly or entirely the hot product gas from 1562°F / 850°C to 650°F / 343°C in the case of malfunction of the gas cooler. The spray cooling nozzles are basically similar to steam attemperator nozzles. The preferred nozzle design is annular type, which is embedded in the refractory of the product gas duct. This arrangement causes no restrictions for the gas flow and protects the nozzles against erosion caused by the dust in the product gas.

Partial Gas Combustor

Partial Combustion Air Nozzle is used to inject hot air into the hot (1562 F/ 850°C) product gas stream so that the gas temperature is increased to 2200°F / 1204°C by partial combustion of product gas. The preferred nozzle design is an annular type nozzle, which is embedded in the refractory of the conical section of the product gas duct. This arrangement causes no restrictions for the gas flow and protects the nozzles against erosion caused by the dust in the product gas. The composition of the product gas from the gasifier is summarized in Table 3.3.2-6.

Gas Turbine Air Heater

Gas Turbine Air Heater AH-1 cools the product gas after partial combustion. The properties of product gas are shown in Table 3.3.2-7.

The gas turbine air heater design is very much vendor specific and it is not discussed here, only the process requirements are presented. The design parameters of the gas turbine air heater are summarized in Table 3.3.2-8.

Table 3.3.2-6: Product Gas Composition

Product gas at design conditions (Gasification Plant base load):		
Composition:		
Carbon monoxide (CO)	16.9 %-vol	
Carbon dioxide (CO_2)	10.8 %-vol	
Hydrogen (H_2)	14.8 %-vol	
Water vapor (H_2O)	10.3 %-vol	
Methane (CH_4)	4.1 %-vol	
Higher hydrocarbons (CxHy)	0.1 %-vol	
Nitrogen (N_2)	42.9 %-vol	
Oxygen (O_2)	0 %-vol	
Sulfuric gases (H_2S+COS)	0 ppmv	
Nitrogenous gases (NH_3+HCN)	1091 ppmv	
Hydrogen chloride (HCl)	0 ppmv	
Heavy tars (mw>200 g/mol)	<10 ppmv	
Particulates	50000 ppmv	
Heating Value:		
HHV (wet base)	149 Btu/scf	5833 kJ/m^{3}n
Molecular weight	24.42	
Gas flow rate in	63422 lb/h	7.99 kg/s
Gas inlet pressure	28.3 psia	1.95 bara
Gas inlet temperature	1562 °F	850 °C
Temp after partial combustion	2200 °F	1204 °C
Combustion air		
Supply pressure	36.3 psia	2.5 bara
Supply temperature	635 °F	335 °C
Design flow rate	14238 lb/h	1.79 kg/s

Table 3.3.2-7: Product Gas Properties After Partial Combustion

Composition:	
Carbon monoxide (CO)	14.7 %-vol
Carbon dioxide (CO_2)	9.6 %-vol
Hydrogen (H_2)	6.5 %-vol
Water vapor (H_2O)	15.5 %-vol
Methane (CH_4)	3.5 %-vol
Higher hydrocarbons (CxHy)	0.1 %-vol
Nitrogen (N_2)	50.1 %-vol
Oxygen (O_2)	0 %-vol
Sulfuric gases (H_2S+COS)	0 ppmv
Nitrogenous gases (NH_3+HCN)	940 ppmv
Hydrogen chloride (HCl)	0 ppmv
Heavy tars (mw>200 g/mol)	<10 ppmv
Particulates	50000 ppmv
Molecular Weight	25.9

Table 3.3.2-8: Gas Turbine Air Heater Heat Exchanger AH-1

Process Design Parameters		
Design load (Gasification Plant base load)		
Gas side		
Gas mass flow	77660 lb/h	9.79 kg/s
Gas inlet pressure	27.6 psia	1.90 bara
Gas inlet temperature	2200 °F	1204 °C
Gas outlet temperature	675 °F	357 °C
Air side		
Air mass flow	234389 lb/h	29.53 kg/s
Inlet pressure	194 psia	13.0 bara
Inlet temperature	661°F	349 °C
Outlet temperature (design)	1400 °F	760 °C
Mechanical Design Parameters		
Gas side		
Mechanical design pressure	29.0 psig	2.0 barg
Air side		
Mechanical design pressure	319 psig	22 barg

Emergency Water Injection

Emergency Spraying Nozzle is used to cool partly or entirely the hot product gas from 1562 °F / 850 °C to below 650 °F / 343 °C in the case of malfunction of the gas cooling. The spray cooling nozzle function is basically similar to steam attemperator nozzles. The preferred nozzle design is annular type, which is embedded in the refractory of the product gas duct. This arrangement causes no restrictions for the gas flow and protects the nozzles against erosion caused by the dust in the product gas. The internal diameter of the product gas duct is approx. 34.6 in / 880 mm. The design parameters of the spraying nozzles at base load are summarized in Table 3.3.2-9.

3.3.2.8 Flare System

The Flare System description refers to the process Flow Diagram of the Gasification Plant (Figure 3.3.2-1).

System Description

The primary function of the flare is to dispose of toxic and combustible gas components safely under relief conditions, by converting them into less objectionable products by combustion. Either elevated or ground flares can accomplish atmospheric discharge of toxic and combustible gases efficiently. An elevated flare has been selected for the project. Elevated flares are used mainly to safely dispose releases of large quantities of combustible gases.

Table 3.3.2-9: Design Parameters of Emergency Spraying Nozzle

Product gas (Gasification Plant base load)		
Total gas flow rate	77660 lb/h	9.79 kg/s
Inlet pressure	28.3 psia	1.95 bara
Inlet temperature	1562 °F	850 °C
Temperature after spray cooling	650 °F 343 °C	
Spray cooling water (preliminary data)		
Supply pressure	58 psia	4.0* bara
Supply temperature	59 °F	15 °C
Design water flow rate	17780 lb/h	2.24 kg/s

The flare system is an air-assisted flare located on the roof of the Gasification Plant building. The burner of the flare system (flare tip) is located on the top of the relief gas pipe. The flare tip is equipped with pilot burners. The pilot flame is continuously operated and ensures the safe ignition of the flare. For steady flare operation support fuel is required. The purpose of the support fuel is to maintain combustion temperature high enough to assure complete burning of all product gas compounds. Natural gas is used as support and pilot burner fuel.

Flare system will be operated at the start-up and shutdown of the gasifier and in the case of emergency shutdown or upset conditions. In those cases, the product gas is directed from the product gas line to the flare where the product gas is burnt off. The product gas is at max. 33.4 psia / 2.3 bara pressure and maximum 650 °F / 343 °C temperature when flare operation can occur. The pressure of the gas will be reduced in pressure control valves to near atmospheric level. The hot gas enters the combustor and will be ignited by the pilot flames.

During the start-up of the gasifier the product gas will be flared until the gasifier achieves adequate pressure and temperature at minimum load, which is about at the 50% capacity of the gasifier. Flaring starts when the heat-up of the gasification system starts. At the beginning only flue gas of natural gas combustion will be flared. After fuel feeding and ignition, the flue gas of biomass combustion is flared. After switching from combustion to gasification product gas of low quality will be flared until reaching the full parameter half load conditions.

In the shutdown sequence, the product gas will be flared after the gasifier capacity is reduced to minimum (50% capacity) at full pressure and temperature. Before that the product gas is still burnt in the gas burner of the boiler. The flaring starts when the gasifier is separated from the boiler. Product gas quality decreases during the shut down period. Since nitrogen and steam is used for gasifier shutdown, the flared gas is a mixture of product gas, steam and inert gas.

In all emergency cases, fuel and air feed to the gasifier will stop first, and then the flare starts operating. Syngas injection to the boiler is stopped at the same time. Similar to the normal shutdown procedure, the role of flare is to let down pressure and reduce product gas injection to the boiler in a controlled manner.

System Design

The flare system is an air-assisted flare located on the roof of the Gasification Plant building. The burner of the flare system (flare tip) is located on the top of the relief gas pipe. The flare is used to burn cooled product gas at off design conditions before it is injected in the boiler. Gasification gas properties at design conditions (Gasification Plant base load) are shown in Table 3.3.2-10. The design parameters of the flare are summarized in Table 3.3.2-11.

Table 3.3.2-10: Product Gas Properties

Composition:		
Carbon monoxide (CO) 1	4.7 %-vol	
Carbon dioxide (CO_2)	9.6 %-vol	
Hydrogen (H_2)	6.5 %-vol	
Water vapor (H_2O)	15.5 %-vol	
Methane (CH_4)	3.5 %-vol	
Higher hydrocarbons (CxHy)	0.1 %-vol	
Nitrogen (N_2)	50.1 %-vol	
Oxygen (O_2)	0 %-vol	
Sulfuric gases (H_2S+COS)	0 ppmv	
Nitrogenous gases (NH_3+HCN)	940 ppmv	
Hydrogen chloride (HCl)	0 ppmv	
Heavy tars (mw>200 g/mol)	<10 ppmv	
Particulates	50000 ppmv	
Heating Value:		
HHV (wet base)	107 Btu/scf	4193 kJ/m^3n
Molecular weight	25.9	

Table 3.3.2-11 Flare Design Parameters

Maximum gasification gas flow	77660 b/h	9.79 kg/s
Maximum gasification gas temperature	650 °F	343 °C
Maximum gasification gas pressure	33.4 psia	2.3 bara
Support and pilot burner fuel	natural gas (HHV=1000 Btu/scf)	

The Flare System shall include the following components:

- Flare tip complete with combustor, burner shell, igniters, pilot burners, etc.
- Flare stack with tip connection
- Connections to gas inlet
- Connections for drain
- Flame front generator and ignition and control panel

The ignition and control system is coupled to the Gasification Plant control system, but the activation of manual ignition sequence should be also possible from the local control panel.

3.3.2.9 Product Gas Ducting System

The arrangement of the Product Gas Ducting is shown in the Process Flow Diagram of the Gasification Plant (Figure 3.3.2-1).

System Description

The function of the product gas ducting is to lead the product gas flow to the gas injectors via the gas coolers. The product gas ducting has to stand operating pressure (system pressure) and temperature with low heat loss and pressure drop.

The product gas ducting between the gasifier cyclone and gas injector including the following:

Section A: 1st cyclone – emergency spray nozzle
Section B: emergency spray nozzle - partial gas combustion nozzle
Section C: partial gas combustion nozzle - gas cooler inlet
Section D: gas cooler exit - gas cooler inlet
Section E: gas cooler bypass duct
Section F: line from Section A to rupture disc PSE569
Section G: gas cooler exit – gas injector
Section H: line from Section G to flare

Design Basis

The product gas ducting between the gasifier cyclone and gas injector at the boiler includes the following sections:

Section A: gasifier – 1st cyclone (refractory lined)
Section B: 1st cyclone – partial gas combustion (refractory lined)
Section C: partial gas combustion – gas turbine air heater (refractory lined)
Section D: GT air heater – gasification air pre-heater (refractory lined)
Section E: air pre-heaters by-pass line (refractory lined with valve)
Section F: branching line from Section B to rupture disc (double shell/insulated)
Section G: gasification air pre-heater - boiler (external insulation)

Section H: branching line from Section G to flare (external insulation)

Section A, B, C, D and E are refractory lined ducts where the outer steel pipe holds the pressure and the inner refractory lining insulates the hot gas from the pressure vessel shell. The refractory lined product gas duct has no external insulation. The refractory lining has to protect against erosion (abrasive effect of fine dust) and corrosion (hot corrosion) and has to act as a thermal insulation as well.

The refractory lining includes two layers having different duties.

The inner layer (i.e. direct contact with the product gas) has to stand high temperatures, erosion, the effect of sintering (sinter may form on the refractory surface) and reducing atmosphere.

The outer layer of the refractory lining is between the inner layer and the pressure shell of the duct. The main task of this layer is thermal insulation to keep the temperature of the pressure shell at the designed value.

Product gas line section F (between Section B and rupture disc) has a double shell structure to ensure safe operation and relief of product gas to the rupture disc. This pipe is insulated with ceramic fiber (kaowool) between the two steel pipes.

Sections G and H are externally insulated ducts. A thin steel plate covers the insulation layer. The inner steel pipe holds the pressure and the insulation lining holds the surface temperature of the covering plate at the design value.

3.3.2.10 Nitrogen Generation System

The Nitrogen Distribution System description refers to the process Flow Diagram of the Gasification Plant (Figure 3.3.2-1).

System Description

Nitrogen is used in the Gasification Plant for inertization, pressurization and feeding of fuel and limestone, pressurization and inertization of ash discharge, as purge gas in the gasifier and flare, pressurizing the closed loop cooling water system and fluidizing gas during shutdown. Nitrogen is generated in an on-site nitrogen generation plant. Due to the small capacity of the nitrogen plant, the nitrogen generator is of membrane type.

Nitrogen plant comprises the following major equipment:

- Air compressor equipped with air intake filter and after cooler
- Air receiving tank
- Filter skid removing dust, water and oil from the air
- Air heater
- N_2 generator unit (membrane packages)
- Nitrogen receiver tank

The air compressor supplies air of 188 psig /13 barg pressure through an after cooler at 50-104 F/10-40 °C to the air receiver tank. Air is directed from the air receiver tank to the filter skid where dust, water and oil condensate is removed from the air. Following the filter unit the air is heated up by an electrical heater at least 7 °C above the dew point temperature. This ensures that no liquids are formed which can damage the membranes in the nitrogen generation system. In the membrane type nitrogen generator air is separated into product nitrogen stream and reject stream consisting of oxygen, nitrogen, argon, water vapor, carbon dioxide and other gases. The membrane packages are operated at 104-122 °F / 40-50 °C temperature, the pressure of discharge nitrogen is 130-145 psig / 9-10 bara. The purity of the product nitrogen is 98%-vol.

The generated nitrogen is stored in a nitrogen receiver tank at 87-145 psia / 6-10 bara pressure. The Nitrogen Distribution System distributes nitrogen from the nitrogen receiver tank to the consumers in the Gasification Plant.

Design Basis

The design parameters of the nitrogen generation system are shown in Table 3.3.2-12.

Table 3.3.2-12: Nitrogen Generation System Process Design Parameters

Required N_2 generation capacity	516 lb/h	0.065 kg/s
Estimated normal N_2 consumption	357 lb/h	0.045 kg/s
Peak N_2 consumption (from tank)	3800 lb/h	0.48 kg/s
N_2 purity	>98 %vol.	>98 %vol.
N_2 supply pressure, approx.	145 psia	10 bara

Available air for nitrogen generation is ambient air at 66 °F / 19 °C temperature and 70% relative humidity.

3.3.3 Boiler Modification and Internal Air Heating

Objectives

The major objective of the Boiler Modification study is to develop a design for the in-furnace high temperature/high pressure air heater AH-3 for installation in #2 Power Boiler at the DeRidder, Louisiana site. The following have also been performed:

- Determine furnace performance, steam temperature analysis, flue gas temperature analysis, boiler efficiency, fuel flow, and boiler system heat balance.
- Review grate size (capacity/turndown), over-fire air/under-grate air flow splits, combustion air temperature (under-grate air (UGA) limitations) and UGA tempering air system (if required).
- Circulation study for original 225,000 lb/hr steam flow and 250,000 Lb/hr steam flow (with syngas injection).
- Evaluate vibrations in the economizer
- Review boiler auxiliary systems (i.e., fans).
- Analyze the operational scenarios of the biomass gasifier and/or AH-3 off-line

Findings and Conclusions

Based on the above analyses, the following are findings and conclusions of the study:

- At 125% maximum continuous rating (MCR), 250,000 lb/hr steam flow, 320,000 lb/hr AH-3 air inlet flow, the AH-3 outlet temperature is 1400°F. The superheater steam outlet pressure is 850psig and the SH steam outlet temperature is 825°F. The syngas flow is 77,660 lb/hr and the dried bark (20% moisture by weight) flow is 38,250 lb/hr.

- AH-3 consists of 18 assemblies with 2 tubes per assembly for a total of 36 air flow circuits. The design pressure is 300 psig. The tube size is 2.75-inch OD and the wall thickness is 0.220 inch. The material selected is SB-407 800 HT with the exception that the final 7 tubes of each assembly will be Haynes 230 material.
- The results of the circulation study shows that no DNB is expected in the furnace waterwall with Boiler #2 having the AH-3 in the furnace and operating at 250,000 lb/hr steam flow (125% MCR) and 320,000 lb/hr HT HP AH air flow.
- The syngas injection system has 12 nozzles each at 6 in ID and located on the rear wall below the present HMZ OFA nozzles and above the furnace grate.
- There may be half standing wave frequency vibrations in the economizer due to coupling with the natural frequency of economizer duct plate.
- The UGA air temperature limitation is 480°F because there is an extensive fabric type grate seal.
- The FD fan and ID fan requirements will have to be reviewed as operation of the boiler above 225,000 lb/h may exceed the capacity of one or more of these fans.
- The system off line analysis indicates that air must be flowing to the AH-3 at all times when boiler #2 is in operation.

3.3.3.1 Description of #2 Power Boiler

The furnace water walls are 2.50 inch OD tubes, 0.188-inch tube wall thickness, 3.0 inch spacing (centerline-to-centerline), and the tube material is SA 178-A. The furnace depth is 18 ft 11 inch and the furnace width is 18 ft 11 inch. The furnace volume is 15633 ft^3 and the furnace projected water wall area including the exit plane is 3925 ft^2. The furnace is mounted over a traveling grate (forward direction from rear of furnace to front of furnace) and the effective grate area is 358 ft^2.

The furnace is fired by four pneumatic bark fuel distributors on the front wall. There are two auxiliary natural gas fired burners on the rear wall. Preheated air is supplied to the furnace via the UGA plenum and twelve (total) new HMZ over-fire air nozzles of which six are on the front wall and six are on the rear wall. The boiler has the capability for fly-ash/cinder re-injection and twelve re-injection nozzles are located on the rear wall. There is a pendant style radiant superheater (primary SH) in the upper portion of the furnace. The primary SH is composed of 18 platens at 12 inch spacing having two tubes per platen assembly for a total of 36 primary steam flow circuits. The tubes are 1.75 inch OD and the tube wall thickness is 0.149 inch. The tube material is SA 209-Tl. The radiant primary SH is supplied with saturated steam flow from the upper drum of the boiler and flows into primary steam outlet header for steam temperature control (if necessary) by spray water. The steam flows to the inlet header of the high temperature superheater (HTSH), a convection type secondary superheater and thence to the superheated steam outlet header. The HTSH tubes are 2.125 inch OD with 0.149-inch tube wall thickness and are made of SA 209-Tl material.

The boiler bank has a 60-inch ID upper drum and 42 inch ID lower drum. The front tubes of the boiler bank are 2.0 inch OD and have a tube wall thickness of 0.148 inch. The. rear tubes of the boiler bank are 2.50 inch OD and the tube wall thickness is 0.148 inch. The boiler bank tube material is SA 178-A.

The economizer is a counter flow type having 2.0-inch OD tubes and tube wall thickness of 0.180 inch. The economizer tube material is SA 210-AI.

The present air heater is a tubular type with flue gas flowing inside the tubes and air in counter cross flow on the outside of this two-pass system. The air heater tube size is 2.50 inch OD and the tube wall thickness is 10 gauge. The AH tubing material is reported to be SA 1015 (1) grade steel.

The dust collector was supplied by Gaines Equipment (Louisiana) and has 88 tubes that were changed from 9-inch diameter to 14-inch diameter.

The present forced draft fan on #2 Power Boiler is by Chicago Blower Corp. (size 6000AF, *D/1903,* 91% DWDI, N3, C/1200, W/IVC). The test block condition is 117,000 CFM at 17.2 in.w.c. outlet static pressure. For a steam flow of 225,000 lb/hr, the predicted system resistance is 14 in.w.c. at 75,000 CFM (original FD fan sheet)

The present (original) ID fan is a Buffalo Forge 2715 H14 and the test block conditions are 237128 ACFM @ 28.93 in.w.c. static suction, 445°F, and 0.0439 lb/ft^3 density.

The original boiler performance design conditions for #2 Power Boiler are 200,000 lb/hr steam flow at 100% MCR, 825°F/859 psig, 350°F feedwater temperature, 30% excess air, and 4,500 Btu/lb HHV/50% moisture by weight bark fuel. The 2001 upgrade design conditions for the boiler are 300,000 lb/hr steam flow (150% MCR), 825°F/850psig, 360°F feedwater temperature 30% excess air, and 4,526 Btu/lb HHV/52.5% moisture by weigh bark fuel. The original design conditions (100% MCR; 112.4% MCR by Combustion Engineering and upgrade design conditions by Alstom Power) are listed in Table 3.3.3-1.

The current design case of 250,000 lb/hr steam flow at 125% MCR, 825°F/850 psig 353°F feedwater temperature, 30% excess air, and 4,526 Btu/lb/52.5% moisture by weight bark fuel is also shown in Table 3.3.3-1.

3.3.3.2 Implementation of Bark Boiler #2 Modifications

The major modifications to Boiler #2 are the addition of wet bark dryer(s), bypassing of the present tubular air heater, injection of syngas for NOx reduction and combustion improvement, and the addition of a high temperature/high pressure air heater in the furnace. The purpose of the wood-waste dryer(s) is to remove moisture from 52.5% by weight wet bark. The dried bark moisture content will be 20% by weight.

The tubular air heater as presently installed, preheats the air for the under-grate air system, HMZ over-fire air system, and auxiliary natural gas burners (cooling air flow when not in use or natural gas combustion air when in use). Therefore, at present, all of the preheated combustion air flows through the tubular air heater. The preheated combustion air (after bypassing of the tubular air heater) will be supplied as exhaust air extracted from a HRSG at the same temperatures as normally supplied for UGA and OFA.

The high temperature/high pressure air heater is a parallel cross flow continuous radiant air heater that will be installed to fit above the top of the auxiliary natural gas burners and below the furnace arch tip. The air heater is composed of 18 assemblies, 2 air flow circuits per assembly for a total of 36 air flow circuits. The assemblies are on 12-inch centers, 3 % inch tube spacing, and the tube OD is 2 % inches. The air enters through a lower header at 661°F, 180 psig, and exits through an upper header at 1400°F. The design pressure is 300 psig.

Table 3.3.3-1: Boiler Performance – Original Design and 2001 Upgrade

Items	Units	Original Design # 22378		Adv. Power Sys. Step 3	2001 Upgrade. Design
Steam Flow	lbs/hr	200,000	225,000	250,000	300,000
Steam Temp/press	ºF/psig	825/850	825/850	825/850	825/850
Feedwater Temp	ºF	350	350	350	360
Blowdown	lbs/hr	0	0	2,000	6,000
Bark Fuel Flow	lbs/hr	69,310	78,090	64,421	106,980
Carbon Loss	%heat input	1	1	1	2
Excess Air T AH In	%	30	30	20	30
Air Temps					
-to Fan	ºF	80	80	80	80
-to Furnace	ºF	352	383	375	430
Air thru FD Fan	lbs/hr	293,000	330,000	392,477	441,800
Exit Gas Flow	lbs/hr	362,000	407,000	506,318	558,200
Exit Gas Temp	ºF	352	366	238	439
GI (fuel)	MMBtu/hr	311.9	351.4	436.0	484.2
GHRR	MMBtu/hr - ft2	813,000	915,000	814,500	1,262,100
Thermal Efficiency	%	70.25	69.89	78.9	66.88
Bark Fuel Analysis(wet)					
Carbon	%wt	26.15	26.15	25.65	25.65
Hydrogen	%wt	3.15	3.15	3.04	3.04
Oxygen	%wt	20.25	20.25	16.08	16.08
Nitrogen	%wt	0.05	0.05	0.10	0.10
Sulfur	%wt	0.00	0.00	0.00	0.00
Ash	%wt	0.4	0.4	2.63	2.63
H2O	%wt	50.00	50.00	52.50	52.50
Total	%wt	100.00	100.00	100.00	100.00
HHV	Btu/lb	4500	4500	4526	4526

3.3.3.3 Bark Boiler #2 Performance

The boiler performance for implementation of Boiler #2 modifications is shown in Table 3.3.3-2 Heat and Mass Balance. The three cases of implementation/modification are labeled as Step 1, Step 2, and Step 3. For Step 1, the boiler load is 125% MCR at 250,000 Lb/hr steam flow. The fuel is wet bark (52.5% moisture by weight) and syngas (670°F) and the excess air is 20%. The present tubular air heater has not been removed from #2 Boiler. AH-3 has not been installed in the furnace of #2 Boiler.

For the Step 2, the boiler load is 125% MCR at 250,000 Lb/hr steam flow, the fuel is dried bark (20% moisture by weight) and the syngas (670°F) and the excess air is 20%. The present tubular air heater is bypassed and the preheated combustion air for UGA and OFA will be extracted from HRSG-1. Air heater AH-3 has not been installed in #2 Boiler.

For the case Step 3, the boiler load is 125% MCR at 250,000 Lb/hr steam flow. The fuel is dried bark (20% moisture by weight) and syngas (670°F) and the excess air is 20%. The present tubular air heater is bypassed. Combustion air for UGA and OFA is supplied by external HRSG-1 and HRSG-2. Air heater AH-3 is installed in the furnace of #2 Boiler.

In Step 1 (no heating surface modifications in Boiler #2) bark is fed to the boiler at the as-received moisture of 52.5 wt%. Syngas from the gasifier enters the furnace at 670°F as the second fuel stream. The total fuel input to the boiler is 378.7 MMBtu/h. The boiler produces 250,000 lb/h of superheated steam at 850 psig and 825ºF. Usable heat output from the boiler is 281.1 MMBtu/h, for a boiler efficiency of 74.2%

In Step 2 the tubular air heater is bypassed, all bark fuel to the gasifier and boiler is dried using waste heat in the boiler flue gas, and hot air is supplied heated air to the furnace (OFA and UGA from a portion of the hot vitiated air from the HRSG-1 exhaust. Bark is fed to the boiler from the dryers at a moisture of 20 wt%. Syngas from the gasifier enters the furnace at 670°F as the second fuel stream. The total fuel input to the boiler is 354.0 MMBtu/h. The boiler produces 250,000 lb/h of superheated steam at 850 psig and 825ºF. Usable heat output from the boiler is 281.1 MMBtu/h and boiler efficiency is 79.4%. Low-level waste heat from the boiler flue gas is used to perform the majority of the fuel drying rather than high-level heat in the furnace. This, plus the recovery of waste heat from HRSG-1 exhaust improves the overall boiler efficiency.

In Step 3 the tubular air heater remains bypassed, combustion air is supplied from the combined exhaust of HRSG-1 and HRSG –2 , and all bark is dried to 20-wt% with boiler flue gas. Syngas from the gasifier enters the furnace at 670°F as the second fuel stream. The internal air heater AH-3 is installed in the furnace, heating high-pressure air from the compressor of the second externally recuperated gas turbine, GT-2. The total fuel input to the boiler is now 436.0 MMBtu/h. The boiler again produces 250,000 lb/h of superheated steam at 850 psig and 825ºF. The boiler now also produces 320,000 lb/h of high-pressure air at 1400ºF and 178 psig for expansion through GT-2. Usable heat output (steam plus air) from the boiler is 343.9 MMBtu/h, for an effective boiler efficiency of 78.9%. Low-level waste heat from the boiler flue gas is again used to perform the majority of the fuel drying, rather than high-level heat in the furnace. The high-level furnace heat is therefore available for heating air to the turbine, which produces 17 MW of self-generated electricity. The overall system, including the external air heater AH-1 and its associated externally recuperated gas turbine provides 34 MW of new self-generated electricity to the mill, plus 250,000 lb/h of 850 psig superheated steam and about 129,000 lb/h of process steam including 100,000 lb/h of 250 psig steam from the HRSGs.

Table 3.3.3-2 Boiler #2 Heat and Mass Balance
(Sheet 1 of 3)

	Units	Step 1	Step 2	Step 3
Load	%	125	125	125
Fuel Heat Input				
Bark Fuel	10^6 Btu/hr	234.3	209.6	291.6
Syn Gas	10^6 Btu/hr	144.4	144.4	144.4
Total	10^6 Btu/hr	378.7	354.0	436.0
Heat Output				
SH Steam	10^6 Btu/hr	281.1	281.1	281.1
HPHT Air Heater	10^6 Btu/hr	NA	NA	62.8
Total	10^6 Btu/hr	281.1	281.1	343.9
Efficiency (Steam + Air)	%	74.2	79.4	78.9
Water/Steam Flow				
Outlet SH Steam	lb/hr	250000	250000	250000
Spray	lb/hr	0	0	0
Blowdown	lb/hr	2000	2000	2000
Feedwater	lb/hr	252000	252000	252000
Water/Steam Pressure				
Feedwater	psia	1012	1012	1012
Drum	psia	962	962	962
SH Outlet	psia	865	865	865
Fuel/Air/Flue Gas Flow				
Excess Air	%	20	20	20
Bark Fuel	lb/hr	51750	27500	38250
Syngas	lb/hr	77660	77660	77660
Total Combustion Air	lb/hr	293957	315253	392477
Flue Gas	lb/hr	421705	418926	506318

Table 3.3.3-2 Boiler #2 Heat and Mass Balance
(Sheet 2 of 3)

	Units	Step 1	Step 2	Step 3
Load	%	125	125	125
Fuel/Air/Flue Gas Temp				
Bark Fuel	°F	80	80	80
Syn Gas	°F	670	670	670
Ambient Air	°F	80	80	80
Backpass Tubular AH Outlet	°F	375	NA	NA
External Air to Furnace	°F	NA	375	375
Bark Fuel (as fed)				
Water	wt. Dec.	0.5250	0.2000	0.2000
Nitrogen	wt. Dec.	0.0010	0.0017	0.0017
Hydrogen	wt. Dec.	0.0304	0.0512	0.0512
Oxygen	wt. Dec.	0.1608	0.2708	0.2708
Carbon	wt. Dec.	0.2565	0.4320	0.4320
Ash	wt. Dec.	0.0263	0.0443	0.0443
Total	wt. Dec.	1.0000	1.0000	1.0000
HHV	Btu/lb	4526	7622	7622
Syn Gas Fuel				
CO	Vol %	14.626	14.626	14.626
CO 2	Vol %	9.542	9.542	9.542
H2	Vol %	6.410	6.410	6.410
H2O	Vol %	15.545	15.545	15.545
CH4	Vol %	3.505	3.505	3.505
N2	Vol %	50.204	50.204	50.204
C2H4	Vol %	0.021	0.021	0.021
C6H6	Vol %	0.053	0.053	0.053
H2S+COS	Vol %	0	0	0
NH3+HCN	Vol %	0.094	0.094	0.094
O2	Vol %	0	0	0
Total	Vol %	100.00	100.00	100.00
HHV	Btu/SCF	105.608	105.608	105.608

Table 3.3.3-2 Boiler #2 Heat and Mass Balance
(Sheet 3 of 3)

	Units	Step 1	Step 2	Step 3
Load	%	125	125	125
HT HP Air Heater				
Air Flow	lb/hr	NA	NA	320000
Air Inlet Pressure	psia	NA	NA	208
Air Outlet Pressure	psia	NA	NA	178
AirΔP	psi	NA	NA	30
Air Inlet Temperature	°F	NA	NA	661
Air Outlet Temperature	°F	NA	NA	1400
Air ΔT	°F	NA	NA	739
Number of Circuits	--	NA	NA	36
Tube OD	inch	NA	NA	2.75
Tube ID	inch	NA	NA	2.31

3.3.3.4 Internal High Temperature High Pressure Air Heater AH-3

This section discusses the preliminary design developed for the internal air heater AH-3. The external air heater is discussed in Section 3.3.4. The design of the air heaters is determined in large part by the gas turbine to which they are coupled in the externally recuperated gas turbine (ERGT) cycle. Two gas turbines were considered during the evaluation. The first was a Titan 130 manufactured by Solar Turbines Inc. This turbine was used as the baseline for the study, and all heat exchanger designs, both internal and external, were based on inlet air conditions of 810ºF and 250 psig. Later in the study, the Alstom GT 35 engine was selected as it is more readily adapted to the ERGT cycle and produces 20% more power (17 MWe) than the Titan 130. The GT 35 is a lower pressure machine with air leaving the turbine's compressor at 179 psig and 661°F. The air heater designs based on the Titan 130 can be considered to be conservative from a mechanical design and cost estimating standpoint for the purposes of the current study. Designs specific to the GT 35 engine will be developed in the detailed design phase.

AH-3 is designed as a parallel cross-flow tube bundle to be located in the furnace above the top of the natural gas auxiliary burners are below the furnace arch tip. There are eighteen (18) assemblies at 12-inch spacing and two air flow circuits per assembly for a total of 36 air flow circuits. The tube OD is 2.75 inch and the tube thickness is 0.220 inch. The air heater as located in the furnace is shown in Figure 3.3.3-1. Selected elevation views of the conceptual air heater are shown in Figures 3.3.3-2 and 3.3.3-3.

The air enters at 250 psig and 810°F from the lower air header and leaves the tubes at 215 psig and 1400°F to enter the upper air header. The design pressure was selected to be 300 psig for the study.

A metal study was conducted for the air heater tube bundle based on the air conditions discussed above. The metal study was completed for a case of 125% MCR, 250,000 Lb/hr steam flow, 77,660 lb/hr syngas flow to the furnace, 35753 lb/hr dried bark flow at 20% moisture by weight, and 320,000 lb/hr HT HP air heater flow at 810^{o}F, 250 psig inlet conditions.

The first part of the metal study was done for the case of no air flow imbalance from tube circuit to tube circuit. The results of this part of the metal study are shown in Figure 3.3.3-4. The bulk air temperature and average tube metal temperature versus tube number is shown as a graph for individual continuous tube circuits. A sketch in the upper left portion of Figure 3.3.3-4 shows the continuity of each tube circuit. That is, tube circuit 1 is a continuum of tubes 1, 4, 5, 8, 9, 12, 13, 16, 17, 20, 21, and 24 and tube circuit 2 is a continuum of tube 2, 3, 6, 7, 10, 11, 14, 15, 18, 19, 22, and 23. Maintaining a constant tube wall thickness along the tube circuit length will provide for minimum pressure drop. The average tube metal temperature at the outlet of tube 24 in circuit 1 is 1404^{o}F and the bulk average air temperature with no air flow imbalance is 1378°F. The ASME minimum tube wall thickness for SB-407 800 HT material at 300 psig/1404°F average metal temperature is 0.1318 inch (Le. 0.132 inch). The selected tube wall thickness of 0.220-inch is greater than 0.132-inch and is acceptable. The average tube metal temperature at the outlet of tube 23 in circuit 2 is 1376°F and the bulk average air temperature with no air flow imbalance is 1352°F. The ASME minimum tube wall thickness for SB-407 800 HT material at 300 psig/1376°F is 0.1173-inch (i.e. 0.118 inch). The selected tube wall thickness of 0.220-inch is greater than 0.118-inch and is acceptable.

Each of the thirty two (32) tube circuits will never have perfectly equal air flow. In practice, it is customary to allow for tube-to-tube flow imbalance at the design stage. The second part of the metal study was done for the case of air flow imbalance from tube circuit to tube circuit. The results of this part of the metal study are shown in Figure 3.3.3-5. The bulk air temperature, bulk air temperature plus air temperature imbalance, and average tube metal temperature versus tube number is shown as a graph for individual continuous tube circuits. A sketch in the upper left portion of Figure 3.3.3-5 shows the continuity of each tube circuits.

The average tube metal temperature at the outlet of tube 24 in circuit 1 is 1554°F and the bulk air flow temperature with imbalance is 1528°F. The limiting metal temperature for SB-407 800 HT material is 1500°F for this condition, the minimum ASME wall thickness is 0.1893 inch (0.189 inch) for SB-407 800 HT material at 300 psig and 1500°F temperature limit. In this case, a new developmental material Haynes 230 (ASME CC2063) is selected to perform at 300 psig/1554°F and the minimum tube wall thickness for Haynes 230 material is 0.1229-inch (0. 130-inch). Therefore, tube wall thickness of 0.220 inch is greater than 0.130 inch and is acceptable. Tubes 20, 21,and 24 would be made of Haynes 230 material. The average tube metal temperature at the outlet of tube 23 in circuit 2 is 1500°F. The limiting metal temperature for SB-407 800 HT material is 1500°F. Again, for this condition, the minimum ASME wall thickness is 0.1893-inch (0. 189-inch) for SB-407 800 HT material at 300 psig and 1 500°F temperature limit. In this case again, the new developmental material Haynes 230 (ASME CC 2063) is selected to perform at 300 psig/1523°F and the minimum tube wall thickness for Haynes 230 material is 0.1088-inch (0.109-inch). Therefore, a tube wall thickness of 0.220-inch is greater than 0.109-inch and is acceptable. Tube 22 and 23 would be made of Haynes 230 material.

3.3.3.5 Circulation Study

A circulation study was done to estimate the circulation ratio for the case of 250,000 lb/hr steam flow, 30% excess air, dried bark fuel and syngas fuel, no tubular air heater, and air heater AH-3 in the furnace (Case A). Also, a circulation study was done to estimate the circulation ratio for the case of 225,000 lb/hr steam flow, 30% excess air, wet bark fuel, and tubular air heater as presently installed in the boiler (Case B).

The circulation ratio is defined as the ratio of the total emulsion flow to the total steam generated.

For case A, the circulation ratio is 8.4 and the peak waterwall heat flux is 51,000 Btu/hr-ft^2 to the furnace waterwall. The average water wall tube metal temperature is 600°F and the calculated ASME minimum tube wall thickness is 0.1053 inch, which is less than the tube wall thickness of 0.188-inch.

For case B, the circulation ratio is 8.19 and the peak waterwall heat flux is 58,000 Btu/hr ft^2 to the furnace waterwall. The average waterwall tube metal temperature is 627°F and the calculated ASME minimum tube wall thickness is 0.1071-inch, which is less than the tube wall thickness of 0.188-inch.

For case A, the inside tube wall temperature is 584.3°F at peak heat flux and the temperature difference $\Delta Tx = Ti - Tsat$ is 46.8^{o}F. The nucleate boiling regime is individual bubble regime.

For case B, the inside tube wall temperature is 609.3°F and the temperature difference $\Delta Tx = Ti - Tsat$ is 69.4^{o}F. The nucleate boiling regime is individual bubble regime.

In general, the departure from nucleate boiling heat transfer regime, DNB, occurs at ΔTx somewhat greater than 100^{o}F.

The excess temperature difference ΔTx is defined as the excess temperature above the boiling point. The maximum heat flux at the excess temperature ΔTx is of the order of approximately 500,000 Btu/hr ft^2 and at this point DNB will begin as ΔTx increases. The maximum peak heat flux for case A and case B will not cause DNB.

The results are shown in Table 3.3.3-3.

3.3.3.6 Furnace Syngas Injection

The furnace cross-section dimensions are 18 ft 11-inch width and 18 ft 11-inch depth. The stoker floor is at elevation (relative to sea level) 227 ft 0-inch and the top of the stoker grate is approximately 231 ft 0-inch. The centerline elevation of four (4) pneumatic bark distributors are located on the front wall of the furnace and the bark is injected in a trajectory along the depth of the furnace toward the rear wall.

Twelve HMZ over-fire air nozzles are located at elevation 245 ft 3-inch on the rear wall. The centerline of the two auxiliary natural gas burners is at elevation 250 ft 9-inch and the natural gas burners are located on the rear wall. The location of the syngas injection ports will be at an approximate elevation of 235 ft on the furnace rear wall using existing deactivated OFA nozzles and through new nozzles in the front wall of the boiler at an approximate elevation of 233 ft. The front nozzles will be located below the front header and no pressure parts will be modified.

The nozzles will be sized to insure adequate velocity for uniform mixing in the furnace. A backup reburning fuel system using natural gas mixed with a small portion (10-15%) of recirculated flue gas will be available to replace syngas in the event that the gasifier is off line.

3.3.3.7 Vibrations in the Economizer

The economizer tube bundle is located in the boiler back pass duct that is 17 ft 10 1/2 inch overall height, 18 ft 10 1/4-inch wide and 8 ft 0-inch depth. The ductwork plate is 3/16-inch carbon steel. The economizer is a counterflow type having 24 circuits of 2-inch OD tube. External plate stiffeners are arranged in a manner such that the basic plate sizes resulting are 2 ft x 8 ft, 2 ft x 1 ft 8 l/8-inch, and 2 ft x 7 ft 9-inch and the natural frequencies of these plates are 20.5 HZ, 39 HZ, and 1.4 HZ (resp.) as considered to be rigidly supported by the stiffener arrangement.

For the case of 125% MCR, steam flow 250,000 lb/hr, dried bark (20% moisture by weight) and syngas fuel input, flue gas flow of 525,709 lb/hr at 30% excess air, HT HP AH installed in furnace, 605°F economizer flue gas exit temperature, and no external original tubular air heater, the resulting Strouhal vortex shedding frequencies are 79 HZ at the flue gas inlet to the economizer, 72 HZ in the mid section of the economizer, and 65 HZ at the flue gas exit of the economizer.

The flue gas standing half wave frequencies at the gas inlet to the economizer are 46 HZ along the width and 108 HZ along the depth. The flue gas standing half wave frequencies at the midsection of the economizer are 44 HZ along the width and 103 HZ along the depth. The flue gas standing half wave frequencies at the economizer gas exit are 42 HZ along the width and 98 HZ along the depth. Standing half waves are opposite in phase of flue gas pressure pulsations.

The half standing wave frequency of 42 HZ along the depth at the economizer exit is nearly synchronous with the natural frequency of the 2 ft x 1 ft 8 1/2 inch duct plate of 39 HZ. Therefore, a source frequency is nearly coupled with a natural frequency and vibration may occur.

3.3.3.8 Combustion System

The purpose of this section is to review grate capacity and turndown combustion air flow apportionment between over-fire air and under-grate air, combustion air temperatures and air tempering systems (if required).

The No. 2 boiler original performance design grate heat release rate for 200,000 lb/hr steam flow and 225,000 lb/hr steam flow and 30% excess air is 813,000 Btu/hr-ft^2 and 915,000 But/hr-ft^2, respectively based on grate bark fuel heat input. The No. 2 boiler upgrade performance design grate heat release for 300,000 lb/hr steam flow and 30% excess air is 1,262,100 Btu/hr-ft^2 based on grate bark fuel heat input, which is higher than generally recommended.

For Step 3 of the current plant modifications, 250,000 Lb/hr steam flow, 20% excess air the grate heat release rate is 762,000 Btu/hr-ft^2 based on grate bark fuel heat input. The GHRR for Step 3 is equal to the GHRR of Bark boiler #2 as originally designed and is far lower than the upgrade design heat release rate. An acceptable grate turndown is a four-to-one ratio and the purpose of limiting the turndown is to insure a bed of ash and fuel are on the grate at all times during operation.

Table 3.3.3-3: Circulation Study Results

Case		A	B
	Units		
Load	%	112.5	125
Excess Air	%	30	30
Fuel	Type	Wet bark @ 50% moisture by weight	Dried bark @ 20 % moisture by weight
Fuel	Type	NA	Syngas .
Tubular Air heater	Y/N	Yes	No
HT HP Air heater AH-3	Y/N	No	Yes
Steam flow	Lb/hr	*225,000*	*250,000*
Blowdown	Lb/hr	0	2000
FW Flow	Lb/hr	*225,000*	*252,000*
FW Pressure'	Psia	993	1012
Drum Pressure	Psia	943	962
SH Outlet Pressure	Psia	865	865
FW Temp	ºF	350	353
Econ Outlet FW Temp	ºF	477	479
Drum Sat. Temp	ºF	537.5	539.9
Peak Waterwall Heat Flux	Btu/hr-ft^2	51000	*58000*
Waterwall A vg Tube Temp	ºF	600	627
Inside Tube Wall Temp, Tt	ºF	584.3	609.3
$L\backslash x = Tit - T_{SAT}$	ºF	46.8	69.4
Waterwall Tube OD	Inch	2.5	2.5
Design Pressure	Psig	1025	1025
Waterwall Tube Material	Type	SA-178A	SA-178A
Waterwall Tube Thickness	Inch	0.188	0.188
ASME Tube Minimum	Inch	0.1053	0.1071
Circulation Ratio	Lbm/Lbm	8.40	8.19

The path of combustion air to Bark boiler #2 as originally designed and with the upgrade HMZ over-fire air system is as follows:

- The forced draft fan supplies air to the tubular air heat for combustion air heating.
- The heated air exits the tubular air heater in two paths: one path to the left side (west) of Bark boiler #2 and one path to the right side (east) of Bark boiler #2.
- The heated air takeoff for air supplied to the auxiliary natural gas burners is the juncture (both air paths).
- The air paths continue and have two more junctures apiece whereby each side path supplies over-fire air for 1/2 of the furnace wall and 1/2 of the front furnace wall OFA requirements.
- The air paths continue and deliver the under-grate air to the left side (west) and right side (east) of the under-grate air plenum of the stoker.

The grate seal (fabric belt type) has a temperature limitation of 480°F (maximum) and is designed for ±25 in. w.c.

With the introduction of syngas above the grate as approximately 1/3 of the boiler fuel input, the OFA/UGA air flow split will be changed to accommodate an increase in OFA and decrease in UGA.

The combustion air temperature to the furnace (OFA/UGA) for the proposed modifications are in the range of 375ºF, well below the maximum of 480°F for the UGA limitation.

3.3.3.9 Boiler Auxiliary Systems

The purpose of this section is to evaluate the forced draft fan, induced draft fan, and overfire air fan.

Forced Draft Fan

The installed forced draft fan on Bark boiler #2 is a Chicago Blower Corp. (size 6000 AF, D/1903, 91% DWPI, Al3, C/1200, VIVC) fan rated at 17.2 in.w.c. outlet static pressure, air flow of 117,000 CFM/110^{o}F, 0.0707 lb/ft^2 density. The air mass flow rate is 496,314 lb/hr.

Induced Draft Fan

The installed (original) ID fan on Bark boiler #2 is a Buffalo Forge 2715 H1U with a test block rating of 237128 ACFM@ 28.93 in.w.c. static suction at 445°F and 0.0439 Lb/fl^2 density. The corresponding flue gas mass flow rate is 624,595 lb/hr.

Over-fire Air Fan

There are no overfire air fans on Bark boiler #2. The overfire air to the upgraded HMZ OFA system is supplied through the ductwork from the outlet(s) path(s) of the installed tubular air heater.

Summary of FD Fan and ID Fan Performance

A summary of FD fan and ID fan performance is listed in Table 3.3.3-4. The cases selected for comparison are the original design load of 200,000 lb/hr steam flow and 225,000 lb/hr steam flow, the Entec Study[2] load of 250,000 lb/hr steam flow, and the Step 1, 2, and 3 loads of 250,000 lb/hr steam flow.

In Implementation Step 1 woodwaste for the gasifier only is dried from 52.5% moisture by weight to 20% moisture by weight. As-received bark at 52.5% moisture is fed to the boiler. In this Step the dryer is fired with natural gas, the combustion products of which are mixed with a small amount of recirculated flue gas from the exit of the boiler ID fan. The dryer output is the dried woodwaste for the gasifier and the moist dryer exhaust gas. The exhaust gas is added to the boiler tubular air heater exit gas flow and the combined flow enters the dust collector.

In Step 2 and 3, the tubular air heater is bypassed. Flue gas from the economizer exit flows directly to the wood waste dryer to dry the moist bark fuel. A portion of the flue gas bypasses the dryer and recombines with the dryer exhaust at the dust collector inlet. This increases the flue gas temperature at the dust collector above the 175ºF exhaust temperature of the dryer.

The primary benefit of the bark dryer(s) is that low-level waste heat from the boiler flue gas is used to dry the bark rather than high-level heat in the gasifier and stoker boiler. As a result, more high-level heat is available to drive the gasification reactions in the gasifier and to produce superheated steam and air in the boiler. An additional benefit is that the portion of bark moisture removed in the dryer bypasses the boilers upper furnace and goes directly to the dust collector. No. 2 boiler is already known to be operating at high velocity and pressure drop in the upper furnace. Bypassing a portion of the bark moisture directly to the dust collector significantly reduces the mass flow of water vapor in the upper furnace. Comparing the Entec case to the Step 3 case (both producing 250,000 lb/h of steam) in Table 3.3.3-4 shows that even though bark feed and feed moisture are higher in Step 3, water vapor from feed moisture in the upper furnace is significantly lower (45,000 lb/h for the Entec case vs. only 12,374 lb/h for the Step 3 case.

Another advantage of the advanced power system is that a portion of the air required for combustion of the bark fuel is supplied by the gasification air compressor, which supplies air for partial combustion of a portion of the total system bark requirement in the gasifier. This air no longer needs to be supplied by the FD fan, extending its capacity to higher loads. Comparing the original boiler design cases producing 200,000 and 225,000 lb/h of steam to the Step 1 and 2 cases producing 250,000 lb/h shows that the additional steam production is accomplished with the same (Step 1) or lower (Step 2) FD air flow. For the 250,000 lb/h cases, the Step 3 boiler FD air flow is essentially the same as the Entec case despite producing a about 5 MW of electricity from heated high pressure air in addition to the 250,000 lb/h of superheated steam.

Comparison of the total flue gas flow for the 250,000 lb/h cases shows that all of the advanced power system cases result in less or equal flue gas to the ID fan (including all bark moisture) than the Entec case. Again, Step 3 produces 5 MW of electricity in addition to the steam with essentially the same flue gas flow to the ID fan.

3.3.3.10 Systems Off-Line

The purpose of this section is to analyze the operating methods and contingencies with either the Biomass Gasifier (Syngas) and/or the high temperature/high pressure air heater out of service.

Biomass Gasifier Off-Line

When the gasifier is off-line, a mixture of natural gas and recirculated boiler flue gas will be routed to the syngas reburn injection nozzles to maintain NOx reduction and combustion improvement through reburning and to keep the injection nozzle from overheating. GT-1 can be operated in a non-recuperated manner with natural gas until the gasifier is brought back online and gas can be reduced as air is again preheated with hot syngas in AH-1.

Table 3.3.3-4: FD Fan and ID Fan Performance

. Item	Units	Original PB Sheet	Original PB Sheet	Entec Study	Step 1	Step 2	Step 3.
Load	%	100	112.5	125	125	125	125
Steam Flow	lb/hr	200,000	225,000	250,000	250,000	250,000	250,000
Gasifier Bark Feed, as fed	lb/h, (% moisture)	0	0	0	23,621 (20)	23,621 (20)	23,621 (20)
Stoker Bark Feed, as fed	lb/h, (% moisture)	69,310 (50)	78,090 (50)	90,000 (50)	51,750 (52.5)	27,500 (20)	38,250 (20)
Total System Bark Feed, as fed	lb/h, (% moisture)	69,310 (50)	78,090 (50)	90,000 (50)	75,371 (42.3)	51,121 (20)	61,871 (20)
Total System Bark Feed, as rec'd	lb/h, (% moisture)	69,310 (50)	78,090 (50)	90,000 (50)	91,532 (52.5)	86,098 (52.5)	104,203 (52.5)
Fuel Moisture through upper furnace	lb/h	34,655	39,045	45,000	31,893	10,224	12,374
FD Fan							
Air Flow	lb/hr	293,000	330,000	316,724	293,957	315,253	392,477
Temperature	ºF	80	80	94	80	80	80
ID Fan							
Flue Gas Flow	lb/hr	362,000	407,000	579,057	485,759	481,321	576,069
Temperature	ºF	352	366	367	336	210	238

In Steps 2 and 3, the dryer can continue to dry bark for the boiler. In Step 1 the dryer would be idled or shut down until the gasifier was restarted.

Externally Recuperated Gas Turbine GT-1 or GT-2 Off-Line

The gas turbines are required to provide air to the external (AH-1) and internal (AH-3) air heaters. Air flow must be maintained to AH-1 at all times when the gasifier is running and to AH-3 whenever the No.2 Boiler is running. However, each of the turbines compresses enough air to supply both air heaters. Therefore, if one turbine is down, half of the operating turbines air can be routed through the off-line turbine's air heater and back to the operating turbines combustor, maintaining full power output from the remaining turbine while reducing its natural gas demand by about half.

External Air Heater /Syngas Cooler AH-1 Off-Line

If AH-1 is off-line, the gasifier must be shut down to avoid overheating AH-1 and the syngas injection nozzles on No. 2 boiler.

Internal Air Heater AH-3 Off-Line

If AH-3 is off-line, No. 2 Boiler cannot be operated unless another means is available to cool the air heater tubes.

No. 2 Boiler Off-Line (and Gasifier Off-Line)

As currently designed, the gasifier cannot be operated when No. 2 Boiler is offline because the boiler is the only user for the syngas. However, both gas turbines can be operated on natural gas without external recuperation from the air heaters, with all turbine exhaust air routed through their respective HRSGs for 250-psig-steam production.

3.3.4 External High Temperature High Pressure Air Heater AH-1

This section discusses the preliminary design developed for the external air heater AH-1. The internal air heater design is discussed in Section 3.3.3.4.

Compressed combustion air to the externally recuperated gas turbine GT-1 combustor is pre-heated in an external heat exchanger using hot syngas from the gasifier. Three conceptual design approaches were developed for the external air heater including two with air on the tube side and one with syngas on the tube side (firetube design). All designs were based on the air conditions for the Titan 130 engine, with inlet air at 250 psig and 810ºF. In all cases syngas flow is 77,660 lb/hr and 26-psia inlet pressure. Design 3 is used as the basis for the study cost estimate for AH-1.

Design 1

In this design the external heat exchanger, AH-1, is designed to have high-pressure air flowing inside the 2.50-inch OD and 0.22 in. thick tubes. syngas at 77,660 lb/hr and 26-psia-inlet pressure flows across the outside of the tubes in a counterflow direction and is cooled from 2,200°F inlet temperature to 865°F outlet temperature. Air at 810°F inlet temperature and 265-psia inlet pressure flows inside the tubes. Air flow is 235,000 lb/hr with an outlet temperature of 1400ºF. Total outside surface area of the tubes is estimated to be 7,480 ft^2. The heat exchanger consists of 36 tube assemblies each with 2 tubes per assembly for a total of 72 circuits. The tube spacing is ST=3.50 in. and 56=3.75 in. The casing inside dimensions are 5 ft. 3 1/2 in. deep, 10

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ft. 7 in. wide, and 19 ft. 8 1/4 in. high. A conceptual design of the heat exchanger is shown in Figure 3.3.4-1.

Design 2

In this design AH-1 will have hot syngas flowing inside the tubes and air flowing outside the tubes in a pure counterflow arrangement. The syngas conditions are the same as in Design 1. The tube OD is 2.50 inches arranged in a rectangular array of 8 tubes x 9 tubes = 72 tubes (circuits) on 3-inch centers. The shell side dimensions will be 24 ½-inch width x 27 ½-inch height and the exposed tubing length is to be 100 ft having a total outside tube heat transfer area in excess of 4700 ft^2.

Design 3

In this design AH-1 is designed as a shell and tube heat exchanger to have high pressure air flowing inside the 2.50-inch OD and 0.22 in. thick tubes. Syngas at 77,660 lb/hr and 26-psia inlet pressure flows across the outside of the tubes in a counterflow direction and is cooled from 2,200°F inlet temperature to 865°F outlet temperature. Air at 810°F inlet temperature and 250-psig inlet pressure flows inside the tubes. Air flow is 235,000 lb/hr with an outlet temperature of 1400°F. The total outside surface area of the tubes is estimated to be 8,000 ft^2. The heat exchanger consists of 171 U-shaped tubes each 65.1 ft. long. The inside diameter of the heat exchanger is 9.6 ft and the overall length is 39.1 ft. Tube material is assumed to be high chrome nickel alloy HK-40.

3.3.5 Power Generation

3.3.5.1 Power Island Design Evaluation

Power generation for the plant will be accomplished by employing an externally recuperated gas turbine. The combustion air for the ERGT will be preheated by utilizing the excess sensible heat from the woodwaste gasification and stoker combustion processes. The following tasks were completed to define the most favorable power generation system configuration and system development path:

- Propose and analyze different candidate cycles for an ERGT
- Assess the viability of modifying existing Solar gas turbine designs to the ERGT cycle.
- Develop system installation, interconnection and commissioning requirements based on a standard Solar Titan 130 package to define layout and utilities requirements for the DeRidder site
- Assess the viability of gas turbines from other manufacturers for modification to the ERGT cycle
- Assess the market potential for an ERGT in the forest products industry from a turbine manufacturer's perspective

Six different ERGT cycles were considered for an ERGT system comprised of two Titan 130 sized turbines. Of these six options two were selected as the most suitable based on the practical operational constraints. These two options were thermodynamically analyzed over a range of operating parameters. Potential operating cost savings and mechanical design feasibility were also evaluated.

Three different ERGT development strategies were considered in defining the potential system configurations:

- Development for a near-term application and requiring only moderate modification,
- A long-term engine development program requiring a significant developmental effort and considerable modification to the present engine designs.
- Modifications and developmental efforts required specifically for Solar engines to operate in the ERGT configuration with current operating constraints (temperatures and material considerations).

Information was provided by Solar on the layout, installation and commissioning of their standard Titan 130 gas turbine package. These data included the arrangement and layout drawings of a standard T130 package and information regarding its various pumps, compression systems and cooling systems. This information was used to help assess the feasibility and economics of modifying the existing Powerhouse at the DeRidder paper mill to accommodate the gasification-based power generation system.

Finally Solar assessed the market potential for an ERGT from a gas turbine manufacturer's perspective.

The following sections summarize the results of the power island design evaluation. The complete results are presented in Solar's final report in Appendix B.

Proposed Cycle Concepts

The initial study was based on a power generation turbine system that provides 27 MW of electrical energy output. Configurations based on two Titan 130 size gas turbines were therefore evaluated. Six different ERGT cycles for two Titan 130-sized turbines were considered. Two of the cycles included two independent turbines, each with its own air heater and gas combustor to boost the air temperature to the required turbine inlet temperature. In one of these cycles the air heater and combustor were in series (Option 1) and the other in parallel (Option 2). In the remaining cycles two turbines were combined with a single heat exchanger and various single or dual combustor arrangements. Cycle diagrams are presented in Appendix B.

Of these six cycle options the first two, with independent turbine-air heater/combustor combinations, were selected as the most suitable based on the practical operational and cost constraints. These two cycle options were thermodynamically analyzed over a range of operating parameters, including turbine rotor inlet temperature (TRIT), pressure drop across and exit temperature of the air heater, combustor and turbine cooling, and humidification of the air at the turbine compressor inlet. Power generation and cost performance were compared to a standard Titan 130 operating on natural gas at the turbine's baseline ISO conditions, which are detailed in Appendix B.

It was found that cycles based on Option 2 provide a slightly greater power output as compared to Option 1. However, their savings in fuel cost is small when compared to a standard gas turbine and much smaller than that of a cycle based on Option 1. Since the first demonstration of this technology will most likely be with air heater exhaust temperatures in the range of 1400ºF to 1500ºF, Option 2 does not provide enough fuel cost savings to warrant development. Therefore, Option 1 was selected as the preferred cycle and the rest of the analysis that was conducted is based on this cycle option.

Detailed parametric, thermodynamic analyses of Option 1 were then conducted for two TRITs to determine the effect of HP/HT air heater pressure drop and exit temperature on the net turbine power output, natural gas consumption, turbine exhaust temperature and fuel cost. In these simulations, compressed air was used as the cooling medium for the turbine. The results of these analyses were again compared with those of the baseline engine.

The TRITs chosen for comparison were 2100ºF and 1900ºF. The TRIT of 2100ºF is currently state of the art for industrial turbines. The lower TRIT was adopted based on the assumptions that mechanical design of some of the ERGT components (such as the scroll) might force the system to be designed for a lower TRIT and that operation at lower TIRT could be cost effective when natural gas costs rise.

The results indicate that increasing the heat exchanger exit temperature and lowering its pressure drop, while increasing the TRIT, will maximize the ERGT savings and make the cycle more attractive. Note that unless the TRIT is increased beyond the current standard engine TRIT level of 2100ºF, there will always be a small penalty on the net turbine KW by using an ERGT. However, the ERGT will provide higher exhaust heat relative to the standard gas turbine when both are compared at the same TRIT.

Additional thermodynamic analysis of Option 1 was conducted to evaluate the effect of using steam as the cooling medium for the turbine. These simulations were performed for a TRIT of 2100ºF. The variables involved in this analysis were the HP/HT air heater pressure drop and exit temperature, while the parameters used for the comparison were the net turbine power output, natural gas consumption, turbine exhaust temperature, cost of fuel per KW-hr and the percentage of savings in fuel cost.

This analysis assumes that no steam enters into the turbine flow stream. However, it must be noted that with respect to the forest product industry application the analysis is not evaluating a combined cycle and that steam is assumed to be available in abundance. Therefore some leakage of steam can be permitted in the actual design. Thus the design constraints on the turbine cooling circuit can be made less restrictive and will hopefully have a small impact on the engine cost.

Due to the pressure drop across the HP/HT heat exchanger, an ERGT with an air-cooled turbine will always have a net electrical output lower than a standard gas turbine. However an ERGT with a steam-cooled turbine can provide an electrical output greater than that of a standard gas turbine for an added pressure drop of up to 20 psi. The results further show that an ERGT can produce an increase in electrical output of about 8% by switching the cooling medium of the turbine from compressed air to steam (comparisons being made for the same pressure drop across the HP/HT heat exchanger).

Along with a higher electrical output, the ERGT with steam-cooled turbine also provides a higher exhaust temperature that aids in co-generation. Further, the ERGT with a steam cooled turbine generates a fuel cost savings of at least 23% over a standard gas turbine, while an ERGT with compressed air cooled turbine blades generates a fuel cost savings of at least 18%. Thus using steam as a turbine cooling medium results in an additional 5% of fuel cost savings.

Based on the above discussion, it can therefore be concluded that thermodynamically an ERGT with a steam-cooled turbine is a preferred option.

Thermodynamic analysis of the Option 1 cycle was extended to encompass the effects of using a fogger/humidifier at the inlet of the gas turbine. The use of the humidifier changed the inlet air condition from a temperature of 80ºF and a RH of 60% to a temperature of 70ºF and a RH of 100%. The analysis was conducted to study the effect of the humidifier on an ERGT with an air-cooled turbine and an ERGT with a steam-cooled turbine. The results indicate that installation of the humidifier at the inlet of the ERGT using an air-cooled turbine increases the electrical output by about 5%, while the exhaust temperature is decreased by less than 1%. In addition, the installation of the humidifier also increases the savings in fuel cost by an additional 3%. All these comparisons are made for the same HP/HT heat exchanger pressure drop.

Similar comparisons made for an ERGT using a steam cooled turbine reveal that using a humidifier at the inlet of the ERGT increases the electrical output by 5%, while the exhaust temperature drops by less than 1% and the fuel cost savings is increased by about 3%.

Thus over all, installing a humidifier at the inlet of an ERGT and using steam as the turbine cooling medium increases the electrical output by at least 14% and the fuel cost savings by an additional 7% when compared to an ERGT that uses air as a turbine cooling medium and does not have humidification equipment installed at the inlet. It can therefore be concluded that thermodynamically an ERGT with a steam-cooled turbine and a humidifier at the compressor inlet is a preferred option.

Based on the results of the thermodynamic analyses summarized above, the most favorable cycle configuration was determined to be Option 1, with air heater and combustor in series. Increasing the HP/HT Heat Exchanger exit temperature and reducing the pressure drop across the heat exchanger will help maximize the fuel cost savings. The reduction in pressure drop also increases the net electrical output, but an increase in the heat exchanger exit temperature causes the power output to reduce slightly. Of the various options considered, the most beneficial in terms of fuel cost savings and the net power output is the ERGT cycle that uses steam as the cooling medium for its turbine and has a humidifier installed at the inlet.

Titan 130 Package

Solar provided information regarding the T130 layout and installation and commissioning. These include arrangement and layout drawings of a standard T130 package and information regarding the various pumps, compression systems and cooling systems required. This information is presented in Appendix B, and was used to develop a preliminary layout (Figure 3.61) for the turbines and their utility and interface requirements in the DeRidder mill.

To minimize the time and cost required to develop an ERGT system based on the cycles discussed above, the ERGT should be developed by modifying an existing gas turbine rather than developing a completely new ERGT design. The suitability of the Titan 130 for modification to an externally recuperated configuration was therefore evaluated. Minimum modifications would include modifying the combustor and providing a passage for the airflow to and from the HP/HT heat exchanger. Candidate engines for such a modification include:

- A gas turbine that has a side mounted can combustor
- A gas turbine that has a silo combustor
- A gas turbine that has already been designed to run in a recuperated cycle

Each of these three configurations can easily support the airflow to and from the HP/HT heat exchanger. However, the present production Titan 130 gas turbine that Solar offers does not fit any of the above criteria and is unsuitable for the ERGT application in its present form. The effort required to modify the Titan 130 so that it can accommodate an ERGT cycle could be comparable to that of developing a new gas turbine. Further, given the present expected maximum heat exchanger exit temperature of 1450°F, Solar's developmental cost to produce two turbines for the current project would more than offset any fuel cost savings.

Equipment Modifications

An assessment was made of the state of the technologies involved in an ERGT cycle and what improvements are possible in the near term. The conclusions drawn from these assessments are presented below.

Combustor

For production engines at Solar, the combustor inlet temperatures range from approximately 690ºF to 1250ºF. Present material and premixed combustion technology limitations hinder the design of gas turbine combustors with inlet temperatures greater than 1450ºF and the design effort required to achieve higher inlet temperatures would be considerable.

HP/HT Air Heater

The present study has assumed that the HP/HT air heater can be operated at a pressure drop of 30 psi and an exit air temperature of 1400°F. With material evaluation and significant engineering effort the exit temperature most likely can be increased to 1500°F in the near term. However, the high pressure drop (30 psi) is incompatible with most modern gas turbines due to instability in compressor operation. To overcome this hurdle either a gas turbine that can accommodate the higher pressure drop needs to be identified or developed, or the heat exchanger pressure drop needs to be reduced below 15 psi.

Design of an entirely new gas turbine for an ERGT for the Forest Products Industry is presently not economically viable for Solar.

Reduction of the heat exchanger pressure drop poses technical, cost and practical challenges for the internal air heater in the stoker boiler, where there is normally a limited space within the furnace to fit the air heater. This is because both the surface area and volume of the air heater would have to be increased in order to reduce the airside pressure drop. This will reduce the airside velocity, which would adversely affect the convective heat transfer coefficient, causing the heat exchanger temperatures to rise. Higher metal temperatures will require improved materials to be used to maintain the useful life and operational safety of the heat exchanger.

The space constraint would be less of a problem for the external air heater, which will be a stand-alone heat exchanger between the gasifier and stoker boiler. There is also the option to use a firetube (hot syngas inside the tubes) design for this air heater, which would reduce the airside drop to well below 15 psi.

Steam Cooled Turbine

Although the ERGT using a steam cooled turbine looks attractive, developing such a turbine cooling system was judged to be prohibitively expensive for Solar. Of particular concern for Solar were smaller gas turbines where first cost is a critical buying criterion. Solar felt the capital cost would be substantially higher than regular air-cooled gas turbines due to the complexities of the cooling circuits and control systems and that the high development and manufacturing cost of such a system would nullify the fuel cost savings seen in the cycle analysis, especially as the combustor and heat exchanger designs limit the combustor inlet temperature to 1450°F.

This is not expected to be the case for all turbine designs and manufacturers. Steam-cooled turbines are already commercially available from several major turbine manufacturers, including GE. It should also be pointed out that steam cooling for this application could be applied to the combustor only and would be much simpler than if cooling of the vanes and blades were required. Finally, while steam cooling would increase the efficiency and reduce the cost of power generation, it is not a requirement for the ERGT particularly for the first demonstration engines.

Alternate Turbine Selection

The unsuitability of the Titan 130 and other Solar turbines for modification to the externally recuperated design resulted in the turbine study being expanded to include other candidate engines for development and demonstration of the ERGT. Discussions were conducted with six turbine manufacturers for this purpose, including GE, Rolls Royce, Pratt & Whitney, Hitachi, Siemens-Westinghouse and Alstom-ABB. These discussions led to a letter of interest/proposal from Alstom Power's Industrial Turbine Division in Houston, Texas. Alstom is a global provider of advanced technology in energy and transport infrastructure and are well known for their turbine and boiler technology within the U.S. Pulp and Paper Industry.

The turbine proposed by Alstom is their GT35P, which was specifically developed for the capability to export and import combustion air and has already been applied in an externally recuperated mode. The GT35P can therefore be much more readily modified for the project than Solar's Titan 130, which has never been produced or envisioned for an externally recuperated arrangement.

Alstom's proposal was reviewed with Boise Paper Solutions to insure that the GT35P turbine would be acceptable for the DeRidder mill. With its lower turbine inlet air temperature requirement and higher power output, the GT 35P is a better match for both the project's as well as the DeRidder mill's needs. Two Alstom turbines would produce about 34 MW of self-generated electrical power vs. about 28 MW from two Solar engines. The GT 35P was therefore selected as the basis for the ERGT and power generation system development.

3.3.5.2 Power Generation Process

As discussed above, the final power generation system configuration generates 34 MW of electrical power to increase the in-house generating capacity to approximately 82.5 MW. A simple cycle, externally recuperated gas turbine using natural gas as fuel with a heat recovery steam generator for cogeneration capability is found to be the most efficient power and steam generation system. The gas turbine parameters are:

No. of Gas Turbine Units	2
Type and Model	Alstom GT 35P
Gross Power	17 MWe each (17 MW nominal)
Station Auxiliary Power	581 kW (including HRSG and GT auxiliary)
Net Power	16.42 MWe
Fuel Input	Natural Gas
	17 bar (250 psia)
	3,972 kg/hr (1.1 kg/sec)
Compressor Pressure Ratio	12
Compressor outlet temperature	380°C (661°F)
Turbine inlet temperature	850°C (1562°F)

Brief descriptions of the gas turbine generator components are given in the following paragraphs.

Externally Recuperated Gas Turbine

The gas turbine is a single shaft machine and consists of an air inlet section, compressor section, combustion section and turbine section. The turbine is connected to a synchronous generator through a gear assembly.

Air Inlet Section

The amount of air intake required by a gas turbine generator is approximately 3 times that required by a reciprocating engine. The air entrance is designed to conduct incoming air to the compressor with minimum energy loss

Compressor Section

The primary function of the compressor section is to supply enough air to satisfy the requirements of the combustion burners at required pressure. The compressor must increase the pressure of the mass of air received from the air inlet duct and then discharge it to the burners in the required quantity and pressure.

A secondary function of the compressor is to supply bleed air for various purposes in the engine. The bleed air is taken from any of the various pressure stages of the compressor. The exact location of the bleed port will depend upon the external air heater pressure and temperature requirement. Varying degrees of pressure and heat are available simply by tapping into the appropriate stage.

Combustion Section

The combustion section contains the combustion chambers, igniter plugs, and fuel nozzle or fuel injectors. It is designed to burn a fuel-air mixture and to deliver combusted gases to the turbine at a temperature not exceeding the allowable limit at the turbine inlet. Theoretically, the compressor delivers 100 percent of its air by volume to the combustion chamber. However, the fuel-air mixture has a ratio of 15 parts air to 1 part fuel by weight. Approximately 25 percent of this air is used to attain the desired fuel-air ratio. The remaining 75 percent is used to form an air blanket around the burning gases and to dilute the temperature, which may reach over 2500º F. This ensures that the turbine section will not be destroyed by excessive heat.

The air used for combustion is known as primary air; that used for cooling is secondary air. Secondary air is controlled and directed by holes and louvers in the combustion chamber liner. Igniter plugs function during starting only; they are shut off manually or automatically. Combustion is continuous and self-supporting. After engine shutdown or failure to start, a pressure-actuated valve automatically purges any remaining unburned fuel from the combustion chamber.

Turbine Section

The turbine section consists of multiple stages located immediately next to the engine burner section. Turbines extract kinetic energy from the expanding gases as the gases come from the burners. They convert this energy in to shaft horsepower to drive the compressor and the generator connected to the gas turbine through a gearbox.

Exhaust Section

If the engine exhaust gases are discharged through an exhaust duct and expansion joint into a heat recovery steam generator. An exhaust duct is added to collect and straighten the gas flow as it comes from the turbine.

The expansion joint between the turbine exhaust duct and the HRSG relieves the stresses and prevents movement of the gas turbine or HRSG due to thermal expansion. Description of the heat recovery from the gas turbine exhaust is described in Section 3.3.6 Heat Recovery.

3.3.6 Heat Recovery

3.3.6.1 Heat Recovery Steam Generators (HRSG)

There will be two HRSGs, each connected to a GT35 gas turbine exhaust. Each HRSG will be an unfired, two-pressure, non-reheat, natural circulation, drum type with horizontal gas flow, complete with feedwater stop and check valves, relief valves, continuous and intermittent blowdown system, and economizer bypass. The medium pressure (MP), and the low-pressure (LP) sections will each consist of an economizer, evaporator, and superheater section.

The estimated duty of each HRSG is shown in Table 3.3.6-1.

Table 3.3.6-1 HRSG Performance

Item	Flow kg/hr (lbs/hr)	Temperature ºC (ºF)	Pressure bar (psig)
MP Steam	23,400 (51,600)	369 (695)	18.3 (250)
LP Steam	5,830 (12,850)	177 (350)	4.2 (60)

The HRSGs will be designed and constructed to operate within the maximum exhaust gas flow and temperature ranges of the CGT. The HRSG will be designed for outdoor installation.

The HRSG drums and internals will be sized for the required steam separation (purity) at the predicted HRSG performance data for the minimum HRSG drum pressure. In addition, the steam drums will be designed to accommodate surges associated with startup, shutdown, and rapid load changes.

The HRSG low-pressure superheater section will be designed for “dry” operation during startup and when there is maximum low-pressure steam demand. No steam flow through the low-pressure superheater will be needed for internal cooling because the low-pressure superheater tubes will be designed for the maximum expected gas temperature.

The HRSG scope of supply includes a single 40 m (~130 ft) high, 6.50 m (21.33 ft) diameter, A36 structural carbon steel stack with a divider plate at the base to accept gas flows from both HRSGs.

The HRSG will not have an integral deaerator. Steam cycle deaeration will be performed in a separate deaerator tank. The two HRSGs will share a common deaerator.

3.3.6.2 Modularization

To reduce field erection costs, the HRSG will be modularized with the following features:

- HRSG and associated piping and platforms will be shop-fabricated to the maximum extent transportable via highway.
- The HRSG will have top-supported MP sections; the LP sections will be bottom supported.
- Gas path insulation will be ceramic fiber blanket.

Figure 3.3.6-1 provides a schematic and expected performance of a single HRSG.

3.3.7 Balance of Plant

Major balance-of-Plant (BOP) system required for the biomass gasification-Based Power Generation plant include:

- Cooling Water System
- Compressed Air System
- Fire Protection System
- Electrical Distribution System
- Bark Conveying and Delivery System
- Natural Gas Supply System

3.3.7.1 Cooling Water System

Function and Description of the System

The purpose of the Cooling Water System is to supply cooling water to the various plant systems. Major systems and equipment requiring cooling water include:

- Gas turbine systems components-primarily lube oil coolers
- Compressed air system components-primarily inter-cooler, after-cooler, and lube oil cooler
- Gasification system components

A schematic diagram of the Cooling Water System is shown in Figure 3.3.7-1. The cooling water system equipment will consist of an open cooling tower loop and a closed loop. The closed loop is intended for supplying cooling water to certain components requiring demineralized water. The open loop will consist of a cooling tower, circulation pumps, and associated piping and valves. The closed loop system is provided with a surge tank, cooling water pumps, heat exchanger, chemical treatment equipment, and associated piping, valves and instrumentation.

Major Equipment

Cooling tower

No. of cells – 1
Type of construction – Mechanical Draft, wooden
Heat load – 4.5 x 10^6 kJ/hr
Circulating water – 55x10^3 kg/hr, 45C/25C

Cooling tower fans

No. of fans – 1

Rating – 2-speed, 25 kW, 380 V

Circulating Water Pumps

No. of pumps – 2-100%

Type - horizontal centrifugal

Rating – 22,500 kg/hr at 3.5 bar of discharge pressure, each

Motor - 1500 rpm, 5 kW, 380V each

Cooling Water Pumps

No. of pumps – 2-100%

Type - horizontal centrifugal

Rating – 22,500 kg/hr at 7.0 bar of discharge pressure, each

Motor - 1500 rpm, 10 kW, 380V each

Cooling Water Surge Tank

No. of tanks – 1

Capacity – $4.5m^3$

Material – carbon steel

Cooling Water Heat Exchangers

No. of heat exchangers – 2-100%

Type - straight tube, double pass, shell and tube

Rating – 25,000 kg/hr thru shell and 25,000 kg/hr thru tubes, 2.25×10^6 kJ/hr, each

Shell side temp. and pressure – 25C inlet; 45C outlet; 2 bar

Tube side temp. and pressure – 49C inlet; 30C outlet; 7 bar

3.3.7.2 Compressed Air System

Function and Description of the System

The Compressed Air System supplies clean, dry, oil free compressed air for operation of the pneumatic instruments and controls. The system also supplies compressed air to the different parts of the proposed improvements for miscellaneous compressed air activities.

A schematic diagram of the Compressed Air System is shown in Figure 3.3.7-2. The compressed air system includes air compressors, air receiver, instrument air pre-filters and after filters, air dryers, air headers, control valves and instrumentation. The compressors operate on receiver air pressure signal and keep the receiver charged to 125 psig. The receiver supplies service air on demand to the service air header network throughout the plant. The instrument air from the receivers is routed through the pre-filters, air dryers and after-filters to the instrument air header network through the plant. Individual control valves and instruments, which use instrument air, include individual pressure control valves to reduce air pressure as required.

An instrument air accumulator is provided for the receiver system. This accumulator contains adequate inventory of air to operate receiver system valves and instruments up to 30 minutes. This accumulator is supplied by the instrument air header on the tower and the supply line includes a check valve to maintain sir pressure in the accumulator in the event of a loss of instrument air header pressure.

System Design Basis

The Compressed Air System is designed to supply adequate amount of air for all the air-operated valves and instruments on a peak coincidental demand basis. The amount of service air is estimated based on the experience of this size plant.

Major Equipment

Air compressor

No. of compressors	2 – 100% capacity
Rating	1,700 std. cu. m per hr, 7.0 bar each
Motor	200 kW, 380 V, each

Instrument air receiver

No. of receivers	1
Capacity	9.7 m^3
Design pressure	7.0 bar, ASME Section VIII
Material	carbon steel

Instrument air accumulator

No. of accumulators	1
Capacity	9.7 m^3
Design pressure	7.0 bar, ASME Section VIII
Material	carbon steel

Instrument air pre-filter

No. of units	1 - 100% capacity set
Capacity	1,700 std. cu. m per hr
Design pressure	7.0 bar, ASME Section VIII

Instrument air dryer

No. of units	2 - 100% capacity sets
Capacity	1,700 std. cu. m per hr
Design pressure	7.0 bar, ASME Section VIII

Instrument air after-filter

No. of units	1 - 100% capacity set
Capacity	1,700 std. cu. m per hr
Design pressure	7.0 bar, ASME Section VIII

3.3.7.3 Fire Protection System

Function and Description of the System

The fire protection system consists of a fire water system designed to fight both indoor and outdoor fires of a conventional nature and a FM-200 system, including FM-200 fire extinguishers, for electrical fires in enclosed areas. The entire system is designed in accordance with NFPA codes.

The fire water system is an extension of the existing fire water system at the Boise Cascade facility at DeRidder. It consists of a dedicated 8 inch underground main connected to the existing main and is routed throughout the proposed improvements. Off the main header, 6-inch headers are routed to the GT units, dryers, gasifier complex, and the cooling tower. The cooling tower is protected by an automatic deluge system. The gas turbine and compressor lube oil areas are protected by automatic sprinkler systems.

Fire hydrants are provided to ensure 100% fire hose (2~i inch and/or 1V2 inch) coverage of all areas with combustible material. In addition, sprinkler systems are provided to the lube oil systems areas and the cooling tower.

All electrical equipment rooms, remote electronic stations are provided with total flooding FM-200 systems or FM-200 portable fire extinguishers. The FM-200 total flooding system in the equipment room is designed for concentration of 5 to 7 percent by equipment or sub-floor volume at 70°F. The turbine generator enclosure is provided with an automatic high pressure CO_2 system. This system is actuated by thermal detectors inside the generator enclosure.

Major Equipment

Piping and Valves

8 in. main header
6 in. sub-header
Stand pipes, fire hydrants, fire hose
Sprinklers

Fire Suppression and Extinguishing System

FM-200 total system or High pressure CO_2 fire extinguisher system

3.3.7.4 Electrical Distribution System

This section describes the principal electrical equipment and systems, their functions, and the general criteria upon which the design will be based for the proposed Advanced Gasification-Based Fuel Conversion and Electricity Production System.

Interconnection to Utility

The Boise Paper Solutions DeRidder facility is interconnected to CLECO utility transmission system through a switchyard. Currently, Boise facility does not export any power to the grid. With addition of two 17 MW GTs, there is potential for power export to the grid. This will require additional metering, protection, and controls to the existing switchyard and electrical system.

The Electrical Single Line Diagram is shown in Figure 3.3.7-3. The main step-up transformers for the GTs will be added to the power block and connected to the switchyard HV power circuit breakers by overhead lines. The HV-side potential transformers (PTs) and current transformers (CTs) required for metering, protective relaying, and generator synchronizing will have to be installed as part of the requirement for the interconnection to the Utility.

Gas Turbine System

Electric power will be generated by two 17 MW GT 35 Alstom turbines with a rated voltage of 15 kV. Generator output will be connected to the step-up transformer by a 17.5 kV isolated phase bus. Power will be transmitted to the plant utility system through the facility’s HV switchyard.

The switchyard HV circuit breakers will be controlled from the control room via the DCS. Utility dispatch control interfaces will be provided if required by CLECO.

Auxiliary Power

Per DeRidder Plant Operations, there is no spare power available from the station service transformer for the new gasification and gas turbine system. One two-winding wye-wye 132 kV/4.16 kV station service transformer will be provided rated to supply startup and normal operating power requirements to all new installations.

Distribution system power will be supplied from NEMA Class E2 4.16 kV latched contactors so that the electrical auxiliary power distribution system will not require operator intervention to restore power to the system in case of a system disturbance. LV switchgear will be fed from 4.16 kV/480 V pad-mounted oil-filled transformers. Power circuit breakers will be used to supply 480 V motor control center (MCC) buses. All 480 V loads will be supplied from 480 V MCC buses.

Emergency Power

In case of a total loss of auxiliary power, or in situations when the utility system is out of service, emergency power for generator critical loads such as the turbine lubricating oil system and for common critical loads such as the DCS will be supplied from a 125 V dc stationary battery.

Power Transformers

The main step-up transformer of each generator will be a 15 kV/138 kV, two-winding, delta-wye OA/FA/FA transformer. The station service transformer will be a separate 3.3 kV/132 kV, two-winding, wye-wye, OA/FA transformer. Final HV nominal ratings for the transformers will be selected based on the specific utility system parameters. Each main transformer will be provided with metal oxide surge arresters located adjacent to the HV terminals. The main step-up transformers will be oil cooled and have an air cell conservator system. Accessories will include a magnetic liquid level gauge, dial thermometer, winding temperature equipment, pressure relief device, sudden pressure relay, and bushing-mounted current transformers. The rating of the main step-up transformer will not limit the respective GT 35 turbine generator rated output at 0.85 lagging and 0.9 leading power factors and design conditions.

Medium and Low Voltage Switchgear and Motor Control Center (MCC)

The 4.16 kV switchgear/MCC assembly will be located indoors, will use vacuum interrupters, and will be rated to distribute the full output of the unit auxiliary transformer. The switchgear portion of the assembly will contain the incoming power circuit breaker and feeder breaker for the generator static start system. The switchgear circuit breakers will be electrically operated and have a stored energy mechanism. The switchgear will be bus-connected to the medium voltage MCC (MV-MCC) using a transition cubicle. Each HV compartment in the MV-MCC will contain an externally operated no-load isolating switch, current limiting fuses, slide-out contactor assembly with vacuum interrupters, and control power transformer.

The 480 V load centers will be single-ended, rated 480 V, three-phase, three-wire. Each will be supplied from a solidly grounded delta-wye, 4.16 kV/480 V, oil-cooled transformer. The load centers will use manually operated air-break power circuit breakers. Each power circuit breaker will have a solid-state trip device. Bus volts, incoming amperes, and individual feeder amperes will be displayed on analog meters mounted on the front of the load centers. The load centers will supply power to 480 V MCCs and 480 V panel boards.

480 Volt Motor Control Centers

MCCs will be rated 480 V, three-phase, three-wire and will supply 480 V non-motor loads, motors rated 480 V from 1 hp up to and including 175 hp, and lighting and distribution panel transformers. Thermal-magnetic molded-case circuit breakers will be used for non-motor loads. Each motor starter will consist of a three-pole magnetic-only molded-case circuit breaker, three-phase overload relay with heater elements, three-pole contactor, and control power transformer. Indicating lights will be mounted on the front of each motor starter compartment door. MCC-mounted 480/220 V three-phase, four-wire panel boards will be used to supply power to space heaters and small power loads associated with facility power production processes. MCCs will consist of vertical sections joined together to form rigid, freestanding assemblies. MCCs located outdoors will be installed in supplemental weather-resistant enclosures.

125 Volt DC System

The 125 V dc system will consist of a bank of batteries with static battery chargers, a switchboard, and a panel board. This system will supply dc power to the generator, DCS, switchgear, protection relay panels, and other critical dc loads. DC power to the generator and its auxiliaries will be supplied from a battery and two chargers provided by the supplier as a part of the generator scope of supply.

Uninterruptible Power Supply System

One solid-state uninterruptible power supply (UPS) will supply 220 V ac single-phase critical AC loads. The DCS operator stations will be supplied UPS ac; the power requirements for the DCS controllers and I/O will be met using DC/AC/UPS power as required. The UPS system will include an inverter, constant voltage transformer (CVT), static transfer switch, manual bypass switch, and panel board and will be provided as part of the generator scope.

Electrical Protection, Metering, and Controls

Protective relaying, metering, and instruments will be provided for proper interface and operation and to monitor equipment performance.

Protective Relaying. Protective devices will be coordinated to the extent possible so that electrical disturbances (fault, overload, etc.) are interrupted at the point nearest the fault, with the next upstream protective device providing backup protection. Ground fault protective devices will trip the respective breaker or starter. Protective devices will operate through a lockout relay (86) or equivalent device or circuit to prevent automatic restart/reclose of the equipment.

Protective devices will be rated for the maximum available fault current.

Current sensing relays will be the drawout case type to permit testing and calibration without disrupting the current transformer secondary circuit.

Metering. Shorting-type terminal blocks will be provided to allow instruments to be removed without disrupting current transformer circuits.

Relaying class accuracy for voltage and current transformers will be considered adequate for panel meter applications.

Controls. The generator will be synchronized automatically through its synchronizing system, which is included as part of the generator package. The synchronizing system will control turbine speed/generator frequency, generator voltage, and breaker closure.

The incoming and static start circuit breakers on the 4.16 kV bus will be controlled through the DCS and by locally (switchgear) mounted, four-position, pull-to-lock hand switches.

Breakers and starters that control process loads will be controlled from the control room. Equipment such as HVAC, air compressor, sump pumps, CEMS, etc., will be locally controlled only, with no control room control. Only single composite equipment alarms are included.

The 4.16 kV latched contactors to load center transformers will be controlled locally.

The distribution system control will be local and not expected to be operated from the control room. Should a breaker trip, the operator will go to the breaker front to identify the cause, correct the problem, and reset/reclose.

The 400 V load center incoming and feeder breakers to transformers, panels, and MCCs will be controlled locally at the breakers.

Motor feeders will be controlled either through the DCS or an independent process controller. No locally mounted control will be included at the starter or at the motor location. Motor running/stopped indication will be provided at the MCC.

Electrical Design Criteria/General Requirements

General

The electrical systems, equipment, materials, and their installation will be designed in accordance with applicable industry codes and standards; local, state, and federal regulations; project design criteria; and other requirements as specified in this section.

The following general criteria will be used in designing the electrical system:

- Utility voltage variation maximum is ± 5 percent of nominal kV.
- Utility frequency variation maximum is ± 0.5 percent of 60 Hz.
- Utility available short-circuit is "infinite" for auxiliary distribution system bus rating.
- Utility actual available short-circuit is used for ground grid sizing and HV equipment ratings.
- Phase rotation is A-B-C counterclockwise.
- Equipment short-circuit ratings are based on the maximum available under all operating conditions. No additional margin is provided unless inherent in the final equipment selection.
- Equipment basic insulation levels (BILs) will be ANSI standard:
- Electrical clearances are per the National Electrical Safety Code, ANSI C2.

No security system or equipment is provided.

Motors

Motors will be the squirrel-cage induction type suitable for full voltage, across-the-line starting. Enclosures will be weather-protected Type II (outdoor), open drip-proof (indoor), or totally enclosed fan cooled (TEFC), as required for the specific application. Motors will be rated to provide at least 5 percent margin between the required driven load brake horsepower (bhp) and motor rated horsepower at a service factor of 1.0. Where the 5 percent margin would require using the next larger size motor, an allowable option is to provide a 1.15 service factor motor. Credit will not be taken for service factor capability above 1.0 for steady-state operating conditions.

Motors rated 3/4 to 175 hp and fractional horsepower reversing motors (e.g., motor-operated valves) will be rated 480 V. Motors 1/2 hp and smaller will be rated 220 V ac. DC motors will be rated for 220 V or 215 V, as required by the specific application.

Motors rated 25 hp and above for use in outdoor applications other than HVAC will be provided with space heaters to prevent condensation formation during nonoperational periods. The power for space heaters will be provided from MCC distribution panelboards or control power transformers rated for the space heater.

Grounding, Cathodic Protection, and Lightning Protection

The facility grounding system will consist of buried stranded copper conductors and ground rods, as required. The grounding system design will be based on a maximum available fault current of 20,000 A in the switchyard. Fault duration will be considered to be 20 cycles. Credit will be taken for parallel paths, for multiple connections to equipment, and for fault current returning to the remote sources via transmission line static wires. Compression-type connectors that meet the requirements of IEEE 837 will be used for the buried connections.

Equipment and electrical systems in the power block will be grounded in accordance with the National Electrical Code (NEC).

Equipment grounding is planned to be in accordance with the NEC. Cable tray will be grounded by the tray itself. RSG conduit is self-grounding, and duct banks/trench should use a single conductor per run. The DCS and its I/O cabinets are to have the equipment and the signal circuits grounded at the cabinet location. The use of data highway obviates the need for a "tree"-type grounding system.

Because of the potential hazard should a leak develop, cathodic protection will be provided for the buried coated carbon steel natural gas pipes. Cathodic protection will not be included for any other buried pipe. If the site's soil resistivity is less than 200 ohm-meter; then, cathodic protection will be provided for all buried coated carbon steel pipes.

Lightning protection for buildings and structures will be generally in accordance with NFPA 780. Lightning protection will be provided by a single ionization-type preventor

Lighting Systems

The existing DeRidder plant Lighting system will be used.

3.3.7.5 Bark Conveying and Delivery System

Function and Description of the System

The bark conveying and delivery system is shown in Figure 3.3.7-4 and is described below. Bark is currently delivered from the woodyard to No. 2 Boiler via an existing conveyor. In the proposed design this conveyor will be integrated with additional conveying, drying and screening equipment in a stepwise fashion as follows:

Implementation Step 1 – In this step, only the bark for the gasifier is dried. Wet bark from storage is carried on the existing conveyor up to junction box #14. The junction box separates the bark into two streams. One stream continues on the existing conveyor leading to No.2 boiler. The second stream transports the bark to be dried for the gasifier, with the help of a conveyor, to dryer #1. Dried bark is then transported to junction box #15 linking the existing conveyor leading to No. 2 boiler. This junction box is located downstream of junction box #14. Junction box #15 then separates the dried bark into two streams. Most of the dried bark is transported (on demand from the gasifier metering bin) to a screening and sizing machine, where it is first shredded for size reduction and then screened to separate out oversize and rejects unsuitable for the gasifier. The oversize bark is conveyed back to the existing conveyor leading to No.2 boiler. The oversize material may also be recycled back for re-shredding and re-screening. The dried, sized bark is fed to the gasifier metering bin. In the event that the gasifier metering bin is approaching a full condition, junction box 15 will divert the dried bark flow back to the existing conveyor leading to No. 2 boiler, and the dryer will be throttled back to minimum-throughput hot standby operation until demand for fuel from the gasifier system is restored..

Implementation Steps 2 & 3 – In these steps, the entire quantity of bark is dried using dryers #1 and #2. Wet bark from storage is carried on the existing conveyor up to junction box #14. The junction box then diverts the entire quantity of bark to the dryers. Dried bark is then transported to junction box #15, which splits the dried bark flow as necessary to meet the gasifier and boiler bark demands. Operation of the screening/sizing system and transport to the gasifier metering bin is as described in Step1. In the event that one dryer is out of service, the dried and wet bark streams will be routed as in Step 1. In the event that both dryers are out of service, bark to the gasifier metering bin is stopped and wet bark is routed through junction box 14 on the existing conveyor to No. 2 boiler.

Metering and Feeding to Gasifier. Bark for the gasifier is fed in to two vibrating hoppers each with a variable opening gate. The gates allow the bark to drop to two conveyor belts. The two conveyor belts transfer the bark to two metering bins for weighing and finally feeding the gasifier.

3.3.7.6 Natural Gas Supply System

Function and Description of the System

As described in Section 3.2, Plant Description, additional electricity is generated on-site using externally recuperated gas turbine generators. Natural gas provides about 2/3 of the heat required for the gas turbine generators. Estimated gas requirement is approximately 80,000 cf/hr per gas turbine.

Pressurized natural gas is supplied to the gas turbines with the help two multi-stage compressors, one for each gas turbine. Supply of natural gas to the compressors is from an on-site metering station located approximately 1 km from the compressors. Gas would be supplied at 4.5 bar. through a 7.5 cm diameter pipeline.

Design Conditions:

Inlet Gas Pressure: 4.5 bar
Discharge Gas Pressure: 18 bar

Natural Gas compressor

No. of compressors	2
Type of compressor	Equivalent of ARIEL JGH4 four throw double acting compressor with air-cooled lubricated cylinders and packing.
Rating	6,000 std. cu. m per hr, 890 rpm, each
Motor	350 kW, 4000 V, each

Compressor Accessories

Inlet scrubber with automatic liquid discharge controls, Lubrication system, variable volume clearance pockets, for the first stage cylinders, duplex oil filter, vibration transmitter, oil temperature RTD, and oil pressure transmitter.

Inter-stage scrubber with automatic liquid discharge controls

Air-cooled heat exchanger for inter-stage gas cooling, discharge gas cooling, and compressor oil cooling with electric motor driven fan, hot dip galvanized structure, and vibration transmitter.

Coalescing filter/discharge scrubber with automatic liquid discharge controls.

Fabricated steel skid base with 1/4' thick raised pattern floor plate, concrete fill beneath the compressor & driver, and machined mounting surfaces for the compressor & driver.

Piping assembly with welded & flanged connections for 2" pipe size and larger per ASME/ANSI B 31.3. Piping 1-1/2" size and smaller would be XH A-106B threaded with 2000# forged steel fittings. The piping assembly will include a class 150 cast steel gate valve inlet gas isolating valve, a Norriseal Controls cast steel Class 300 discharge piston check valve, a 1" manual blow down valve, a 1" Kimray 1400 SMT automatic discharge pressure control valve, and a class 300 cast steel discharge gate valve isolating valve.

PLC control panel with an Allen-Bradley SLC-500 PLC and PanelView 1000 HMI. The control panel would be mounted on the compressor skid and would be assembled from components UL Listed or CSA Certified for a Class 1, Division 2 hazardous area.

3.3.8 Construction Schedule

A preliminary plant construction schedule is developed and presented in Figure 3.3.8-1. The objective of the preliminary schedule is to:

- Estimate the total duration required for implementing the project
- Identify the long lead time items

The schedule shows the time needed for implementing Step 1 of the project. Schedules for Steps 2 and 3 are not shown as the initiation of these two steps are not known in relation to Step 1. Since all the major equipment of Steps 2 and 3 are duplicates of those of Step 1, it may be stated with a first order approximation that construction schedule for Steps 2 and 3 equipment will be similar to that of Step 1 equipment.

The unique features of the construction schedule are discussed in the following paragraphs.

The overall schedule for the plant is estimated to be twenty-seven months including design, engineering, fabrication, procurement, installation, start-up, testing and commissioning. One notable exception is the time required for development of gas turbines. Development of turbines will necessarily have to precede the schedule indicated above. This development duration is not known yet. The schedule is somewhat conservative to ensure minimum interruption of the operation of the existing plant.

Gasification plant equipment items comprise the longest lead-time items. Design, engineering, fabrication, and procurement will require 18 months. Installation of this equipment will need 9 months. However, there will be an overlap of three months with fabrication schedule. Thus the effective schedule for the gasification equipment is 24 months.

Schedule for all the remaining equipment is enveloped by the schedule for gasification plant equipment as shown in Figure 3.3.8-1.

Schedule for start-up, testing and commissioning is conservatively estimated to be three months. However, it is likely that this duration can be shortened somewhat.

3.4 PLANT OPERATION

The purpose of this section is to provide guidelines for the startup, operation, and shutdown and emergency steps during major upset conditions of major systems and components that will be installed at the DeRidder plant. These guidelines do not substitute for detailed operating procedures and required operator training to be conducted prior to commissioning of the facility.

The major systems and components of the proposed facility are:

- External Air Heater AH-1
- Internal Air Heater AH-3
- Bark Dryer System
- Gasifier
- Gas Turbine Generator
- Dual Pressure Heat Recovery Steam Generator System
- Plant Auxiliaries Systems

These systems will be integrated with the existing bark boiler, DeRidder plant steam and electrical systems. However, each of the dryer, gasifier, gas turbine and HRSG will have its own control systems and will be operated independently but in conjunction with the rest of facilities at DeRidder plant.

Bark Dryer System

The bark dryer system is described in Section 3.2.1. Specific dryer startup and operation will be per dryer vendor procedures.

Gasification Plant Control And Operation

Control Philosophy: The Gasification Plant will be operated on load control. The main control functions of the gasifier are described below.

Gasifier Load Control

The Gasification Plant load is controlled by changing fuel feed rate. The changing fuel feed rate will vary the airflow to the gasifier according to the preset fuel/air ratio.

Gasifier Temperature Control

The gasifier bed temperature indicates the fuel/air ratio in the gasifier. Increasing temperature in the fluidized bed indicates higher airflow than required, decreasing temperature indicates less airflow than required. The temperature control adjusts the gasification airflow to compensate the fluctuations of the fuel quality and to keep the set value of the bed temperature.

Air Feed Control

The gasifier control system calculates the set values for the total airflow based on the fuel/air ratio set value. The total airflow is distributed then to the grid and ash removal pipe. The grid airflow maintains fluidization; the ash removal airflow controls bed material and ash removal from the bed area. Mixed with the air supplies also steam can be fed to the gasifier through the grid and with the ash removal air in case of excess temperatures, but not required for operation at design conditions.

The ash removal airflow is calculated by the gasifier control system based on the bed conditions (bed height or density) and ash removal pipe temperature.

Product Gas Pressure Control

The pressure control valve before the product gas injection to the boiler controls the gasifier gas pressure. This valve will keep the pressure at the required 15.2 psia / 1.05 bara pressure. When load changes the airflow and gasifier pressure will vary accordingly.

Product Gas Temperature Control

Water injection in the product gas line or adjusting the combustion air floe to the syngas combustor are used to control the product gas temperature to AH-1.

Internal Air heater AH-3 Operation

Air Flow will be started to AH-3 from the Gas Turbine GT-2 compressor by starting GT-2 on natural gas prior to starting No. 2 Power Boiler. As the boiler comes online and more heat is available from AH-3, natural gas flow to the GT-2 combustor will be reduced accordingly. In the event that GT-2 must be taken offline while the boiler is operating, sufficient air is available from GT-1 to satisfy the air requirements of both AH-1 and AH-3. In this event, the natural gas flow to the GT-1 combustor will be further reduced to maintain the required turbine inlet temperature.

External Air Heater AH-1 Operation

Air flow will be started to AH-1 from the Gas Turbine GT-1 compressor by starting GT-1 on natural gas prior to gasifier startup. As the gasifier comes online and more heat is available from AH-1, natural gas flow to the GT-1 combustor will be reduced accordingly. In the event that GT-1 must be taken offline while the gasifier is operating, sufficient air is available from GT-2 to satisfy the air requirements of both AH-3 and AH-1. In this event, the natural gas flow to the GT-2 combustor will be further reduced to maintain the required turbine inlet temperature.

Gasification Plant Operation

The Gasification Plant can be operated with all syngas to the flare for brief periods when the Stoker boiler goes off-line but must have air flow to AH-1. Gasification Plant operation also requires auxiliaries like electrical, power, cooling water, etc.

Gasification Plant operation includes the following procedures:

- Start-up
- Shutdown
- Full load operation
- Part load operation
- Load changes

Gasification Plant Start-Up Sequence

The Gasification Plant start-up sequence includes the following steps:

- The gasifier is in cold, ready-to-start condition.
- Air will be supplied to the gasifier through the start-up heater pressure vessel.
- Gasifier start-up heater burning natural gas is in ready-to-start condition.
- Gas cooler air is supplied by the gas turbine.
- Start-up-heater start at 50% capacity. The flue gas is flared.
- Start-up heater in continuous operation with increased capacity. Bed material will be fed to the gasifier.
- Gasifier temperature is raised to the level where fuel feed is reasonable. Fuel feed starts.
- Heating is continued. Ignition of fuel occurs.
- Shifting from combustion to gasification after fuel ignition. The product gas is flared.
- Start-up heater shutdown.
- Fuel feed is gradually increased.
- Gasifier operating at full pressure and temperature at minimum load.
- Gasifier load increased from minimum load to full load.

Gasification Plant Shutdown Sequence

The controlled shutdown of the Gasification Plant includes the following steps:

- Gasifier full load reduced to minimum load.
- At minimum load product gas will be switched from boiler to flare.
- Gasifier pressure and load reduced, sufficient bed removal established.
- Gasification air replaced with steam and inert gas. Continuous bed removal.

Gasification Plant Base Load Operation

At base load operation the Gasification Plant is operated according to load control.

Gasification Plant Load Changes and Part Load Operation

The fluidization velocity limits the part load operation of the Gasification Plant to a minimum gasifier load of about 50%. When changing the load from full load to part load, the fuel input to the gasifier decreases proportionally with decreasing air feed, according to the product gas set point.

Gas Turbine Operation

The gas turbine generator startup operation is a single step operator initiated action. This can be achieved through a local start at the GT control panel or through control room DCS operation. The start command will work when all the required prerequisite, such as combustor temperature, lube oil pressure and temperature conditions, time delay since last start, HRSG drum level, feedwater pump operation, etc. are satisfied. The turbine will go through the normal purge, speedup, fuel injection and fire sequence and arrive at no load status. Once the gas turbine has achieved stable speed, it is ready for generator synchronization. This is also operator-initiated action, achieved through closure of the generator breaker. After generator synchronization, the gas turbine load can be increased manually or through a pre-set load curve to a desired level.

The two gas turbines are completely independent and have independent controls and startup system. Hence, the turbine operation described here is same for both turbines.

Operating Procedure For Abnormal Conditions

No Start. If it is noted during a start procedure that a no light off is indicated within 10 seconds after fuel is applied, the start procedure should be discontinued.

Before attempting a second start, dry motor the turbine for 60 seconds for purging the engine.

If the second start attempt fails, no further start attempt should be made before the cause of the start failure has been determined and corrected.

If the failure to start is attributed to either ignition system, it should be recorded, investigated and remedied at the earliest opportunity.

Unsatisfactory Starts. If an unsatisfactory start should occur, it will most likely be accompanied by one of the following conditions:

A. Hot Starts. A potential hot start is indicated by an abnormally rapid exhaust gas temperature (EGT) rise after light off. By monitoring fuel flow and EGT, a hot start can be anticipated before the 725 C limit is exceeded.

Hot Starts may be caused by:

- Inadequate starter air pressure, resulting in low compressor airflow.
- Faulty starter valve action, preventing proper operation of starter, with same result as item (a)
- Premature starter deactivation
- Incomplete purging of fuel in the combustion chamber from the previous start attempt. Faulty pressurizing valve (hung open) resulting in fuel, under low pressure.
- Faulty turbine control resulting in incorrect sequence scheduling.
- Incorrect scheduling of Inlet Vane Guide (IGV)

B. Hung Starts. A hung start is identified by light off followed by abnormally slow acceleration and rpm stabilization below idle. A hung start may be result of fuel scheduling being either too lean or too rich. A lean hung start is associated with low fuel flow and proportionally low EGT. A rich condition can be recognized by a high fuel flow and an EGT rise, which may tend to develop into an over-temperature condition and possible compressor stall.

Oil System Malfunction. Exercise caution when operating a turbine with oil pressure outside the normal pressure range. Oil pressure fluctuations, or pressure shifts exceeding +/- 5psid (69kpa diff.) is cause for investigation.

Turbine Malfunction

A. the turbine should be shutdown as soon as possible after discovery of a serious malfunction. Severe damage to the turbine can result if turbine operation is continued with a critical deficiency. The longer the delay between detection of a malfunction and turbine shutdown, the more severe will be the resulting damage.

B. The following indications should be recognized as symptoms of a serious turbine malfunction and/or impending failure:

- An increase in turbine vibration accompanied by higher than normal EGT or fuel flow.
- Repeated or uncontrollable turbine stalls.
- Loss of thrust.

- A shift in turbine parameters, or in the relationship of one parameter to another during steady state operation.
- Oil pressure increase or decrease of +/- 5psi or more from the normal steady state operating pressure, and/or an increase in oil temperature, or indications of oil filter bypass.
- Any combination of the foregoing symptoms.

Emergency Operating Procedure

Turbine fire

Internal Turbine Fire. An internal turbine fire may be evidenced by failure of EGT to decrease after turning the fuel off (post shutdown burning). In such case, the turbine should be isolated from the fuel supply. If the fire cannot be extinguished by motoring the turbine or if motoring is not possible, close the fuel shutoff valve and extinguish the fire with fire fighting equipment.

External Turbine Fire: An external turbine fire will be indicated by the fire warning system. The automatic fire retardant system will be activated and the turbine will be shutdown.

Turbine Failure/Malfunction

A malfunctioning turbine is evidenced by abnormal turbine parameters, noise or vibration. Continued operation with a known turbine malfunction may lead to turbine failure. In the event of an actual or impending failure, the turbine should be shutdown by turning the fuel and ignition off. If the turbine operates normally at idle, it should be allowed to idle for 3 minutes prior to shutdown if practical. Restart attempts without through investigation are not advisable, as further damage may result.

Heat Recovery Steam Generator Operation

Heat recovery steam generator (HRSG) operates with the associated gas turbine. The standard boiler controls for feedwater, drum level and boiler pressure control the HRSG operation. The HRSGs are dual pressure/ dual drum. Medium and low-pressure steam generated in the HRSG is tied into respective steam headers in the plant.

During startup, HRSG drum level is kept at low level and vents are kept open. The heat from the GT exhaust is transferred in the HRSG to make steam. When drum pressure rises to 10-15 psig, high point vents on the HRSG and steam lines are closed and steam is fed to the steam headers.

When HRSG drum level falls below low-low level, the master control will trip the GT. Restart of the HRSG and GT is achieved by re-establishing the water level and going through the turbine start sequence as described above.

3.5 PLANT PERFORMANCE

This section summarizes the overall plant performance of the advanced gasification-based power system.

3.5.1 Feed/Product Summary

The overall energy a performance of the modified plant is summarized in comparison to current mill operations in Table 3.5.1-1. The current mill case considers fuel feed to and products from No. 2 Boiler and other gas fired boilers on site as required to match the modified plant steam output. The Step 1, 2, and 3 cases consider fuel feed and products information for the modified plant, including the boiler, gasifier, dryer, ERGTs, and HRSGs. Current mill performance is derived from mill data for No. 2 boiler averaged over a one-year period ending in May 2003. Fuel and products/emissions data for the modified plant is given for all 3 implementation steps.

Table 3.5.1-1 Baseline and Modified Plant Performance Comparison

Input/Outputs	Units	Current No. 2 Boiler	Modified Plant		
			Step 1	Step 2	Step 3
Heat Input to Gasifier	MMBtu/h	0.0	180.5	180.5	180.5
Heat Input to Boiler	MMBtu/h	313.0	378.7	354.0	436.0
Total Heat Input to Boiler + Gasifier	MMBtu/h	313.0	414.8	390.1	472.1
Bark Heat Input to Gasifier	MMBtu/h	0.0	180.5	180.5	180.5
Bark Heat Input to Boiler	MMBtu/h	204.4	234.3	209.6	291.6
Total Bark Heat Input	MMBtu/h	204.4	414.8	390.1	472.1
Natural Gas Heat Input to Boiler	MMBtu/h	108.6	0	0	0
Natural Gas Heat Input to ERGT	MMBtu/h	0.0	143.3	143.3	266.6
Nat. Gas Heat Input to Bark Dryer	MMBtu/h	0.0	6.7	0.0	0.0
Nat. Gas Heat Input to Other Boilers	MMBtu/h	*164.0(267.4)	0.0	0.0	0.0
Total Natural Gas Heat Input	MMBtu/h	*272.6(376.0)	150.0	143.3	266.6
Steam Produced From No. 2 PB	Klb/h	212.5	250.0	250.0	250.0
Steam Produced from HRSGs	Klb/h	0.0	64.5	64.5	128.9
Steam Produced From Other PBs	Klb/h	*102(166.5)	0.0	0.0	0.0
Total Steam Produced	Klb/h	*314.5(378.9)	314.5	314.5	378.9
Electricity Produced from Bark Steam	MW	7.5	12.7	12.7	12.7
Electricity Produced from Gas Steam	MW	3.3	0.0	0.0	0.0
Electricity Produced from ERGT	MW	0.0	17.0	17.0	34.0
Total Electricity Produced	MW	10.8	29.7	29.7	46.7

*For steam production equivalent to the modified plant in Steps 1&2 (Step 3)

The primary objectives of the project are to increase woodwaste (bark) utilization, decrease fossil fuel use, and increase self-generated power in the mill. It can be seen from Table 3.5.1-1 that bark utilization in all cases for the modified plant is significantly higher (90-130%) than current operation. For equivalent steam production, the total natural gas usage for the modified plant is in all cases lower than current operation. For Implementation Steps 1 and 2 gas usage is reduced by 45-47% and for Step 3 by 29 %. While the current average steam production from No.2 Boiler is about 213,000 lb/h, the combined steam production from the boiler and HRSGs will be about 315,000 lb/h for Steps 1 and 2 and 472,000 lb/h in Step 3. In Step 3 this steam production increase will be accomplished at a GHRR equal to the original boiler design. Finally, Self-generated electricity will be increased from the 10.8 MW currently attributable to No.2 Boiler to 29.7MW in Steps 1 and 2 and 46.7MW in Step 3, increases of 175% and 332%, respectively.

3.5.2 Efficiencies

The design efficiency of No. 2 Boiler (heat to generate and superheat steam ÷total heat input to the boiler) is 70%. Calculated boiler efficiencies for the modified plant in Steps 1 and 2 (steam) and Step 3 (cogeneration-steam plus air) are 74.2, 79.4 and 78.9, respectively due to a combination of factors:

- improved carbon burnout in the boiler through partial combustion of a portion of the bark in the gasifier before the boiler, which reduces combustion at the grate back to below its original design value of 813,000 Btu/hr-ft^2
- operation of the boiler with lower excess air as a result of improved fuel and combustion air staging
- Feeding 20% moisture bark to the boiler instead of 52.5%, using more high level heat within the boiler for heating steam and air for power generation rather than vaporizing water from the fuel

For the fully implemented case in Step 3, the thermal efficiency of fuel to electricity conversion is 79.8% in the cogeneration mode, 5% above the project goal of 75%.

3.5.3 Emissions Inventory

3.5.3.1 Upgrading and Modifying the Existing #2 Bark Boiler

Under the proposed project the existing facility will be modified and or upgraded. Under the current regulations, a construction air permit will be required in accordance with 40 CFR Part 51 (Best Available Retrofit Technology, BART).

Also, the boiler may have to comply with the current limits as well as conduct a new source review/PSD review in order to determine if a net increase in emissions is significant; that is, above certain thresholds for any of the six criteria pollutants listed in the PSD rule. This option could trigger PSD permitting, which would require a Best Available Control Technology analysis to determine appropriate pollution control retrofits.

Table 3.5.3-1 lists existing permit conditions for the #2 Bark Boiler.

Table 3.5.3-1: #2 Bark Boiler Data

Item	Units	
Heat Input	MMBtu/hr	454.29
Bark Input	MMBtu/hr	454.29
Nat. Gas Input	MMBtu/hr	262
Boiler Fuel		Bark
Stack Gas Flow	Cu. Ft./Min	198,000.00
Stack Temp	Deg F	155.90
Stack Exit Velocity	ft/sec	46.50
Stack Diameter	Ft.	9.50
Stack Height	Ft	178.15

Table 3.5.3-2: #2 Bark Boiler Air Permit*

	Average	Max	Avg.
	lbs/hr	lbs/hr	lbs/MMBtu
PM-10	26.20	26.20	0.06
SO2	138.88	148.37	0.31
NOx	71.73	72.01	0.16
CO	149.92	149.92	0.33
VOC	43.36	43.49	0.10
Lead	0.03	0.03	0.00

* These emissions are based on the current maximum heat input capacity of the boilers and single point source permit for source ID 79-01 - #2 Bark boiler.

3.5.3.2 Co-Generation Facility at the DeRidder Plant

The proposed cogeneration facility at the Boise DeRidder plant will have advanced gas turbines with external recuperation and heat recovery steam generators for cogeneration. The external recuperation of combustion air will increase the turbine efficiency and cogeneration will provide supplemental steam for the DeRidder plant operation.

For new cogeneration facilities to be installed/constructed, a construction air permit will be required. With improvement in combustor design, the expected NOx levels from the gas turbine will be less than 25 ppm @ 15% O_2. This NOx level can be further reduced to 9-15 ppm @ 15% O_2 using SCR with ammonia injection. This level of NOx meets the current emission standards set by EPA. Since NOx limits are also adjusted for the cycle efficiency, the proposed externally recuperated gas turbine with expected higher efficiency will meet the required emissions regulations. Details on emission regulations for stationary gas turbines are provided in 10CFR60 sub part GG.

Since natural gas is the only fuel whose combustion products will pass through the turbines, SO_2 and PM10 are not an issue and are not addressed here.

3.5.3.3 Gasifier and the Bark Dryer

There is no emission from the gasifier and the bark dryer. The VOC exhaust from the bark dryer will be fed directly into the gasifier as part of the combustion air. The low Btu syngas from the gasifier will be used as the supplemental fuel in the No. 2 Bark Boiler. Hence, no air permits will be required.

3.6 PLANT LAYOUT

As mentioned earlier, the proposed project is envisaged to be an integral part of and located within the premises of the Pulp and Paper Mill owned by Boise Cascade Corporation Southern Operations and located at DeRidder, Louisiana. The project equipment is located near No. 2 Bark Boiler to facilitate close coordination of the proposed project operation in conjunction with the No. 2 Bark Boiler.

The configuration and location of the equipment is shown in the layout drawings, Figure 3.6-1 in 3 sheets. The drawings also show the relative locations of the existing pulp and paper mill equipment.

Availability of space.

The space at the mill is at a premium due to the compact arrangement of the existing equipment. For this reason, the proposed project equipment is located in a somewhat scattered manner to utilize whatever space is available in the vicinity of the No. 2 Bark Boiler. The spaces found to be adequate and suitable are generally located in areas north of the boiler. The general approach in locating the equipment was to minimize lengths of the various interconnecting piping, ducting, and bark conveyor belts.

Gasification Equipment and External Air Heater. Relative arrangement of this equipment is essentially same as that discussed in Section 3.3.2 with the exception of the combustor and the equipment downstream of the combustor. This latter equipment is somewhat rearranged to fit the available space and to minimize lengths of the various interconnecting pipes without altering the functionality of the system.

Gas Turbine Generator #1 (GT-1). GT-1 and associated equipment are closely related to the gasifier because the gasifier thermal energy is utilized to preheat the combustion air supply to the gas turbine. It is desirable that the lengths of the high temperature air pipes be kept at a minimum to minimize heat loss and cost. Thus the GT-1 and the associated equipment are located in the close proximity to the gasifier.

Gas Turbine Generator #2 (GT-2). GT-2 and associated equipment are closely related to the Bark Boiler #2 because the boiler heat is utilized to preheat the combustion air supply to the gas turbine. It is desirable that the lengths of the high temperature air pipes be kept at a minimum to minimize heat loss and cost. Thus the GT-2 and the associated equipment are located in the close proximity to the bark boiler.

Wet Bark Dryers. These dryers are physically large and require a large amount of real estate. Moreover, the dryers are closely associated with the gasifier and the bark boiler because dried bark is supplied from the dryers to the gasifier as well as the bark boiler. In addition, the hot and cold flue gas from the boiler is ducted between the two. The most suitable place available is between the gasifier and the bark boiler.

The dryers are located on two connected elevated platforms approximately 25 ft. above ground level. This is done to avoid interference with the ground level equipment and to keep the roadway underneath free for vehicular traffic.

Piping, Ducting, and Conveyors

Major piping, ducting, and conveyor belts shown in the layout drawings include:

- Hot and cold piping between GT-1 and air heater AH-1 (Nos. 12 and 13 on the layout drawings)
- Product gas piping from syngas cooler to bark boiler #2 (Nos. 10 on the layout drawings)
- Conveyor from conveyor junction box (No. 14 on the layout drawings) to wet bark screening and sizing machine (No. 14 on the layout drawings)
- Conveyor from wet bark screening and sizing machine to dryers (No. 17 on the layout drawings)
- Conveyor (No. 18 on the layout drawings) from dryers to conveyor junction box (No. 15 on the layout drawings)
- Conveyor from junction box (No. 15 on the layout drawings) to the gasifier (No. 11 on the layout drawings)
- Cold and hot flue gas ducting between dryers and boiler #2 (Nos. 19 and 20 on the layout drawings)

3.7 PLANT PERMITTING REQUIREMENTS

For new construction, the following regulatory and permitting requirements are identified:

- A land use permit
- Emission permit for new sources
- Water discharged permit
- Statement of solid and toxic waste generated and method of disposition
- Compliance with noise level at the site boundary

In addition to above, site construction permit, building permit and compliance with local fire code, state boiler code, etc. is required.

Since the Advanced Gasification-Based Power Generation project will be sited in an existing industrial site, some of these activity will require review under the existing permit, while the other activities may require new permitting.

Under the DOE's NEPA Implementing Procedures (10CFR102) a detailed questionnaire and environmental impact will be developed. Following is a brief overview of various permitting and environmental impact work that will be performed by the project.

Land Use Permit

Since the proposed facility will be build on the Boise Paper Solutions' existing DeRidder facility, no new land use permit will be required. Beauregard Parish, LA may require to update the existing permit to reflect changes in the existing facility.

Emission Source Permit

Bark Boiler and Waste Wet Wood Dryer

As stated in Section 3.2, Plant Description and Section 3.3, Major Plant Areas, the gasifier and waste wet wood dryer do not represent a new and continuous emitting source. Hence, no permit will be required. A special filing for flare operation during shutdown and emergency trip of the gasifier may be required.

The existing #2 bark boiler will be modified to accept low Btu syngas as reburn fuel from the gasifier. The design of the new syngas injection and staged combustion/reburning system will improve boiler energy performance and reduce boiler NOx emissions by 30-50% (90-130 tons/year). This does not count NOx reduction that may be associated with the coal-fired purchased electricity that is replaced with biomass and gas-fired self-generated electricity. The project will be required to apply for modification of the boiler, but as it will be an emissions reduction project the Louisiana Department of Environmental Quality (LDEQ) is expected to maintain a positive opinion of the project.

New Gas Turbines/HRSGs

Construction of any new source is subject to New Source Review (NSR). If the source is located in an attainment area, it will also trigger Prevention of Significant Deterioration (PSD) permitting, as required by the Clean Air Act (CAA) for major stationary sources of air pollution in attainment areas.

A major source is any stationary source with the potential to emit more than 100 tons of pollutant per year. The GT35P gas turbines will fall into this category.

The PSD regulations require that new major stationary sources obtain a PSD permit prior to construction to ensure compliance with the applicable NAAQS. To obtain a PSD permit, several steps must be completed:

- Perform a Best Achievable Control Technology (BACT) analysis;
- Conduct an ambient air quality analysis;
- Perform an additional impacts analysis;
- Demonstrate that the project does not adversely impact a Class I area; and
- Undergo adequate public participation.

Bact Requirements

A BACT analysis is done on a pollutant-by-pollutant and unit-by-unit basis considering energy, environmental, and economic impacts to determine the maximum degree of emissions reduction achievable for the proposed source.

Ambient Air Quality Analysis

Since the new sources will be located within the DeRidder Facility, the current ambient air quality data available for this facility will be used.

Additional Impacts Analysis

An impacts analysis is performed to determine the potential effect on soils, vegetation, and visibility in the area surrounding the proposed facility. The direct effect of source emissions and the impacts from general commercial, residential, industrial, and other growth associated with the proposed source may have to be analyzed.

Public Participation

The air permitting process requires public participation, public notice and a public comment period before the reviewing agency takes final action on a PSD application.

Toxics Permitting Requirements

There are no new or additional toxic substances that will be generated from the proposed project. Hence, it is expected that no new toxic permitting will be required.

Liquid Wastes and Discharges

The liquid waste streams expected from the Gasification Plant will include the following:

1. Condensate water from air supply system
2. Condensate water from steam supply in conjunction with air supply
3. Condensate water from nitrogen generation system
4. Floor and equipment drain wastewater

These will not require treatment before discharge.

The gas turbine plant will have HRSG blowdown, which will be collected and send to the existing blowdown collection tank.

The oily water collection system will collect any oil leakage and spill from the gas turbine and send to existing waste collection system at the DeRidder plant.

Solid Wastes

There is one solid waste streams from the Gasification Plant, the bottom ash from the gasifier. The amount of bottom ash is 810 lb/h / 0.1 kg/s

The ash generated during bark and wood waste gasification consists of bed material (spent limestone), some inorganic compounds of fuel ash, unburnt carbon and all impurities fed with feedstock in the gasifier. The carbon content of the ash is below 5%w.

The bottom ash is non-hazardous and will be disposed in the same manner and location as the bark boiler ash.

Noise

During gasification and gas turbine plant operation, there are noise sources from process and plant equipment.

The following noise sources emitting outside the gasification plant:

The flare system noise level will not exceed a value of 75 dB(A) in 1-meter distance from the equipment. The typical noise levels of the flare at ground level are as follows:

- 0 m from flare (building) basis: 72.75 dB(A)
- 10 m from flare (building) basis: 72.57 dB(A)
- 100 m from flare (building) basis: 65.76 dB(A)
- 1000 m from flare (building) basis: 46.72 dB(A)

The following noise sources emitting from the gas turbine power plant:

- Gas Turbine Generator
- Boiler feed pump

The following noise sources emitting from the Balance of Plant Equipment:

- Cooling Tower
- Air Compressor
- Blow-off of air compressor

Venting of depressurized nitrogen from fuel feeding and ash removal systems

- The blow-off noise levels will not exceed a value of 90 dB(A) in 1-meter distance from the equipment.
- The venting noise levels will not exceed a value of 90 dB(A) in 1-meter distance from the equipment.

All the noise sources inside the buildings are in soundproof enclosure, thus the sound power level inside all buildings will not exceed 80 dB(A) at 1-meter distance from the equipment. The noise level will be reduced to 55 dB(A) in special rooms (rooms with medium or low voltage equipment or with electronics and control room).

Greenhouse Gas Emissions

The project would result in a significant reduction in CO_2 emissions resulting from the generation of electricity to meet the mill's power requirements. This may allow the mill to insulate themselves to some extent from future greenhouse gas regulations, which may be promulgated at the state or federal level. A reduction of up to 50,000 tons/yr of carbon dioxide emissions will result from a net reduction in gas usage in the mill as a result of the project. The project will reduce purchased electricity in the mill by 34 MW, resulting in an additional 410,000 tons/yr reduction in CO_2 emissions due to reduction of purchase electricity compared to current operations. The total CO_2 reduction amounts to about 33% of the CO_2 generated to meet the mills purchased electricity requirement. In this analysis, the purchased electricity is assumed to be generated from coal and woodwaste fuel is considered to be CO_2 neutral to the environment.

3.8 CAPITAL COST ESTIMATE

The purpose of this section is to provide capital cost estimates of various systems and components comprising the biomass gasification-based power generation facility.

The costs presented here are preliminary estimates. They do not have the benefit of a detailed estimate due to the pre-design evaluation nature of the study.

3.8.1 Cost Estimating Approach and Basis

Due to the preliminary nature of the study, the capital costs developed here are only indicative of the actual costs that may be expected for implementation of the project. Thus the approach has been to rely on historical data as much as possible with adjustments and extrapolations for differences in size, capacity, and implementation timing. In some instances, budgetary price quotations from equipment manufacturers have also been used. These prices are budgetary because the information given to the manufacturers is preliminary.

The information developed as a part of the pre-design evaluation effort of this study formed the basis for adjustments and extrapolations applied to the historical cost data. In situations where historical data is not available, order-of-magnitude estimates are made based on preliminary equipment specifications generated during the pre-design process.

The format specified by Boise Cascade has been followed to develop the capital cost estimate. The format specifically identifies the various direct and indirect cost parameters that should be used in the estimate. Plant capital cost is comprised of two components:

Capital cost = Direct cost + Indirect cost.

The direct cost of a system or equipment refers to all the costs that can be directly allocated to the system or equipment under consideration. The indirect costs are those that cannot be directly attributed to any specific system or equipment. These costs are applicable to the plant as a whole.

The direct cost, which is also sometimes referred to as construction cost, is determined as follows:

Direct cost = installed equipment cost
+ support facilities applicable to the installed equipment

The installed equipment cost, in turn, is comprised of:

Installed cost = cost of equipment at the manufacturer's plant
+ cost of shipment
+ cost of installation labor
+ cost of materials associated with field installation.

Cost of installation labor is estimated as the product of 'installation labor hours' and 'cost of labor per hour'. Installation labor hour is estimated based the experience of the equipment supplier for similar equipment. Information on hourly cost of labor is provided to the project by Boise Cascade.

Support facilities consist of:

Support facilities = site preparation and auxiliaries
+ buildings and services
+ equipment foundation
+ instrumentation, piping, and electrical.
+ piping
+ electrical.

Costs of support facilities are estimated as percentages of installed cost. Since every piece of equipment is different from the others, these percentages are also different for each equipment. There are no fixed set of percentage values applicable across the board.

The indirect costs are attributable over the entire plant and are discussed below. These costs are generally expressed as fixed percentages of total direct cost of the total plant. Percentage values used here are specified by Boise Cascade and are typical of power projects.

- Consultant's Engineering Services (8%) – services of an architect-engineering firm retained by the owner for design, engineering, procurement, construction management, testing, start-up, and commissioning.
- Owner's Engineering Services (4%) – services performed by the owner particularly related to preliminary assessments, site preparation, permitting, etc.
- Additional Engineering Services (0.5%) – special services sometimes needed for activities, which may have been inadvertently left out of the consultant's engineering services.
- Environmental Engineering (0.5%) – engineering activities related to environmental assessment of the facility construction.
- Capitalized Spares (5%) – generally includes one year's supply of consumable materials and short life items.
- Sales tax is levied at 5% on the direct cost plus all the indirect costs discussed above. Builder's risk insurance is typically 1% of all of the costs above.

The sum of the total direct cost and all of the above indirect costs forms the basis for estimating contingency, which typically ranges from 25% to 30% for order-of-magnitude cost estimates. Boise Cascade specifies 10% contingency based on a detailed cost estimate. Thus for a pre-design evaluation such as the present study, contingency would be much higher. By Nexant/Bechtel experience, even for a cost estimate based on a detailed design, contingency ranges from 25% to 30%. A contingency of 25% is used in the estimate for the present study.

3.8.2 Data Source for the Estimate

The data source for the capital cost estimate is a combination of participants' in-house database and some manufacturers as follows:

Gasification system:	Carbona Corporation
Steam generation system:	Nexant, Inc.
Gas turbine air pre-heating:	Babcock Power Services
Balance-of-plant systems:	Nexant, Inc.
Gas turbine system:	Alstom Power, Inc.
Biomass drying equipment:	AB Torkapparater of Sweden

3.8.3 Estimated Capital Cost

As discussed earlier (Section 2), the plant is envisaged to constructed in 3 steps. Capital cost at the completion of Step 3 represents the total investment required for the proposed plant. The estimated costs are presented in Table 3.8-1 in two sheets. Sheet 1 shows the direct costs, and Sheet 2 shows the indirect costs and the total capital investment. The final project cost at the completion of Step 3 is summarized below:

Direct cost	\$46.0 M
Indirect cost	\$10.3 M
Contingency	\$14.1 M
Total Capital Cost	**\$70.4 M**

3.9 OPERATING AND MAINTENANCE COST ESTIMATE

It is necessary that annual operation and maintenance (O&M) costs of the proposed plant be evaluated both 'before' and 'after' the implementation of the plant to facilitate an economic analysis of the costs and benefits of the plant. The difference between the two ('costs before' minus 'costs after' implementation) represents the cost saving or economic benefit.

The costs involved in operating and maintaining the proposed plant would consist of two components: purchased electricity and fuel; and non-fuel operation and maintenance. Traditionally, the non-fuel operation and maintenance is generally referred to as O&M. The 'before' and 'after' O&M costs are discussed in the following paragraphs. The cost of purchased electricity and fuel is discussed in Section 3.10, Economic Analysis.

Non-fuel O&M. Operation and maintenance of the gasifier and gas turbine plants are the primary contributors to the non-fuel O&M costs. For other systems, these costs are much less and are assumed to be negligible.

O&M costs of the gas turbine plant consist of three components: variable O&M cost, fixed O&M cost, and cost of electricity for supplying compressed gas. Typical industry-average values for variable O&M cost is \$2.2/MWh and that for fixed O&M cost is \$5.1/kW/year. Power requirement for natural gas compression is estimated to be 0.75 MW per gas turbine unit. Cost of electricity is estimated by using the current price of electricity at \$55/MWh.

O&M costs of the gasifier plant consist of three components: materials; labor; and electricity. A typical industry-average value for cost of materials is 2% of capital cost per year. Labor cost is estimated by the total fully-loaded annual salaries (\$0.1 M per operator) of the required number of operators. It is estimated that the plant would require 3 operators to operate the combined gasifier and the gas turbine plants. Considering the number of shifts, operator vacation, and unforeseen events, an industry average multiplier of 4.5 is applied to estimate the total number of operating personnel required. Thus, it is estimated that 14 operating personnel would be required.

Electricity usage is estimated to be 1,000 kW, and the cost is estimated by using the current price of electricity at \$55/MWh.

Table 3.8-1: Estimated Capital Cost - Direct Cost

(Sheet 1 of 2)

DIRECT COST (2003 $)					
Direct Costs:	**Step 1**	**Step 2**	**Step 3**	**Total**	
Gasifier plant equipment					
Fuel Feeding	1,417			1,417	$K
Limestone Feeding	194			194	$K
Air System	740			740	$K
Gasification System	1,770			1,770	$K
Startup Heater	187			187	$K
Gas Cooling	588			588	$K
Flare System	263			263	$K
Ash Discharge	287			287	$K
Cooling Water	32			32	$K
Nitrogen System	600			600	$K
Vendor Engineering (not included in equipment cost)	2,170			2,170	$K
G-T-1 + GT-2 equipment	10,974		10,974	21,948	$K
External air heater #1 (AH-1)	1,547			1,547	$K
HRSG-1 & HRSG-2	1,731		1,731	3,463	$K
Bark boiler #2 modification equipment			2,500	2,500	$K
Dryer equipment	1,317	1,317		2,634	$K
Balance-of-Plant Equipment					
HRSG #1 outlet ducting	70			70	$K
HRSG #2 outlet ducting			88	88	$K
G-T #1 air pre-heating piping	165			165	$K
G-T #2 air pre-heating piping			105	105	$K
G-T NG supply equipment	809		809	1,618	$K
Product gas supply piping	210			210	$K
Bark Conveying and Delivery System	482	321		804	$K
Dryer inlet & exhaust gas ducting	88	81	0	169	$K
Plant cooling water system equipment	101			101	$K
Compressed air system equipment	493			493	$K
Fire protection system equipment	50			50	$K
Plant electrical system equipment	889	86	810	1,786	$K
Total Direct Cost (Construction Cost)	**27,176**	**1,806**	**17,018**	**46,000**	**$K**

Table 3.8-1: Estimated Capital Cost (Sheet 2 of 2)
- Indirect Cost and Total Capital Cost -

TOTAL CAPITAL COST (2003 $)		
Indirect Cost Parameters (Boise Guide):		
Consultant's Engineering Services	8%	% of direct cost
Owner's Engineering Services	4%	% of direct cost
Additional Engineering Services	0.5%	% of direct cost
Environmental Engineering	0%	% of owner's eng. services
Capitalized Spares	3%	% of direct cost
Sales Tax	5%	% of direct cost+Spares+Svcs
Builder's Risk Insurance	1%	% of direct cost+Spares+Svcs+Tax
Contingency	25%	% of (direct + indirect) cost

Indirect Costs:	**Step 1**	**Step 2**	**Step 3**	**Total**	
Consultant's engineering	2,174	144	1,361	**3,680**	$K
Owner's engineering	1,087	72	681	**1,840**	$K
Additional engineering	136	9	85	**230**	$K
Environmental engineering	0	0	0	**0**	$K
Capitalized spares	815	54	511	**1,380**	$K
Sales tax	1,569	104	983	**2,656**	$K
Builder's risk insurance	330	22	206	**558**	$K
Total Indirect cost	**6,111**	**406**	**3,827**	**10,344**	$K
Total Direct + Indirect cost	**33,287**	**2,212**	**20,845**	**56,344**	**$K**
Contingency	8,322	553	5,211	14,086	$K
Total Project Cost	**41,609**	**2,764**	**26,056**	**70,430**	**$K**

3.10 ECONOMIC ANALYSIS

An economic analysis, i.e., estimated costs and benefits of the proposed project to assess the economic viability of the project, is presented in this section. The approach taken to assess the economic viability is to estimate the simple payback period needed to recover the initial investment of the proposed improvements. A simple payback period is computed by dividing the total investment cost by net annual saving realized as a result of operating the proposed plant. Using the guideline suggested by Boise Cascade Corporation, the threshold payback period assumed is 6 years.

Net annual saving is estimated as the difference of total annual savings gained and total annual expenses incurred as a result of operating the proposed plant. A number of parameters are used for the analysis. These are presented in Table 3.10-1.

Table 3.10-1: Input Parameters for Economic Analysis

Input Parameters	Step 1	Step 2	Step 3	
Annual operating hours	8,000	8,000	8,000	Hrs
Biomass feed to gasifier	180.5	180.5	180.5	MMBtu/h
Biomass feed to Boiler #2				MMBtu/h
GT35 power	17	17	34	MW
GT Plant auxiliary power	1.5%	1.5%	1.5%	Percent of gross generation
Natural gas supply to GT #1+ #2 + Dryer	150.0	143.3	266.6	MMBtu/h
Natural Gas saving from PB #1 and Gas Boilers	4	103.4	206.8	MMBtu/h
Natural Gas saving from PB #2	169.2	169.2	169.2	MMBtu/h
Purchase price of biomass	1.77	1.77	1.77	$/MMBtu
Purchase price of electricity	55.00	55.00	55.00	$/MWh
Purchase price of natural gas	4.50	4.50	4.50	$/MMBtu
Escalation of electricity price	3%	3%	3%	
Escalation of natural gas price	3%	3%	3%	
Gas Turbine O&M Cost				
1. Variable	2.2	2.2	2.2	$/MWh
2. Fixed	5.1	5.1	5.1	$/kW/Yr
2. Electricity for Gas Compressor		0.75	1.5	MW
Annual Gasifier System O&M Cost				
1. Materials		2%	2%	Percent of Capital Cost
2. Labor				
A. No. of Operators	14	14	14	
B. Annual Cost per Operator	0.1	0.1	0.1	$M/Op/Yr Includes salaries, additives and overhead
3. Consumption of electricity	1.0	1.0	1.0	MW
PB #2 O&M Cost (Incremental)		0	0	$/Yr Assumed no additional Matl or Labor would be required
Year of cost basis	2,003	2,003	2,003	
Year of construction	2,006	2,006	2,006	

3.10.1 Annual Savings

Total annual savings gained consist of the following:

1. Saving from on-site electricity generation: On-site generation of electricity relieves the plant from buying electricity from grid at a higher cost. Two new gas turbine generating units generate electricity at 17 MW each.

The additional electricity generated on site with the help of the proposed gas turbine generators replaces the equivalent amount of electricity currently purchased from the grid. The cost of replaced electricity represents the cost saving on electricity. The saving is estimated by using the current price of electricity at $55/MWh.

2. Saving from reduced natural gas consumption in Boilers #1 and #2: Auxiliary steam is generated from the exhaust heat of the gas turbine units. This steam supplements steam generation by Boiler #1 (including other gas boilers) and Boiler #2, thus reducing natural gas consumption by these boilers at the rate of 206.8 MMBtu/hr and 169.2 MMBtu/hr, respectively. Saving is estimated by using the current price of natural gas at $4.50/MMBtu.

3.10.2 Annual Expenses

Total annual expenses incurred consist of the following:

1. Cost of natural gas for electricity generation: Natural gas is used to fuel either one (Step 1 and 2) or two (Step 3) gas turbine generator units and one bark dryer (Step1) after implementation of the project. This represents a net expense, as there is no gas turbine generator before implementation. Expense is estimated by using the current price of natural gas at $4.50/MMBtu.
2. Cost of biomass fuel for gasifier: Biomass (wood waste) fuel is used to fuel the gasifier at the rate of 180 MMBtu/hr after implementation of the project. Since there is no biomass consumption in a gasifier before implementation, this represents a net expense. The cost of this biomass consumption is estimated by using the current price of $1.77/MMBtu.
3. Gas turbine O&M expenses: These are discussed in Section 3.9, O&M Cost Estimate.
4. Gasifier O&M expenses: These are discussed in Section 3.9, O&M Cost Estimate.

3.10.3 Payback Period

The economic analysis results including total annual savings and expenses are presented in Table 3.10-2 in two sheets. Sheet 1 shows energy consumption and saving for the current and proposed situations. Sheet 2 presents the cost saving and the resulting payback period.

As discussed in Section 3.8, the total capital investment for the proposed plant is estimated to be $70.4 million. This does not include any subsidy or cost share. From Table 3.10-2, it may be seen that the total annual saving is estimated to be $13.69 million. This results in a simple payback period of 5.1 years.

Savings and payback period was also analyzed assuming an alternative set of natural gas and electricity purchase prices. The alternative prices assumed are $5.00 per million Btu of natural gas and $42.00 per MWh of electricity. The analysis results are presented in Table 3.10-3. As may be seen from the table, the payback period changed very slightly due the alternative set of prices. Simple payback period is increased from 5.1 years to 6.6 years.

3.10.4 Demonstration vs. Commercial Plant Economics

3.10.4.1 Capital Cost

A number of factors are involved in the costs of a demonstration plant and a commercial plant. The capital cost presented in the previous section is for a first-of-a-kind demonstration plant at a specific location, which, in this case, is the Boise Cascade pulp mill at DeRidder in the State of Louisiana. The cost of such a plant tends to be higher than that of a commercial plant because of the development costs associated with a number of key technologies. In this case, the key technologies include gas turbine technology and the external air preheater technology. According to the gas turbine manufacturer (Alstom), gas combustor would require some development effort to operate on externally preheated combustion air. Equipment design and operating parameter envelope need to be established before commercializing the gas turbine system. Similarly, the external air preheater is a novel equipment requiring demonstration of materials compatibility with producer gas, fabricability, heat transfer characteristics, safety characteristics, etc. Safety is particularly critical due to the potential for exothermic reaction and flammability caused by air-gas mixing in case of a leakage. Double wall heat exchanger tubes may be needed to eliminate leakage potential.

An additional factor that would influence the cost is the site condition of the location of the commercial plants. The Boise Cascade DeRidder plant is highly congested requiring spread-out equipment placement and long lengths of high temperature piping, ducting, and conveyors. These have significant impact on the plant cost.

The cost of a commercial plant on the other hand, which could be located at any of the forest product industries facilities, could be potentially lower than that of the first-of-kind plant. These plants would have the benefit of development and operating experience of the first plant. Moreover, the sites may not be as congested as the first plant.

It is difficult to estimate the cost benefit of high temperature piping, ducting, and conveyors without identifying a specific location for a commercial plant. However, a 50 percent reduction may be judged to be appropriate considering the unusual congestion at the DeRidder plant. This would amount to a saving of approximately $1.6M in direct cost. The development cost benefits of gas turbine and external heat exchanger would be approximately $8M and $0.5M in direct costs, for gas turbine and external air heat exchanger, respectively. Thus, the cost of a commercial plant is expected to be as follows:

Direct cost	$35.9 M
Indirect cost	$8.1 M
Contingency	$11.0 M
Total Capital Cost	$55.0 M (2003 $)

3.10.4.2 Annual Expenses and Payback Period

It may be estimated that for a commercial plant the total annual saving would be approximately the same as the first plant at $10-14 million depending on gas and electricity cost assumptions. Thus, with a $55.0 million capital cost, the simple payback period for a commercial plant would be approximately 4 to 6 years.

Table 3.10-2a: Economic Analysis - Energy Consumption and Saving

		Energy Consumption, MWh/Yr or MMBtu/Yr						Energy Saving, MWh/Yr or MMBtu/Yr		
		Current Consumption			Proposed Consumption			Saving = Current - Proposed		
		Step 1	Step 2	Step 3	Step 1	Step 2	Step 3	Step 1	Step 2	Step 3
Electricity										
Annual Electricity Replaced (see Note 1)	MWh	133,960	133,960	267,920	0	0	0	133,960	133,960	267,920
Fuel										
Additional Annual Biomass Consumption in Gasifier and Boiler #2 (see Note 2)	MMBtu	0	0	0	1,683,200	1,485,600	2,141,600	(1,683,200)	(1,485,600)	(2,141,600)
Annual Natural gas consumption for GT#1 & #2 and Dryers (see Note 2)	MMBtu	0	0	0	1,200,000	1,146,400	2,132,800	(1,200,000)	(1,146,400)	(2,132,800)
Annual Natural Gas Consumption Replaced for PB#1 and Gas Boilers (see Note 2)	MMBtu	827,200	827,200	1,654,400	0	0	0	827,200	827,200	1,654,400
Annual Natural Gas Consumption Replaced for PB#2 (see Note 2)	MMBtu	1,353,600	1,353,600	1,353,600	0	0	0	1,353,600	1,353,600	1,353,600

Note 1: Annual Electricity Replaced = Gross GT output * Annual operating hours * (1 - GT plant auxiliary power in % of gross output)
Note 2: Annual Fuel Consumption = Hourly consumption * Annual operating hours

Table 3.10-2b: Economic Analysis - Cost Savings and Payback Period)

		Cost of Energy Consumption, $M/Yr								
		Current Plant			Proposed Plant			Cost Saving, $M/Yr		
Project Implementation Step →		**Step 1**	**Step 2**	**Step 3**	**Step 1**	**Step 2**	**Step 3**	**Step 1**	**Step 2**	**Step 3**
Electricity										
Annual Electricity Replaced by GT	MWh	8.05	8.05	16.10	0.00	0.00	0.00	8.05	8.05	16.10
Fuel										
Additional Annual Biomass Consumption in Gasifier and Boiler #2	MMBtu	0.00	0.00	0.00	2.98	2.63	3.79	(2.98)	(2.63)	(3.79)
Annual Natural gas consumption for GT#1 & #2 and Dryers	MMBtu	0.00	0.00	0.00	5.90	5.64	10.49	(5.90)	(5.64)	(10.49)
Annual Natural gas consumption for PB#1 and Gas Boilers	MMBtu	4.07	4.07	8.14	0.00	0.00	0.00	4.07	4.07	8.14
Annual Natural gas consumption for PB#2	MMBtu	6.66	6.66	6.66	0.00	0.00	0.00	6.66	6.66	6.66
Operation and Maintenance										
Annual Gas Turbine O&M	$M/Yr	0.00	0.00	0.00	0.42	0.42	0.83	(0.42)	(0.42)	(0.83)
Annual Gasifier System O&M Cost	$M/Yr	0.00	0.00	0.00	2.09	2.09	2.09	(2.09)	(2.09)	(2.09)
Annual Power Boiler #2 O&M (incremental) Cost	$M/Yr	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
					Net Annual Saving, $M/Yr.			**7.39**	**8.00**	**13.69**
					Total Plant Capital Cost, $M			**41.6**	**44.4**	**70.4**
					Simple Payback Period, Years			**5.6**	**5.5**	**5.1**

NOTE: Basis for estimating various costs is discussed in Sections 3.8.1 and 3.9.

Table 3.10-3a: Economic Analysis – Alternative - Energy Consumption and Saving

		Energy Consumption, MWh/Yr or MMBtu/Yr						Energy Saving, MWh/Yr or MMBtu/Yr		
		Current Consumption			Proposed Consumption			Saving = Current - Proposed		
		Step 1	Step 2	Step 3	Step 1	Step 2	Step 3	Step 1	Step 2	Step 3
Electricity										
Annual Electricity Replaced (see Note 1)	MWh	133,960	133,960	267,920	0	0	0	133,960	133,960	267,920
Fuel										
Additional Annual Biomass Consumption in Gasifier and Boiler #2 (see Note 2)	MMBtu	0	0	0	1,683,200	1,485,600	2,141,600	(1,683,200)	(1,485,600)	(2,141,600)
Annual Natural gas consumption for GT#1 & #2 and Dryers (see Note 2)	MMBtu	0	0	0	1,200,000	1,146,400	2,132,800	(1,200,000)	(1,146,400)	(2,132,800)
Annual Natural Gas Consumption Replaced for PB#1 and Gas Boilers (see Note 2)	MMBtu	827,200	827,200	1,654,400	0	0	0	827,200	827,200	1,654,400
Annual Natural Gas Consumption Replaced for PB#2 (see Note 2)	MMBtu	1,353,600	1,353,600	1,353,600	0	0	0	1,353,600	1,353,600	1,353,600

Note 1: Annual Electricity Replaced = Gross GT output * Annual operating hours * (1 - GT plant auxiliary power in % of gross output)
Note 2: Annual Fuel Consumption = Hourly consumption * Annual operating hours

Table 3.10-3b: Economic Analysis – Alternative - Cost Savings and Payback Period

		Cost of Energy Consumption, $M/Yr								
		Current Plant			Proposed Plant			Cost Saving, $M/Yr		
Project Implementation Step	→	Step 1	Step 2	Step 3	Step 1	Step 2	Step 3	Step 1	Step 2	Step 3
Electricity										
Annual Electricity Replaced by GT	MWh	6.15	6.15	12.30	0.00	0.00	0.00	6.15	6.15	12.30
Fuel										
Additional Annual Biomass Consumption in Gasifier and Boiler #2	MMBtu	0.00	0.00	0.00	2.98	2.63	3.79	(2.98)	(2.63)	(3.79)
Annual Natural gas consumption for GT#1 & #2 and Dryers	MMBtu	0.00	0.00	0.00	6.56	6.26	11.65	(6.56)	(6.26)	(11.65)
Annual Natural gas consumption for PB#1 and Gas Boilers	MMBtu	4.52	4.52	9.04	0.00	0.00	0.00	4.52	4.52	9.04
Annual Natural gas consumption for PB#2	MMBtu	7.40	7.40	7.40	0.00	0.00	0.00	7.40	7.40	7.40
Operation and Maintenance										
Annual Gas Turbine O&M	$M/Yr	0.00	0.00	0.00	0.34	0.34	0.68	(0.34)	(0.34)	(0.68)
Annual Gasifier System O&M Cost	$M/Yr	0.00	0.00	0.00	1.99	1.74	1.99	(1.99)	(1.74)	(1.99)
Annual Power Boiler #2 O&M (incremental) Cost	$M/Yr	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
			Net Annual Saving, $M/Yr.					**6.20**	**7.10**	**10.62**
			Total Plant Capital Cost, $M					**41.6**	**44.4**	**70.4**
			Simple Payback Period, Years					**6.7**	**6.3**	**6.6**

NOTE: Basis for estimating various costs is discussed in Sections 3.8.1 and 3.9.

4.0 EXPERIMENTAL

Two key technical questions arising from the study were the effect of bark drying on VOC emissions from the plant and material selection for the high-temperature high-pressure air heaters. An experimental evaluation of the VOC emissions from DeRidder bark samples under bark drying conditions was conducted by the Institute of Paper Science and Technology (IPST) at their facilities in Atlanta, GA[3]. The IPST work is discussed in Section 4.1 below.

An experimental evaluation of selected candidate tube materials for AH-1 and AH-3 was conducted by Oak Ridge National Laboratory (ORNL). Test coupons of four tube material were exposed in two different locations in the upper furnace of the DeRidder boiler. The ORNL work is discussed in Section 4.2 below.

4.1 INSTITUTE OF PAPER SCIENCE AND TECHNOLOGY BARK DRYING VOC STUDY

Summary

The VOC and HAP emissions during drying of fresh “bark” were found to be consistent with levels previously reported in the literature for related wood materials. This is not surprising given that the “bark” sample actually contained about 50% wood.

Scope of Work

Fresh “bark” from the Boise #2 Power Boiler in DeRidder, LA was evaluated for Volatile Organic Carbon (VOC) and Hazardous Air Pollutant (HAP) emissions during drying at the Institute of Paper Science and Technology (IPST) in Atlanta, GA. The “bark” sample, shipped to IPST in a five gallon screw capped pail, was stored unopened at 4^0C until analyzed. The “bark” sample, actually a mixture of pine bark and pine wood, contains only about 50% bark with the remainder of the material being wood. The sample material ranges in size from fine particles less than 1mm in diameter to large pieces up to several inches in length. For the lab experiments, the “bark” sample was fractionated into three size categories: fine, medium and large. Total VOC (by EPA Method 25A) and HAP emissions (by Extractive FTIR Spectroscopy) were measured for each of the size fractions during drying to final moisture contents of 30, 20 and 10% by weight.

Apparatus and Experimental Conditions

The hardware setup for measuring the VOC and HAP emissions in real-time during drying of the bark samples consisted of three main components: a heated tube furnace, a Fourier Transform Infrared (FTIR) spectrometer, and a hydrocarbon analyzer. The tube furnace, used to heat and dry the samples to the desired moisture endpoint, was a Thermo Pro, Inc. (Columbus, OH) Model TF12C tube furnace equipped with a hollow alumina sample chamber tube. This tube was heated to 198 ± 2^0C and maintained at this temperature for the duration of the testing. An airflow of 2.55 liters per minute (20^0C and 1 atm), measured through a Gilmont Accucal 220 flow meter, was delivered through the heated tube and across the sample to sweep the VOC and HAP emissions to the detection instruments via heated stainless steel transfers lines. The heated gas exhaust from the tube furnace was routed into the heated 10-meter gas cell inlet of a MIDAC (Irvine, CA) Model I1106 FTIR Spectrometer. This instrument collects and records FTIR spectra of the gas cell contents every 53 seconds and enables the identification and quantification of individual gaseous components in the sample stream. During this testing the following compounds were detected and monitored:

Methanol
Formaldehyde
Acetaldehyde
Pinene

These additional compounds were not detected or monitored.

Propionaldehyde
Acrolein

The outlet from the FTIR gas cell was coupled to the inlet of a J.U.M. Engineering (Germany) Model VE-7 Hydrocarbon Analyzer using a heated stainless steel transfer line. This analyzer, equipped with a flame ionization detector, continuously measures the VOC concentration of the sample stream. An interfaced data system records a VOC concentration data point every 10 seconds.

Sample Preparation and Initial % Moisture Determinations

The "bark" sample container was retrieved from cold storage and opened to remove a large representative grab sample from the center of the sample container. This grab sample was fractionated into three size categories: fine, medium and large. The fine fraction was collected by shaking the grab sample through a No. 6 USA Standard Testing Sieve conforming to ASTM E-11 specifications. This sieve has openings of 3.35mm (0.132in.). This fine fraction was immediately placed and sealed in a clean polyethylene zip-lock bag and weighed. The medium and large fractions remaining in the sieve were quickly fractionated by hand. The large fraction was comprised of bark and wood pieces that had areas greater than approximately 1 in^2. The segregated medium and large fractions were immediately placed and sealed in separate, clean polyethylene bags and weighed. The fine, medium and large fractions constituted 37.35%, 43.89% and 18.76% of the total weight of the grab sample respectively.

The initial percent moistures (wet weight basis) of each of the size fractions were determined by drying three representative aliquots of each fraction to a constant weight at 105^0C. The initial moisture content of each of the size fractions were above the 50% desired target moisture content. The initial moisture content of the fine fraction (51.82%) was somewhat lower than that of the medium fraction (52.66%) which was itself somewhat lower than that of the large fraction (53.30%). A summary of the size fraction and percent moisture data appears in the following table.

	"Bark" Sample Size and Moisture Distribution				
	Wet Weight (g)	Wet Weight (as % of total)	Initial % Moisture (wet basis)	Dry Weight Equivalent (g)	Dry Weight (as % of total)
Fine Fraction	217	37.35	51.82	104.6	37.87
Medium Fraction	255	43.89	52.66	120.7	43.70
Large Fraction	109	18.76	53.30	50.9	18.43
Total	581	100.00		276.2	100.00

Sample Drying Runs

With the instrumentation stabilized, calibrated and operating in a continuous monitoring mode, the fine fraction was analyzed first. Approximately 10.0 grams of sample was weighed to the nearest 0.01 gram into a tubular sample boat constructed from stainless steel wire mesh (100 mesh). The loaded boat was immediately placed into the heated zone of the tube furnace; the airflow was re-established by reconnecting the gas line stopper to the cool inlet end of the furnace tube and a stopwatch was started to monitor elapsed time of the sample in the drying oven. Real-time emission data was monitored on both the FTIR spectrometer data system and the hydrocarbon analyzer data system. For each run, at the end of the drying period, the sample and boat were retrieved from the tube furnace and immediately weighed to the nearest 0.01 gram to determine the final % moisture content of the sample. When sufficient sample runs were collected for the fine fraction at 30%, 20% and 10% final moisture contents, testing of the medium and large fractions were performed. All of the drying tests were conducted over a three-day period.

The following table summarizes the drying times required for each of the three size fractions to each of the three target % moisture endpoints. In addition, the number of valid drying runs that were collected for each size fraction and target % moisture endpoint is presented. The quantitative data from each of these runs were used to arrive at the reported emission values for Total VOC and specific HAPs.

	Drying Time in 198^{0}C Tube Furnace to the Target % Moisture (minutes)			Number of Replicate Drying Runs to the Target % Moisture (number of averaged data points)		
	30%	20%	10%	30%	20%	10%
Fine	6	9	14	9	5	2
Medium	7	11	15	7	4	2
Large	9	13	25	8	5	3

Results

The drying runs for each size fraction to target % moisture endpoints provided a significant volume of data. The FTIR spectroscopy system generated real-time concentration data points for each analyte every 53 seconds and the hydrocarbon analyzer generated VOC concentration data points every 10 seconds. Graphical plots of real-time concentration for Pinene (from the FTIR spectrometer) and Total VOC (from the hydrocarbon analyzer) for each drying run for each size fraction are provided at the end of the report. In addition, graphical plots from the FTIR spectrometer for two long drying runs for each size fraction are presented which show the real-time concentrations of each of the monitored analytes. For the duration of each valid drying run, the raw concentration data were integrated and processed along with airflow, sample weight and percent moisture values to provide final results in micrograms of target analyte emitted per oven dry gram (µg/g o.d.) of sample to each target % moisture endpoint. The results are summarized in the following tables.

	Pinene					
	30% Final Moisture		20% Final Moisture		10% Final Moisture	
	Average concentration µg/g o.d.	standard deviation µg/g o.d.	Average concentration µg/g o.d.	standard deviation µg/g o.d.	Average concentration µg/g o.d.	standard deviation µg/g o.d.
Fine Fraction	142.6	13.1	200.3	21.3	374.4	4.5
Medium Fraction	82.1	9.7	141.2	8.9	182.1	17.8
Large Fraction	52.9	19.5	77.5	14.5	179.8	42.0

	Total VOC as C_3H_8 (Method 25A)					
	30% Final Moisture		20% Final Moisture		10% Final Moisture	
	Average concentration µg/g o.d.	standard deviation µg/g o.d.	Average concentration µg/g o.d.	standard deviation µg/g o.d.	Average concentration µg/g o.d.	standard deviation µg/g o.d.
Fine Fraction	265.7	23.1	357.2	28.0	625.7	0.4
Medium Fraction	168.8	29.0	257.1	22.2	338.6	22.1
Large Fraction	109.6	36.1	148.5	21.5	307.0	34.3

	Methanol					
	30% Final Moisture		20% Final Moisture		10% Final Moisture	
	Average concentration µg/g o.d.	standard deviation µg/g o.d.	Average concentration µg/g o.d.	standard deviation µg/g o.d.	Average concentration µg/g o.d.	standard deviation µg/g o.d.
Fine Fraction	6.3	0.8	8.6	1.4	13.5	2.0
Medium Fraction	2.9	0.2	5.5	0.7	8.1	0.1
Large Fraction	4.6	1.0	7.9	0.6	23.6	4.2

	Formaldehyde					
	30% Final Moisture		20% Final Moisture		10% Final Moisture	
	Average concentration µg/g o.d.	standard deviation µg/g o.d.	Average concentration µg/g o.d.	standard deviation µg/g o.d.	Average concentration µg/g o.d.	standard deviation µg/g o.d.
Fine Fraction	2.3	0.2	3.9	0.3	9.1	0.1
Medium Fraction	4.4	0.7	7.0	1.7	10.5	0.2
Large Fraction	2.9	1.8	3.2	1.0	11.2	5.9

	Acetaldehyde					
	30% Final Moisture		20% Final Moisture		10% Final Moisture	
	Average concentration µg/g o.d.	standard deviation µg/g o.d.	Average concentration µg/g o.d.	standard deviation µg/g o.d.	Average concentration µg/g o.d.	standard deviation µg/g o.d.
Fine Fraction	2.6	1.6	2.0	1.1	1.8	0.8
Medium Fraction	0.2	0.4	0.4	0.5	0.8	0.5
Large Fraction	0.0	0.0	0.0	0.0	0.0	0.0

Total VOC as C_3H_8 (Method 25A)

		Average Concentration of VOC (µg/g o.d.)		
Fraction	Percent of Total	30% Moisture	20% Moisture	10% Moisture
Fine	37.87	265.7	357.2	625.7
Medium	43.70	168.8	257.1	338.6
Large	18.43	109.6	148.5	307.0
Total	100.00	194.6	275.0	441.5

Pinene

		Average Concentration of VOC (µg/g o.d.)		
Fraction	Percent of Total	30% Moisture	20% Moisture	10% Moisture
Fine	37.87	142.6	200.3	374.4
Medium	43.70	82.1	141.2	182.1
Large	18.43	52.9	77.5	179.8
Total	100.00	99.6	151.8	254.5

Methanol

		Average Concentration of VOC (µg/g o.d.)		
Fraction	Percent of Total	30% Moisture	20% Moisture	10% Moisture
Fine	37.87	6.3	8.6	13.5
Medium	43.70	2.9	5.5	8.1
Large	18.43	4.6	7.9	23.6
Total	100.00	4.5	7.1	13.0

Formaldehyde

		Average Concentration of VOC (µg/g o.d.)		
Fraction	Percent of Total	30% Moisture	20% Moisture	10% Moisture
Fine	37.87	2.3	3.9	9.1
Medium	43.70	4.4	7.0	10.5
Large	18.43	2.9	3.2	11.2
Total	100.00	3.3	5.1	10.1

Acetaldehyde

		Average Concentration of VOC (µg/g o.d.)		
Fraction	Percent of Total	30% Moisture	20% Moisture	10% Moisture
Fine	37.87	2.6	2.0	1.8
Medium	43.70	0.2	0.4	0.8
Large	18.43	0.0	0.0	0.0
Total	100.00	1.1	0.9	1.0

Findings

The following findings can be summarized from the data.

1. Total VOC and HAP emissions are influenced by both sample size fraction and final moisture levels.
2. The lower the final moisture level, the higher the Total VOC and Total HAP emissions. For example, Total VOC emissions in the fine size fraction are 1.3 times greater at 20% final moisture than at 30% final moisture, and 2.4 times greater at 10% final moisture than at 30% final moisture.
3. The finer the sample size fraction, the greater the VOC and Pinene emissions for a specific final moisture endpoint. For example, at 30% final moisture content, the Total VOC emissions from the fine size fraction are 1.6 times greater than the emissions from the medium size fraction and 2.4 times greater than the emissions from the large size fraction.
4. Total VOC emission correlates directly with Pinene emission. As Pinene emissions increase, Total VOC emissions increase. Pinene is likely the major contributor to Total VOC.
5. The HAPs detected (in order of highest to lowest concentration) were Methanol, Formaldehyde and Acetaldehyde. Propionaldehyde, Acrolein and Phenol were not detected. In all but the fine size fraction, methanol and formaldehyde accounted for more than 90% of the Total HAP emissions. In the fine size fraction, acetaldehyde contributed to between 7 and 23% of the Total HAP emissions, depending on the final moisture endpoint.
6. As the sample size fraction increases, the data variability increases due to reduced sample homogeneity.
7. The larger the sample size fraction, the longer the drying period required to reach the target % moisture endpoint. For example, the 10 g fine size fraction test specimens took 14 minutes to reach 10% final moisture whereas the 10 g large size fraction specimens took 25 minutes to reach 10% final moisture.
8. The largest Methanol emission measured was from the large size fraction dried to 10% moisture. This probably occurred due to severe over drying along the wood chip edges and along the fibrous bark pieces as a result of the extended drying times required to reduce the bulk % moisture down to 10%.
9. Acetaldehyde was not detected in any of the large size fraction runs. Trace levels were detected in the medium size fraction and somewhat higher concentrations were detected in the fine size fraction.
10. The reported VOC and Pinene emissions are consistent with reported values for related wood materials. Refer to attached "Mechanisms of Terpene Release During Sawdust and Flake Drying" by Sujit Banerjee, 2001.
11. The reported HAP emissions are consistent with reported values for related wood materials[4].

4.2 HIGH-TEMPERATURE HIGH-PRESSURE AIR HEATER MATERIALS TESTING

Summary

Four different candidate tube materials, including SA-213P91, SA-213TP347H, SB-213-800H, and SS-353 were tested in oxidizing and reducing regions of the DeRidder No.2 Bark Boiler.

For reducing atmosphere application only 800 H and 353 MA materials appear to be suitable candidates for AH-1 based on the fact that these materials did not sag during the 3-month period (over 2000 hrs) of in-furnace testing.

In the oxidizing zone of the in-boiler testing all of the materials appeared to have held up during 2 separate series of tests.

Laboratory analysis will be performed on the surviving coupons to determine the final material selections

Scope of Work

Testing of candidate air heater tube materials was conducted in the No.2 Boiler in DeRidder, Louisiana. Tube coupons are located in two sections of the boiler representing conditions expected for the air heater tubes in AH-1 and AH-3. Erosion, corrosion, and fouling experience gained in testing these coupons will allow better material selections to be made for the demonstration plant air heaters. The following tasks comprise the tube material study.

1. Test condition definition and selection of coupon location
2. Selection of tube materials for coupon testing
3. Design of test coupons and modules
4. Development of I&C and data collection system
5. Procurement and characterization of the selected tube materials
6. Fabrication of the test coupons and modules
7. Installation of the test modules at the selected locations
8. Short and long term testing
9. Characterization of coupon samples after exposure
10. Report and recommendations

Design, preparation and installation of the test coupon assemblies were done by the Oak Ridge National Laboratories (ORNL) with direction and support from DeRidder mill personnel and GTI.

Design of Test Coupon Assemblies

The separate air heaters will have their tubes exposed to different gas environments – oxidizing in the stoker and reducing after the gasifier. Anticipated flue gas and syngas temperatures for the air heaters are of the order of 2000-2200ºF. The target exit air temperature is 1400ºF minimum with the eventual goal of air temperatures as high as 1800ºF. Sample tubes were therefore be tested in two different zones in the boiler: near the grate where fuel rich conditions simulate the reducing atmosphere after the gasifier, and at the top of the furnace in front of the existing superheater banks. The tube locations in No. 2 Boiler are shown in Figure 4.2-1. Four different materials, including SA-213P91, SA-213TP347H, SB-213-800H, and SS-353 were selected for testing in both locations. ¾" pipe (1.05" OD) was selected for the test tube coupons based on the availability compressed air (total 200 scfm) for cooling at the mill.

Test Condition Location and Tube Sample Configuration

Based on the project requirements and available data from No. 2 Power bark firing boiler at Boise, two test locations of test tube coupons were identified; one location about 6 ft above the grate considered as a reducing environment, the other location near the top of the boiler in an oxidizing environment as shown in Figure 4.2-1.

The first test panels provided for testing of 4 test samples per panel as shown in Figure 4.2-2 for both top and bottom test locations. ORNL purchased the ¾" pipe and fabricated the bend on the 4 ½ ft long samples. The bends faced opposite the direction of flue gas flow. ORNL assembled the samples in the test panels. The 2nd test panels were designed and fabricated by GTI and provided for testing of 2 test samples per panel for the top location of the boiler.

Measurement Schematics and Data Collection System

To measure metal and air temperature five (5) thermocouples were installed by ONRL on each tube; four (4) of Type K 1/16" diameter with 310 SS sheath attached to the outside surface of the test tube coupon approximately every 14" along the length; and one (1) Type K 1/8" diameter with 310 SS sheath to measure the air temperature on the inside of the tube at the exit of the tube. In the first series of tests the thermocouples were mounted on the topside of the tube and attached with 310 stainless shim stock bands. In the second series of tests, 1/8"x 3/8' x 3/8" Stainless steel machined weld pads were employed to secure the tip of the thermocouple and along the length. In the second series the thermocouples were mounted on the bottom side of the tube facing the flue gas flow.

A control valve, pressure transducer, surge tank and pressure regulator (as shown in Figure 4) allowed automatic control of the compressed air flow to each individual test tube coupon. In some cases air flow control was based on air temperature and in others based on test tube coupon skin temperature. GTI assisted ORNL in selection of controls and instrumentation including pressure regulators, control valves and pressure transducers and piping/electrical configuration. ORNL assembled the piping/electrical for each test tube panel. Preliminary check out of the electrical operation and thermocouple recording/control in a data-logging computer with necessary Labview software was performed at ORNL.

Fabrication of the air heater tube testing assemblies was completed at ORNL including attachment and testing of thermocouples. The assemblies were transported to the DeRidder mill by ORNL personnel and installed through existing access doors on the West side of Level 2 and Level 5 on No. 2 Power Boiler on March 19, 2003. Identification of the tubes and their associated thermocouples and pressure transmitters is given below.

Lower Door

Tube #	Material/Thk, in.	Skin T/C #	Air T/C #	Pressure
1	353 MA / 0.12	1,2,3,4	5	1
2	800 H / 0.22	6,7,8.9	10	2
3	P91 / 0.16	11,12,13,14	15	3
4	347 H / 0.22	16,17,18,19	20	4

Upper Door

Tube #	Material/Thk, in.	Skin T/C #	Air T/C #	Pressure
5	353 MA / 0.12	21,22,23,24	25	5
6	347 H / 0.22	26,27,28,29	30	6
7	P91 / 0.16	31,32,33,34	35	7
8	800 H / 0.22	36,37,38,39	40	8

Installation of Test Tube Coupons in Boise's DeRidder No. 2 Bark Fired Power Boiler.

The first series of test panels were installed on March 19, 2003 in the No. 2 Power Boiler. A cold test was performed with the plant compressed air to check out the instrumentation and measure the air pressures in each sample tube before inserting the test panels in the boiler. Once installed several issues arose. The test panel installed near the top of the boiler between the 1st and 2nd banks of superheater tubes was affected by the operation of the sootblower, which operated 8 to 10 times per day. As a result of sootblower operation, the tubes were bent and some thermocouples were detached. Another issue was that the thermocouples on the bottom panel failed prematurely. Nevertheless preliminary information indicated that the test samples in the bottom location experienced skin temperatures of the order of 1400 to 2000 F as shown in Figure 4.2-3. After 3 months of testing the samples were removed and it was noted that P91 steel sagged in the reducing atmosphere in the lower portion of the furnace.

A 2nd series of new test tube coupons with thermocouples attached on the tubes under a pad welded to the tube was installed on June 23, 2003. For this 2nd series of tests new test panels were made for the top of the boiler and contained two tubes per panel. The new test panels were installed in front of the steam superheater tubes where the soot blower would not affect them. After over two months of operation 12 of 35 thermocouples are still providing data. In this 2nd series of tests, 347H stainless steel sagged in reducing atmosphere. Inspection of the 2nd series of test tube coupons is scheduled for Nov. 3, 2003. At that time it is planned to remove the 2nd set of test tube coupons for laboratory analysis.

Findings to Date

Although the target air temperatures of 1400ºF were not achieved for the short tube samples in either test location because of insufficient length/surface area of the samples, the sample tube skin temperatures in the rich condition approached 1800ºF. Furthermore, the tube samples were exposed to flue gases from bark firing in the boiler for over 2000 hrs under variable conditions. For reducing atmosphere application only 800 H and 353 MA materials appear to be suitable candidates for AH-1 based on the fact that these materials did not sag during the 3-month period (over 2000 hrs) of in-furnace testing. In the oxidizing zone of the in-boiler testing all of the materials appeared to have held up during both series of tests and accordingly further laboratory analysis is required to narrow down the selection for a oxidizing application.

5.0 RESULTS AND DISCUSSION

The plant design study and economic analysis for the proposed Advanced Power Generation System was conducted to evaluate the technical and economic feasibility of the system for the DeRidder mill and other similar mills in the Forest Products Industry. The primary performance goals for the advanced system are to provide increased self-generated power production for the mill, and to increase wastewood utilization while decreasing fossil fuel use. Additional goals are to reduce boiler NOx and CO_2 emissions. The objective of the current study is to determine the technical and economic feasibility of an Advanced Power Generation System capable of meeting these goals so that capital investment decision can be made regarding its implementation at the mill.

5.1 FINDINGS FOR THE DERIDDER MILL

The study revealed that the original system configuration, with all high-pressure air heated in an air heater located in the upper furnace of the bark boiler was not feasible due to the limited physical space and heat available in the furnace. An alternative design was developed utilizing the excess chemical and sensible heat in the gasifier syngas stream to heat additional high-pressure air in a second, external heat exchanger/syngas cooler between the gasifier and the boiler. Each air heater will provide air to one of two ERGTs.

It was determined that the gasifier would require bark to be dried for reliable feeding and so a bark dryer was added to the design using waste heat from the boiler flue gas as the drying medium. A second dryer was then added to dry bark for the stoker boiler in order to make more high-level heat in the furnace available for air heating.

An experimental study was conducted with bark from the DeRidder mill to determine what VOC emissions might be expected from drying bark and exhausting the resulting moist flue gas to the atmosphere through the existing No. 2 Boiler flue gas cleaning system. VOC emissions from the DeRidder were found to be in the expected range for similar materials and the calculated emissions are expected to be within permitted limits.

The mill was found to be somewhat steam-limited during certain periods of the year and the target steam production from No. 2 Boiler was set at 250,000 lb/h for the new system, 125% of the original boiler MCR. HRSGs were added to the ERGT exhausts for about 129,000 lb/h of additional process steam generation including 100,000 lb/h of 250-psig process steam.

It was determined that the internal air heater could be located in the upper furnace just below the arch tip and that, with dried bark as fuel, high pressure air could be heated to the required 1400ºF. It was also determined that the required syngas injection nozzles can be located at appropriate positions on the boiler. A circulation study for the modified boiler was found to be acceptable. It was determined that the boiler would operate at its design GHRR in spite of the increased steam and air production. The required FD fan flows will be similar to current operation at similar steam loads. ID fan capacity will also be similar and will be near its limit with the full system integration in Step 3.

The original gas turbine selected, the Titan 130 by Solar Turbines, proved to be too difficult to modify for externally recuperated operation. A suitable alternative was found in the Alstom GT 35P, an engine that has already been used in a recuperated mode. The GT-35P has the added advantage of generating an additional 3 MW (17MW vs. 14MW for the Titan). It also operates at a lower pressure ratio, so the pressure rating for the air heaters can be lowered. Finally, the

GT 35P can provide and use the total amount of air heated in both air heaters, meaning that both air heaters can stay in service even with one turbine off-line.

It was determined that candidate air heater tube materials are available that should be suitable for operation at heated air discharge temperatures of 1400 ºF and possibly higher. It was further determined that a workable externally recuperated gas turbine cycle can be developed at this air temperature. An experimental study was conducted to test candidate air heater tube materials inside the furnace of No. 2 Boiler. Tube samples were positioned in the boiler to test performance under both oxidizing and reducing conditions. Several of the tested materials have survived for over several thousand hours and exposure of the surviving materials is continuing.

The mill was found to have limited but sufficient space in the vicinity of No. 2 Boiler to install the necessary plant equipment. A gasification plant design was developed for the system that provides sufficient syngas for the system and has an acceptable footprint. Integration of the gasifier and associated equipment into the overall operation of the No. 2 Boiler was determined to be feasible.

The modified plant design was found to meet the primary objectives of the project for increased bark utilization, decreased fossil fuel use, and increased self-generated power in the mill. Bark utilization in all cases for the modified plant is significantly higher (90-130%) than current operation compared to the 50% design goal. For equivalent steam production, the total natural gas usage for the modified plant is in all cases lower than current operation. For Implementation Steps 1 and 2 gas usage is reduced by 45-47% and for Step 3 by 29 %. While the current average steam production from No.2 Boiler is about 213,000 lb/h, the combined steam production from the boiler and HRSGs will be about 315,000 lb/h for Steps 1 and 2 and 379,000 lb/h in Step 3. In Step 3 this steam production increase will be accomplished at a GHRR equal to the original boiler design. Calculated boiler efficiencies for Steps 1 and 2 (steam) and Step 3 (cogeneration-steam plus air) are increased from the original design value of 70% to 74.2, 79.4 and 78.9, respectively due to a combination of improved burnout, operation with lower excess air, and drier fuel. For the fully implemented case in Step 3, the thermal efficiency of fuel to electricity conversion is 79.8% in the cogeneration mode, 5% above the design goal. Finally, self-generated electricity will be increased from the 10.8 MW currently attributable to No.2 Boiler to 29.7MW in Steps 1 and 2 and 46.7MW in Step 3, increases of 175% and 332%, respectively.

Environmental benefits derived from the system include a reduction in NOx emissions from the boiler of about 30 – 50% (90-130 tons/year) as a result of staged combustion (reburning) with syngas in the boiler, improved carbon burnout and operation at lower excess air. This does not count NOx reduction that may be associated with any coal-based purchased electricity that is replaced.

The project would provide a significant reduction in CO_2 emissions from the generation of electricity to meet the mill's power requirements. This may allow the mill to insulate themselves to some extent from future greenhouse gas regulations, which may be promulgated at the state or federal level. A reduction of up to 50,000 tons/yr of carbon dioxide emissions will result from a net reduction in gas usage in the mill as a result of the project. The project will reduce purchased electricity in the mill by 34 MW, resulting in an additional 410,000 tons/yr reduction in CO_2 emissions due to reduction of purchased electricity compared to current operations. The total CO_2 reduction amounts to about 33% of the CO_2 currently generated to meet the mills purchased electricity requirement. In this analysis, the purchased electricity is assumed to be generated from coal and woodwaste fuel is considered to be CO_2 neutral to the environment.

The three-step implementation plan for installation of the modified plant was found to be an acceptable approach to reduce technology risk. The initial implementation step will install the gasification island, the external air heater, one turbine, one HRSG and one bark dryer. No.2 Boiler will not be dependent on these systems so that shakedown and debottlenecking will not jeopardize steam production. In Step 2, the second bark dryer will be installed once performance with the first dryer is judged to be acceptable. In Step 3, the internal air heater will be installed in No. 2 boiler and the second turbine and HRSG will be added only after all other new plant components have been successfully demonstrated.

A Systems Off-Line analysis of the proposed plant indicates that an acceptable means can be developed to deal with the loss of one of the major systems, including one of the ERGTs, which would put its associated air heater at risk for overheating. Since one GT 35P can supply enough air for both air heaters, the remaining turbine can utilize both air heaters with reduced natural gas firing rate until the other turbine is brought back on-line. As designed, the gasification plant and associated air heater and ERGT cannot be run unless No.2 boiler is on-line because the boiler is the only user for the syngas. However, if alternative uses for this fuel gas can be found in the mill, the gasification island and one ERGT producing 17 MW of electricity could be operated independent of the boiler.

Preliminary designs were developed and equipment, materials and operating costs identified for all major plant systems and the Balance of Plant equipment. The resulting economic analysis was developed as a simple payback period computed by dividing the total investment cost by net annual savings realized as a result of plant operations. The total capital investment for the proposed plant is estimated to be $70.4 million. This includes a relatively large 25% contingency and does not include any subsidy or cost share. The total annual saving is estimated to be $13.69 million. This results in a simple payback period of 5.1 years. Savings and payback period was also analyzed assuming an alternative set of natural gas and electricity purchase prices. The alternative prices assumed are $5.00 per million Btu of natural gas and $42.00 per MWh of electricity. The analysis results are presented in Table 3.10-3. As may be seen from the table, the payback period changed very slightly due the alternative set of prices. Simple payback period is increased from 5.1 years to 6.6 years.

The overall construction schedule for the plant is estimated to be twenty-seven months including design, engineering, fabrication, procurement, installation start-up, testing and commissioning. This does not include development time for the turbines, which will have to commence before the rest of the project. The gasification plant equipment is expected to have the longest lead times at 18 months.

5.2 APPLICATION OF THE TECHNOLOGY TO OTHER MILLS

The results of the study for the DeRidder site are expected to be applicable to many paper mills in the U.S. firing wastewood, clarifier solids and other biosolids for steam and electric power production. These waste materials can all be successfully gasified and, with the proposed advanced power system configuration, utilized for electric power generation.

Approximately 3.2 quads ($3.2x10^{15}$ Btu)of energy is consumed annually by the Forest Products Industry with the majority, about 2.7 quads, consumed by the Pulp and Paper Industry.[5] Over half of this energy is derived from recovered biomass sources, including about 30% (0.41

quad/yr) from bark and wood residues typically burned in stoker boilers. The balance of the industry's energy requirements are supplied primarily from fossil fuels and purchased electricity, including about 0.59 quads/yr of natural gas and 0.21 quads/yr of purchased electricity. Annual expenditures by the industry for gas and electricity are about $4.4 billion, representing about 72% of total energy expenditures.

The advanced power system is designed to increase a mill's capacity to self-generate electrical power and steam from biomass while decreasing its dependence on fossil fuels for steam and power production. When applied to a gas-cofired boiler as in the DeRidder case, the technology can eliminate gas usage in the boiler by substituting biomass-derived syngas as reburn fuel. If the boiler is operating below its maximum steaming capacity due to biomass combustion limitations at the grate, the gasification of a portion of the biofuel outside the boiler followed by injection and combustion of the syngas in the boiler can restore its full steaming capacity. This can result in further reduction of gas (or other fossil fuel) usage by the mill as less gas will be used for steam and power generation from package boilers.

The energy and environmental impacts of the advanced power system on the DeRidder mill are discussed in the findings above. The impacts on other mills with woodwaste-fired boilers will depend on the boiler capacity, the extent to which it is cofired with gas or other fossil fuel, how much fossil fuel is used for steam and power generation in other boilers and the price and emissions associated with purchased power. We estimate that there are about 200 woodwaste boilers in the industry consuming about 245 million Btu/hr of bark and other woodwastes each. If the advanced power technology were applied to20 % of this boiler capacity using gas in a similar manner for cofiring and supplemental steam and power generation, the potential increase in self-generated power would be over 8,000 million kWh/yr, or about 13% of the total power purchased by the industry.

Assuming that purchased electricity is generated from coal and considering woodwaste fuel to be CO_2 neutral to the environment, the substitution of self-generated biomass-based power for coal based purchased power in 20% of the industry's woodwaste-fired boiler capacity would reduce CO_2 emissions to the environment by over 10 million tons/yr.

With an average NOx production of about 0.25lb NOx/MMBtu from woodwaste combustion, application of the technology to 20% of the industry's woodwaste boiler capacity has the potential to reduce NOx emissions by over 10,000 tons/yr.

6.0 CONCLUSIONS

The overall conclusion of the study is that while significant engineering challenges are presented by the advanced power system, particularly for the design and fabrication of the internal and external air heaters, these challenges can be met with operationally acceptable and cost effective solutions. The benefits of increased wastewood utilization, reduced fossil fuel usage and increased self-generated electric power can be realized in an economic manner, with a simple payback period on the order of 6 years. Significant environmental benefits will also be realized in the form of reduced emissions of NOx and CO_2.

The results of the study for the DeRidder site are expected to be applicable to many paper mills in the U.S. firing biosolids for steam and electric power production. These waste materials can all be successfully gasified and utilized for electric power generation. The proposed system makes this feasible by significantly reducing the technology risk and cost of typical gasification-based power system using IGCC. IGCC technology hurdles, including the cost, complexity and reliability of key components are eliminated in the proposed system, which does not require high pressure gasifiers and biomass feeding systems to meet the gas turbines pressure ratio or hot gas cleanup systems to meet the turbines stringent inlet gas requirements. The proposed system, using a low pressure gasifier and feeding systems coupled with the use of high-pressure heated air as the working fluid in externally recuperated gas turbines, provides a system much more consistent with typical pulp and paper mill powerhouse operations. The advanced power system offers a near-term solution to the problems of applying advanced gasification-based technology to meet the energy needs and reduce the environmental impact of the U.S. Forest Products Industry.

7.0 REFERENCES

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3. M. Buchanan, Institute of Paper Science and Technology, "Total VOC and HAP Emissions from Fresh Bark," Report prepared for GTI, April 2003

4. Banerjee et al., "HAPs Release from Wood Drying," 2000.

5. MECS 1994-Manufacturing Energy Consumption Survey, Energy Information Administration, 1997

8.0 LIST OF ACRONYMS AND ABBREVIATIONS

ACFM	Actual Cubic Feet per Minute
AH	Air Heater
ASME	American Society of Mechanical Engineers
ASTM	American Society for Testing Materials
BACT	Best Achievable Control Technology
BART	Best Available Retrofit Technology
BFD	Block Flow Diagram
BOP	Balance-of-Plant
CAA	Clean Air Act
CEM	Continuous Emissions Monitor
CFR	Code of Federal Regulation
CGT	Combustion Gas Turbine
CLECO	Central Louisiana Electric Company
DNB	Departure from Nucleate Boiling
DOE	U.S. Department of Energy
EGT	Exhaust Gas Temperature
EPA	Environmental Protection Agency
FD	Forced Draft
FGR	Flue Gas Recirculation
FPI	Forest Products Industry
FTIR	Fourier Transform Infrared
GHRR	Grate Heat Release Rate
GT	Gas Turbine
HAP	Hazardous Air Pollutant
HCV	High Calorific Value
HRSG	Heat Recovery Steam Generator
H&MB	Heat & Mass Balance
HMZ	Horizontal Mixing Zone
HTSH	High Temperature Superheater
ID	Inside Diameter
	Induced Draft
IGV	Inlet Vane Guide
LCV	Low Calorific Value
MCR	Maximum Continuous Rating
MJ/hr	Mega (Million) Joules per hour
MMBtu/hr	Million Btu per hour
MSW	Municipal Solid Waste
NEPA	National Environmental Policy Act
NSR	New Source Review
OD	Outside Diameter
OFA	Overfire Air
PM	Particulate Matter
PSD	Prevention of Significant Deterioration
SCM	Standard Cubic Meter
SH	Superheater

ACRONYMS AND ABBREVIATIONS - Continued

UGA	Undergrate Air
VOC	Volatile Organic Carbon, Volatile Organic Compounds
µg/g o.d.	micrograms per gram on an oven dry weight basis
ug/g o.d.	micrograms per gram on an oven dry weight basis
ppmv	parts-per-million-volume
in.	inches
in^2	square inches
mm	millimeter
g	grams

9.0 FIGURES

The following figures are presented in this section:

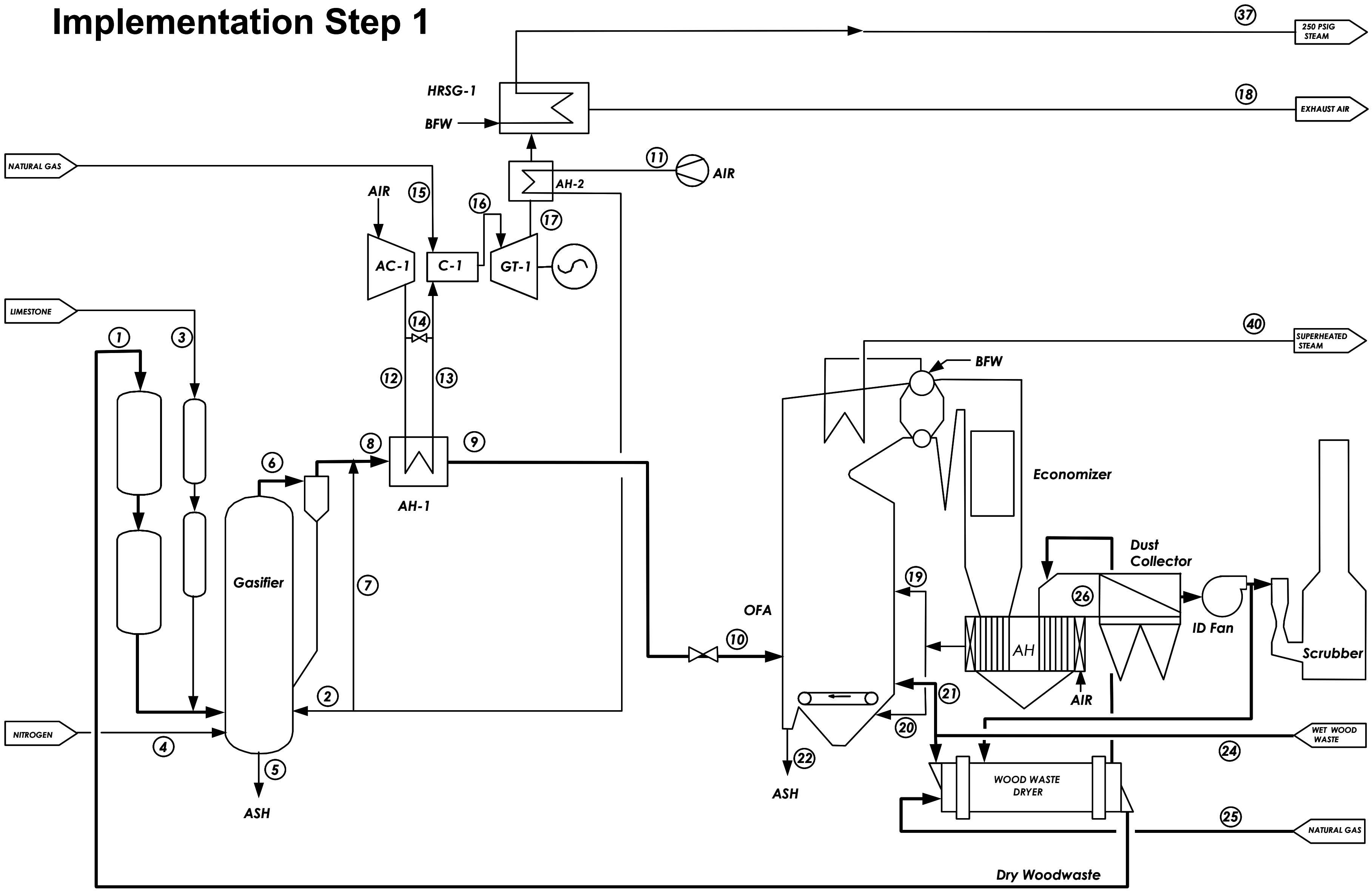


Figure 3.1.1-1 Step 1 Process Flow Diagram

Implementation Step 2

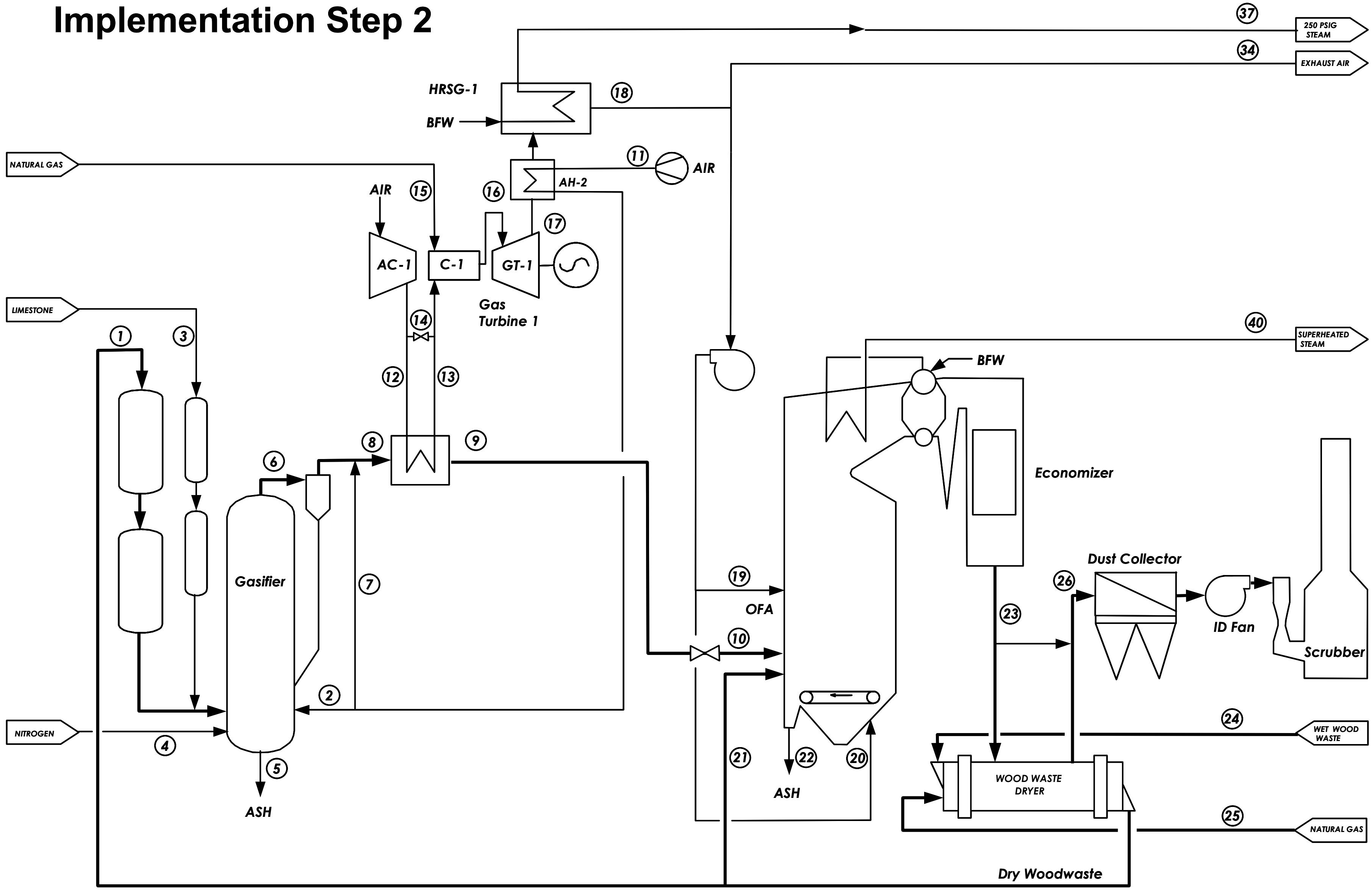


Figure 3.1.1-2 Step 2 Process Flow Diagram

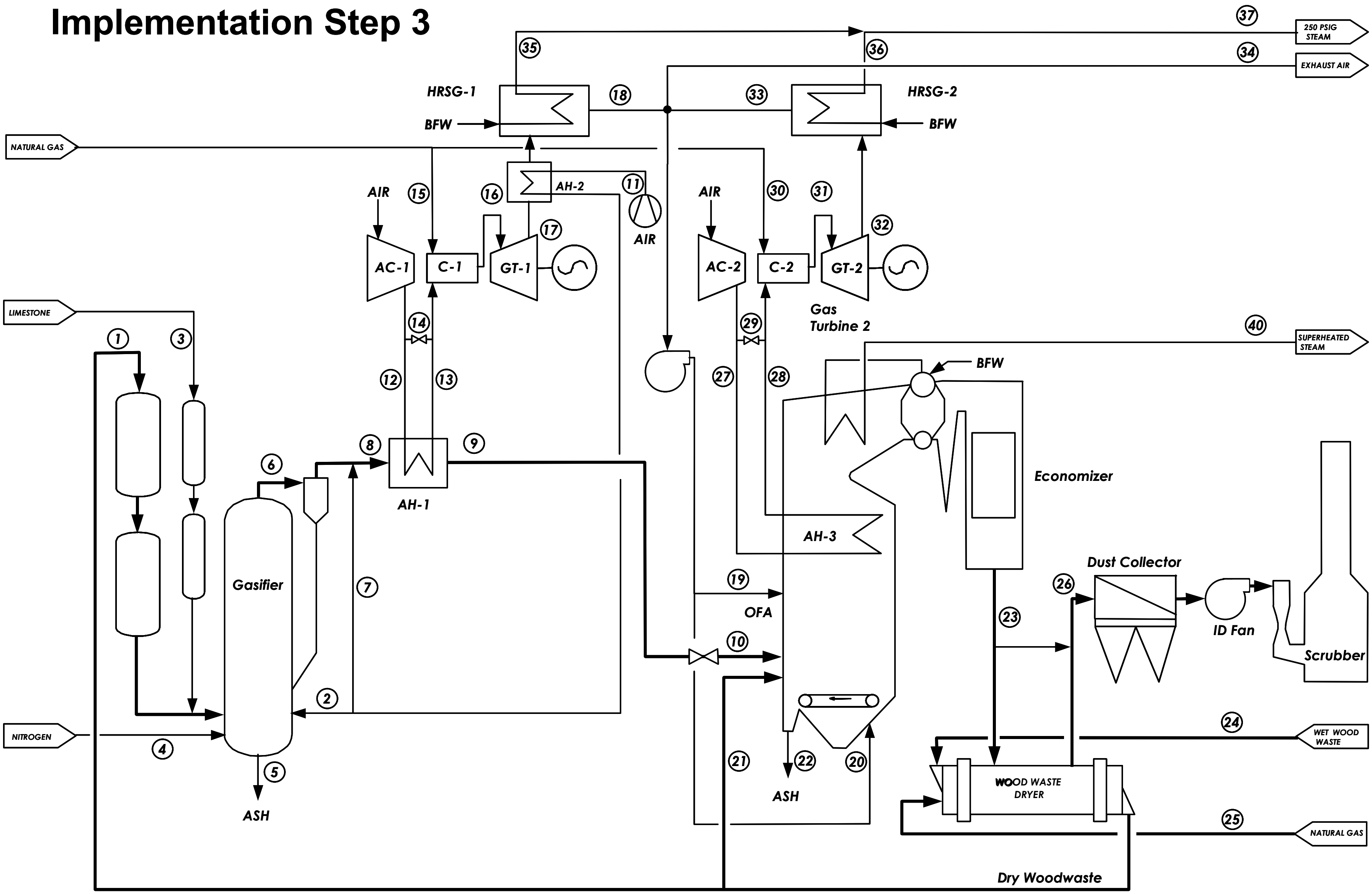


Figure 3.1.1-3 Step 3 and Overall Plant Process Flow Diagram

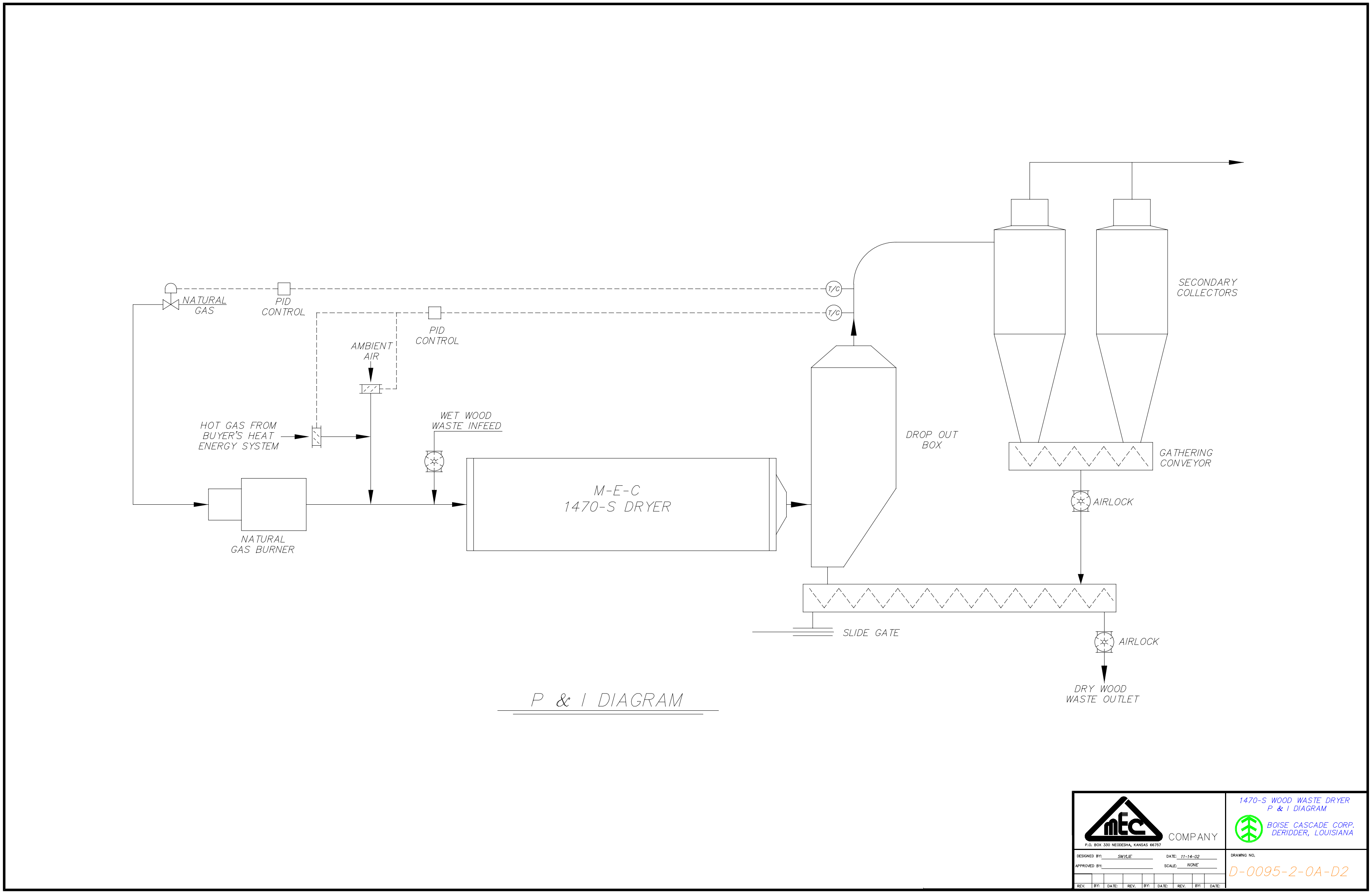


Figure 3.3.1-1: Biomass Dryer Process Flow Diagram

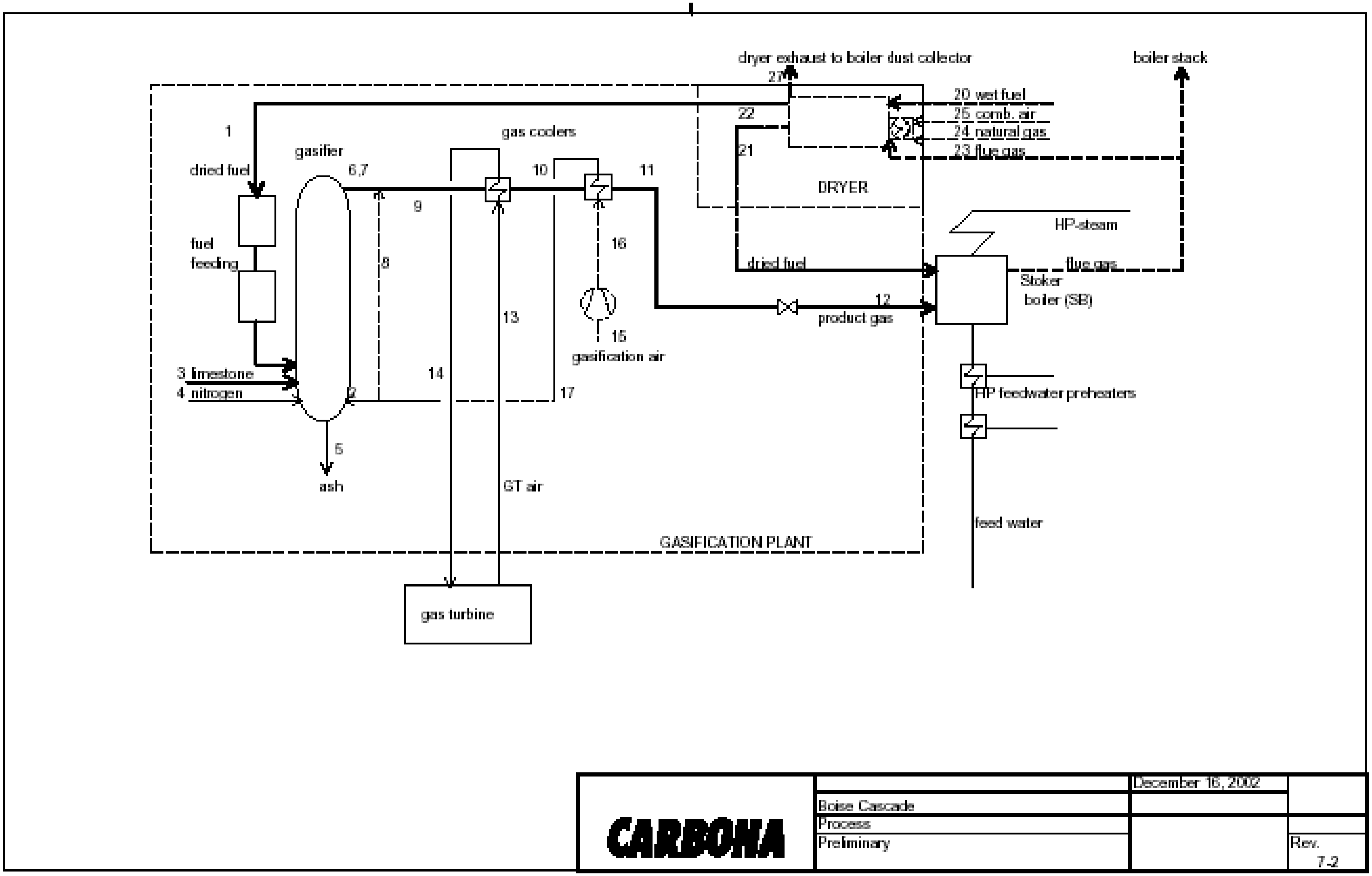


Figure 3.3.2-1: Gasification Process Flow Diagram - Step 3

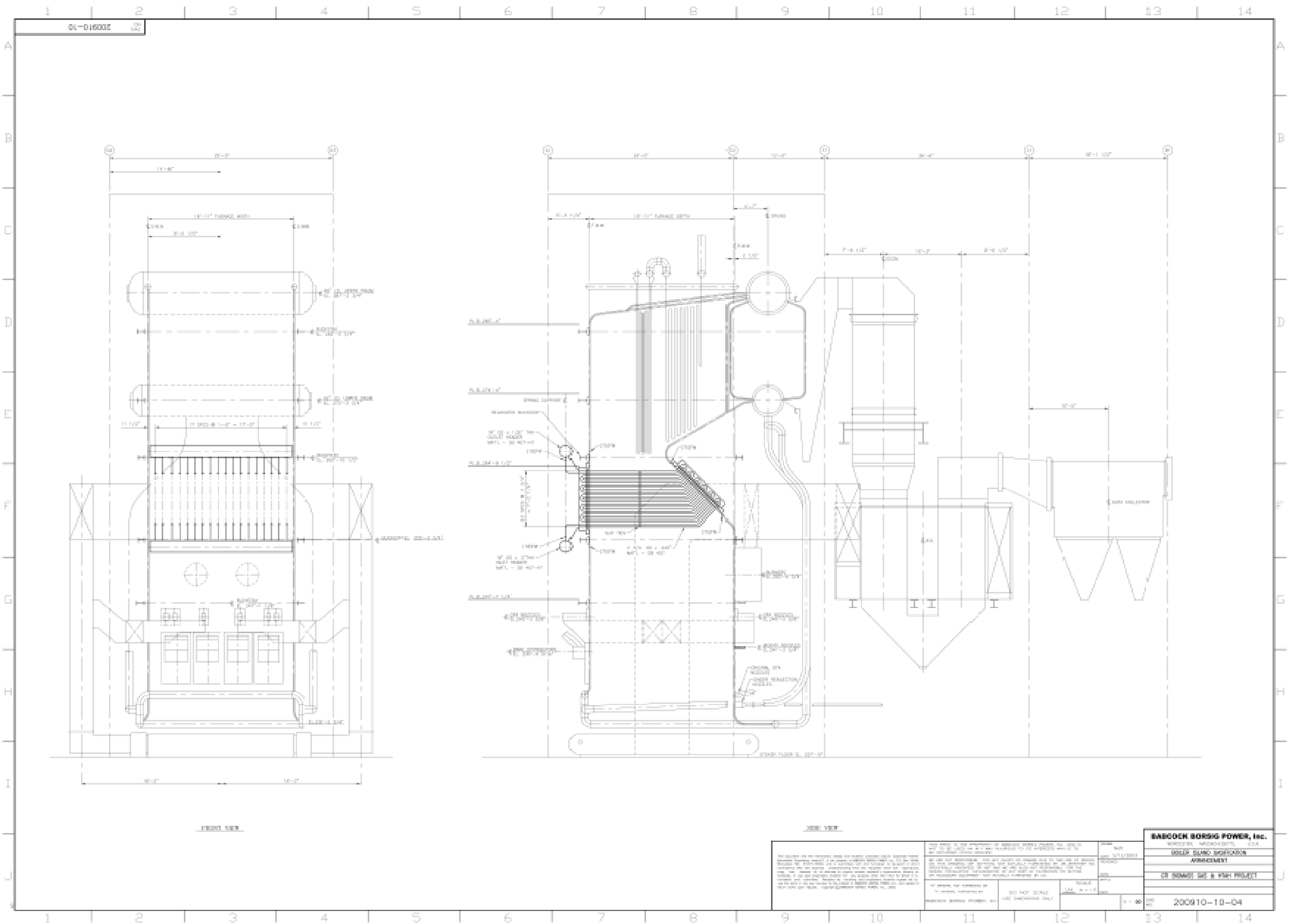


Figure 3.3.3-1 Internal High Temperature High Pressure Air Heater

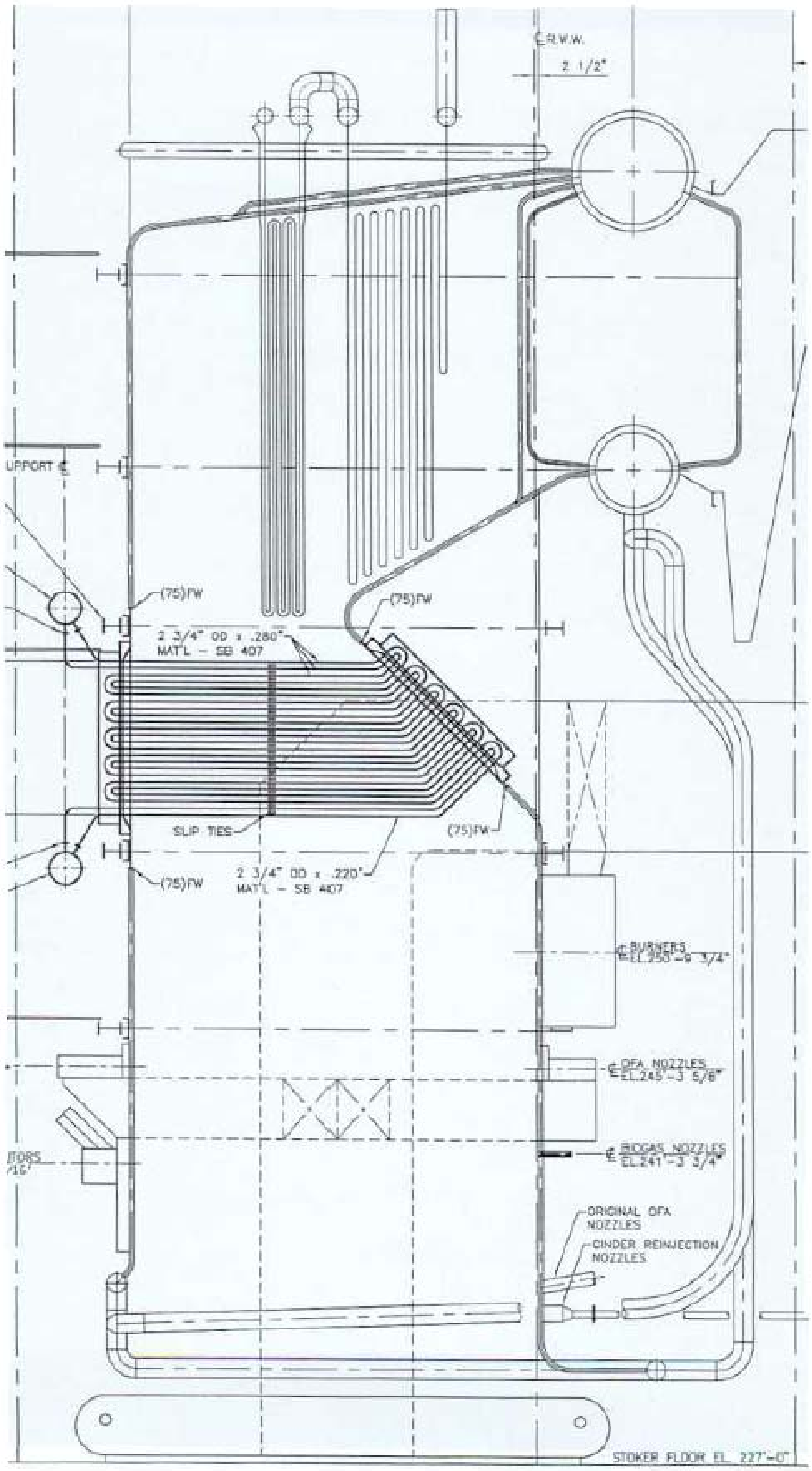


Figure 3.3.3-2 Internal HTHP Air Heater – Selected View 1

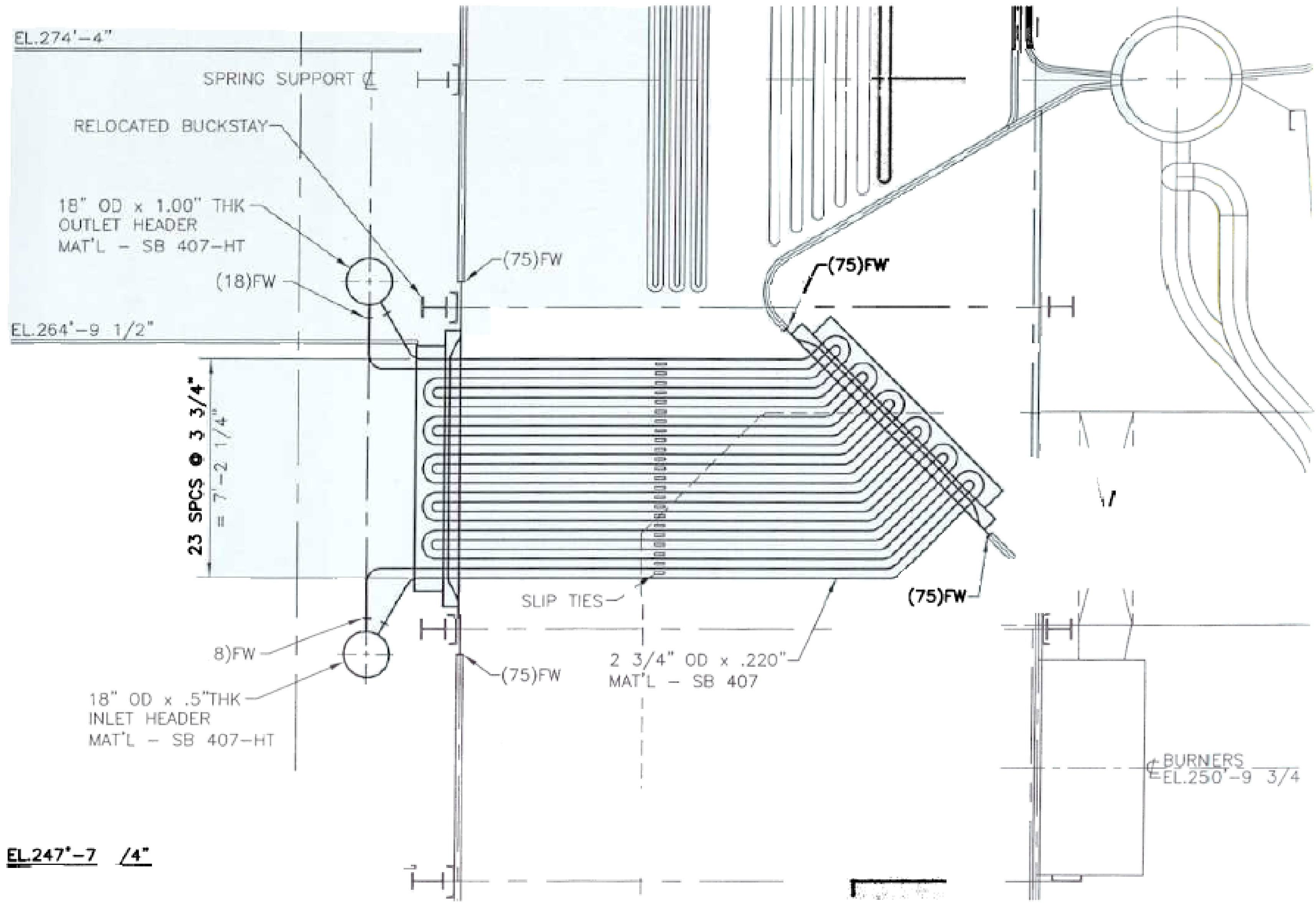


Figure 3.3.3-3 Internal HTHP Air Heater – Selected View 2

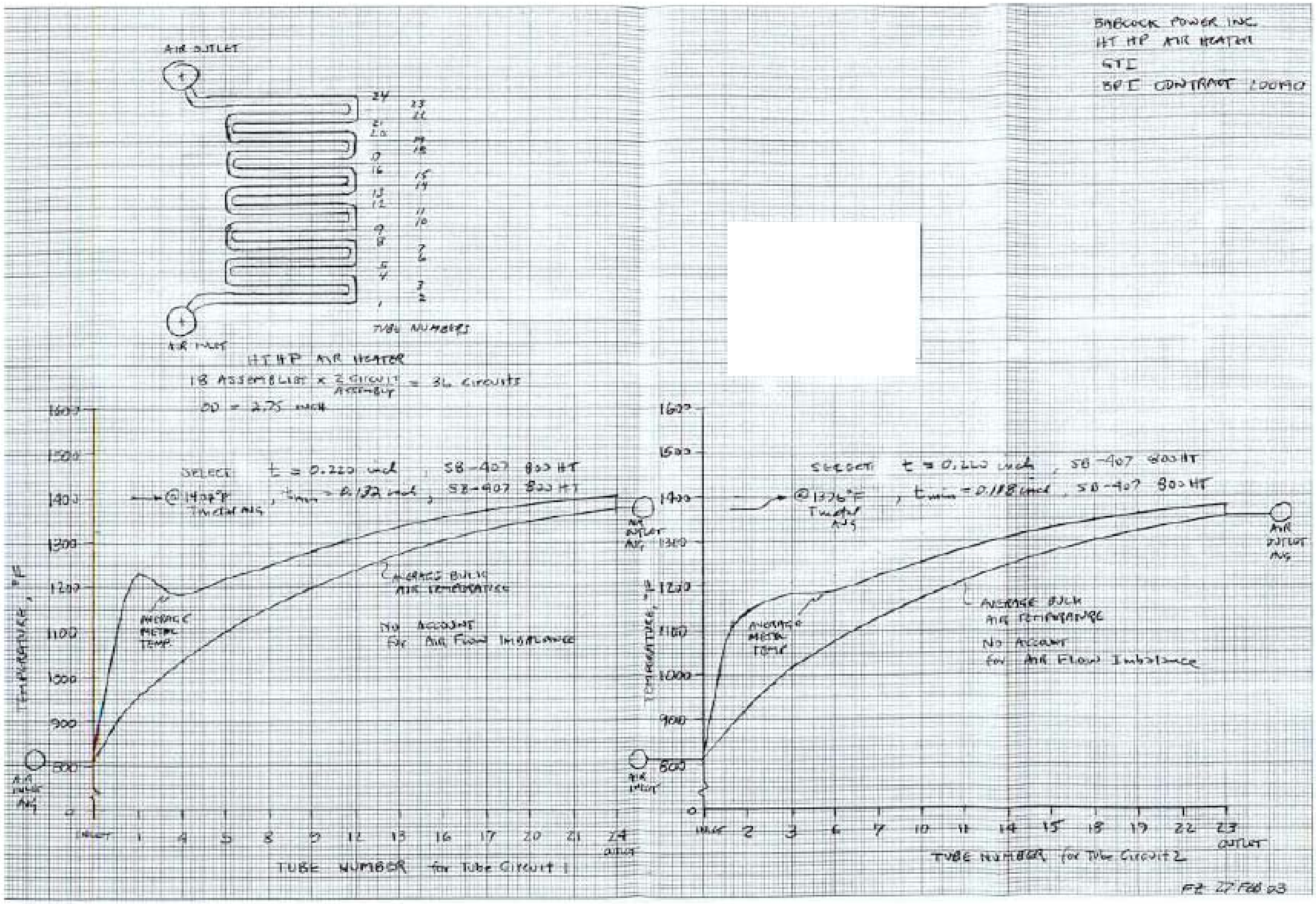


Figure 3.3.3-4 Metal Study for Internal HTHP Air Heater – No Air Imbalance

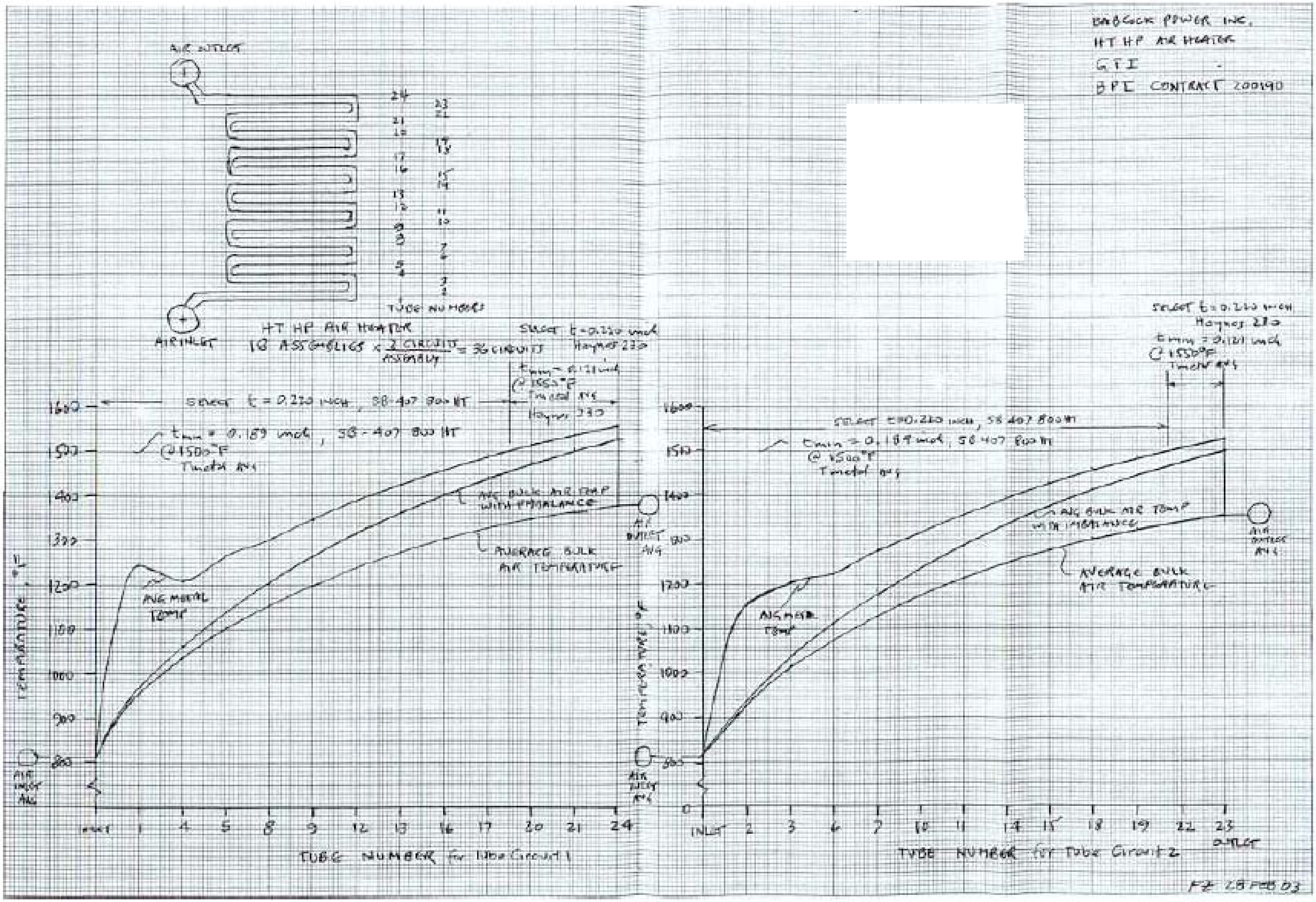


Figure 3.3.3-5 Metal Study for Internal HTHP Air Heater – With Air Imbalance

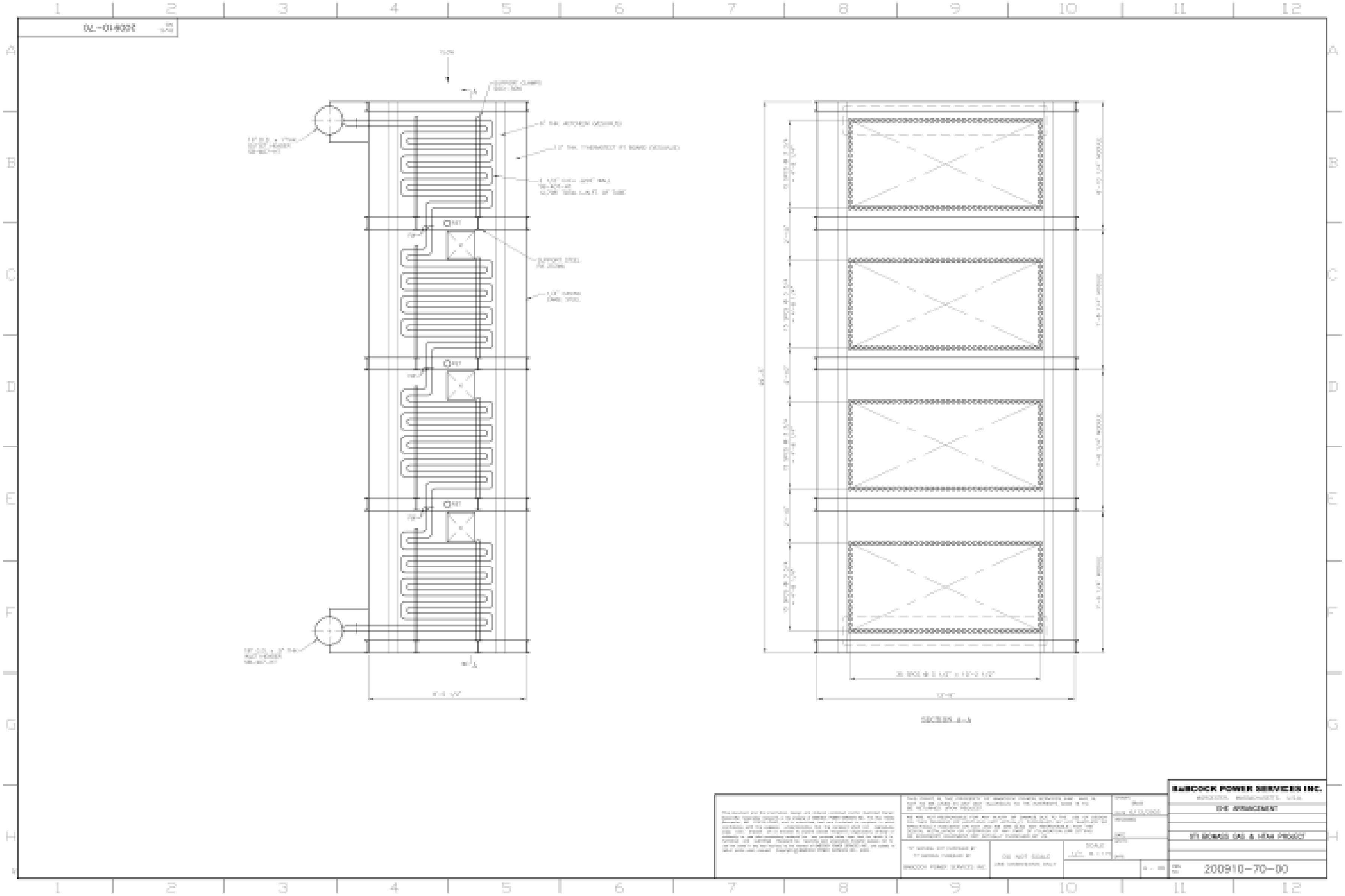
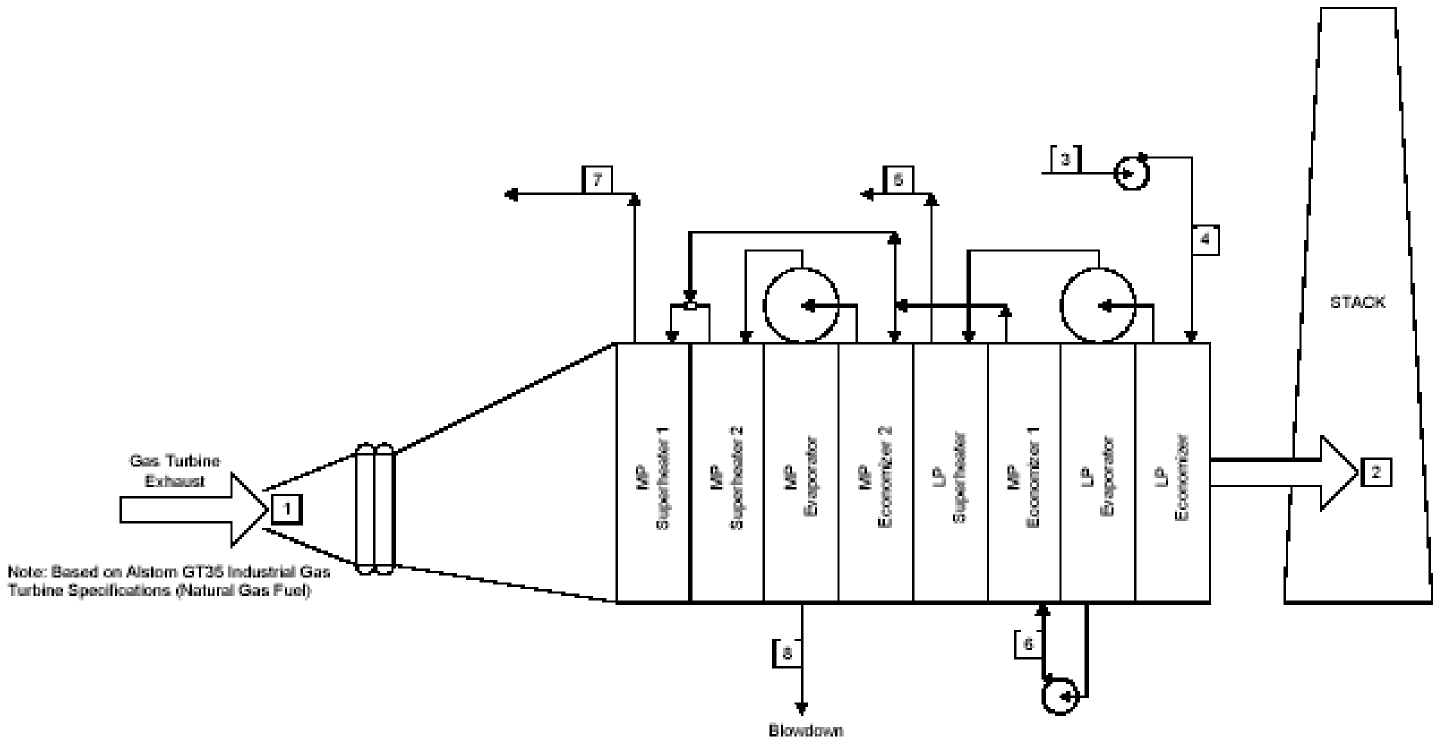


Figure 3.3.4-1 External High Temperature High Pressure Air Heater-Design 1

Single Train Dual Pressure HRSG



Estimated HRSG Performance (not guaranteed)

Stream No.	Units	1	2	3	4	5	6	7	8
Mass Flow	kg/hr	332,280	332,280	29,463	29,463	5,830	23,633	23,396	236
Temperature	ºC	375.0	129.5	30.0	31.1	176.7	146.9	369.4	187.9
Pressure	bar	1.02	1.02	1.00	4.55	4.17	18.47	18.27	18.27
Enthalpy	kcal/kg	93.2	28.8	30.0	31.2	671.0	148.0	760.7	190.7

Mexant, Inc.
HRSG Performance
GTI - Boise DeRidder
Rev A BPatel
Date: 06/12/03

Figure 3.3.6-1 HRSG Arrangement and Performance

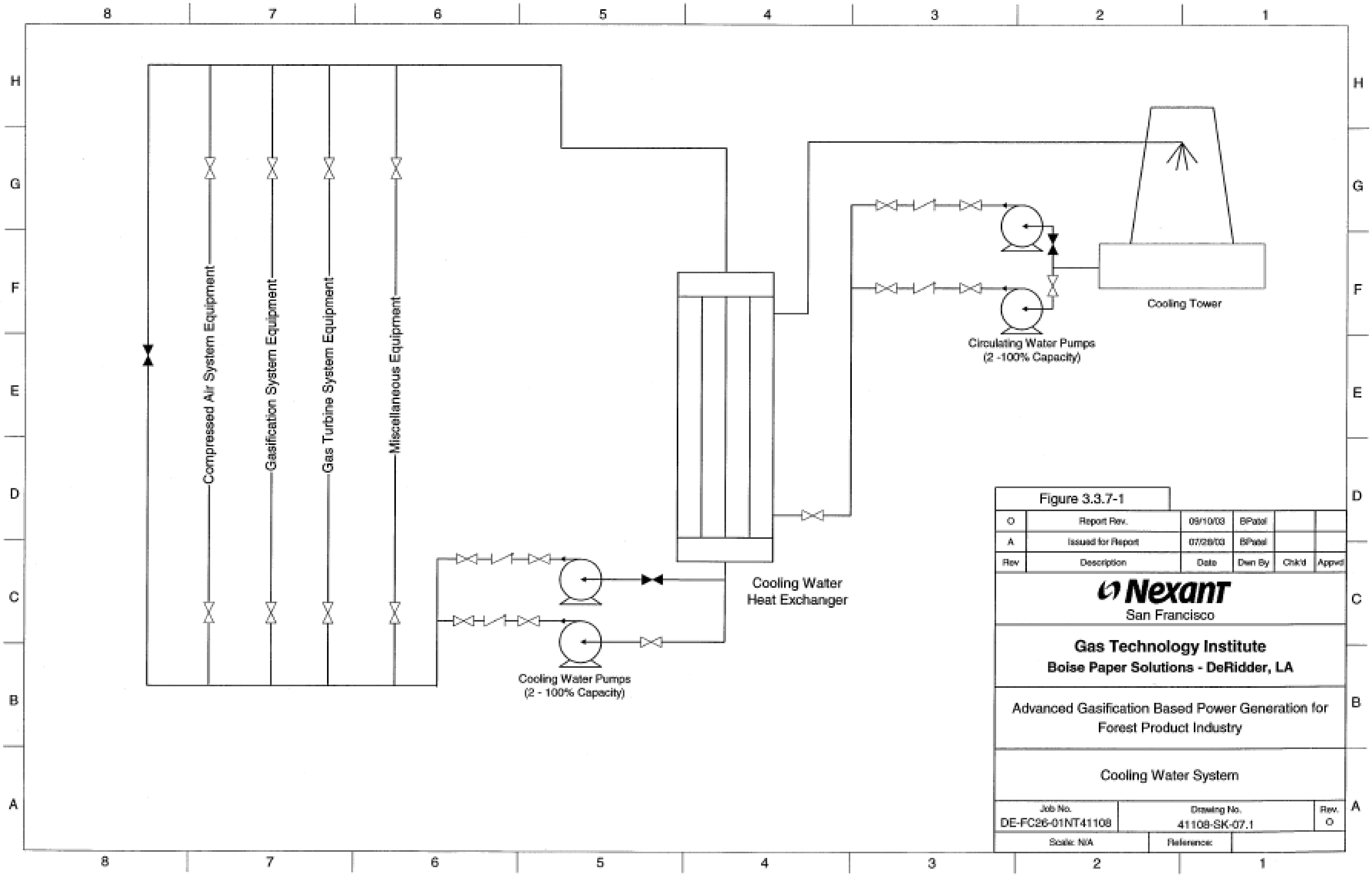


Figure 3.3.7-1 Cooling Water System

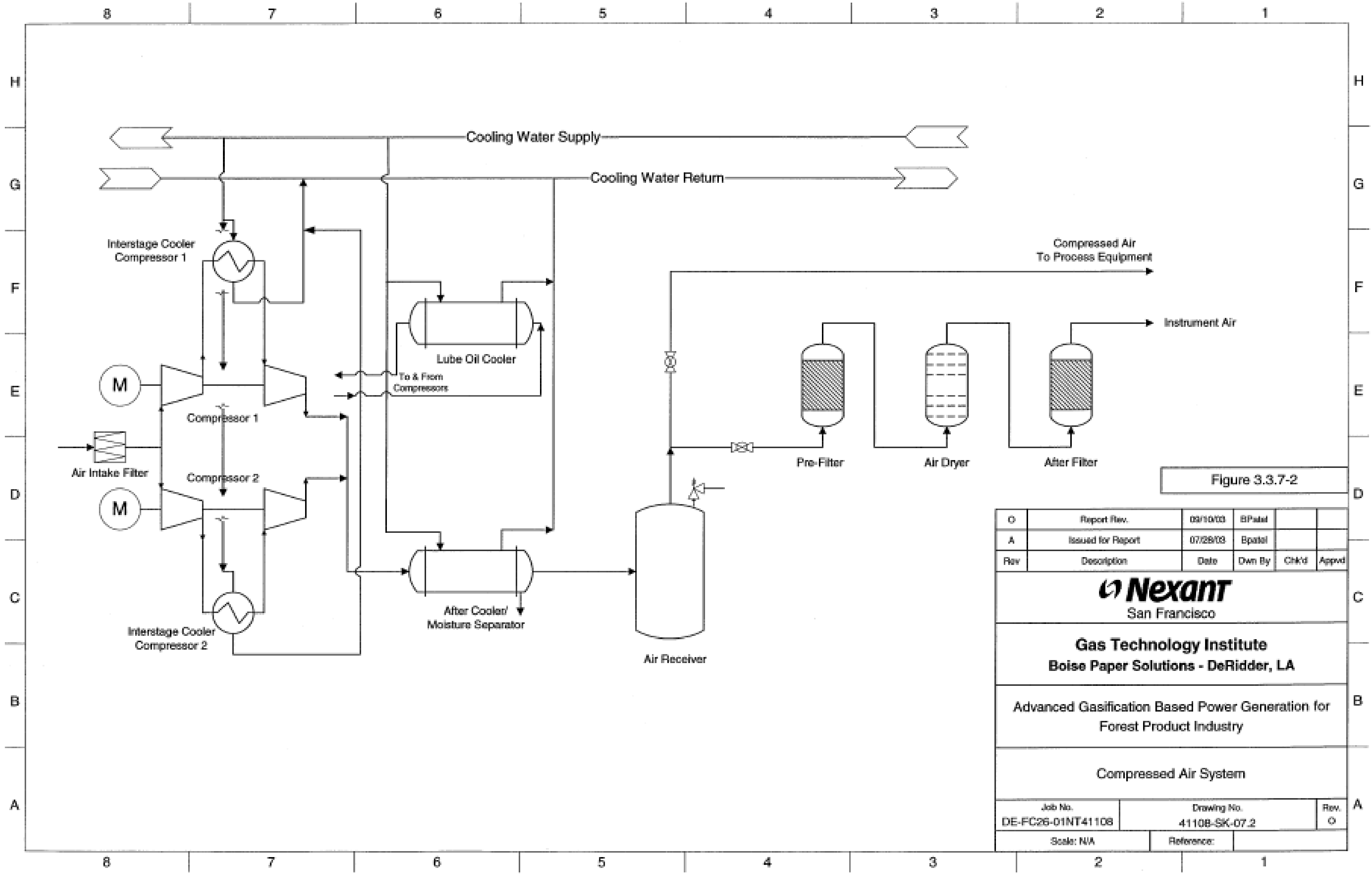


Figure 3.3.7-2 Compressed Air System

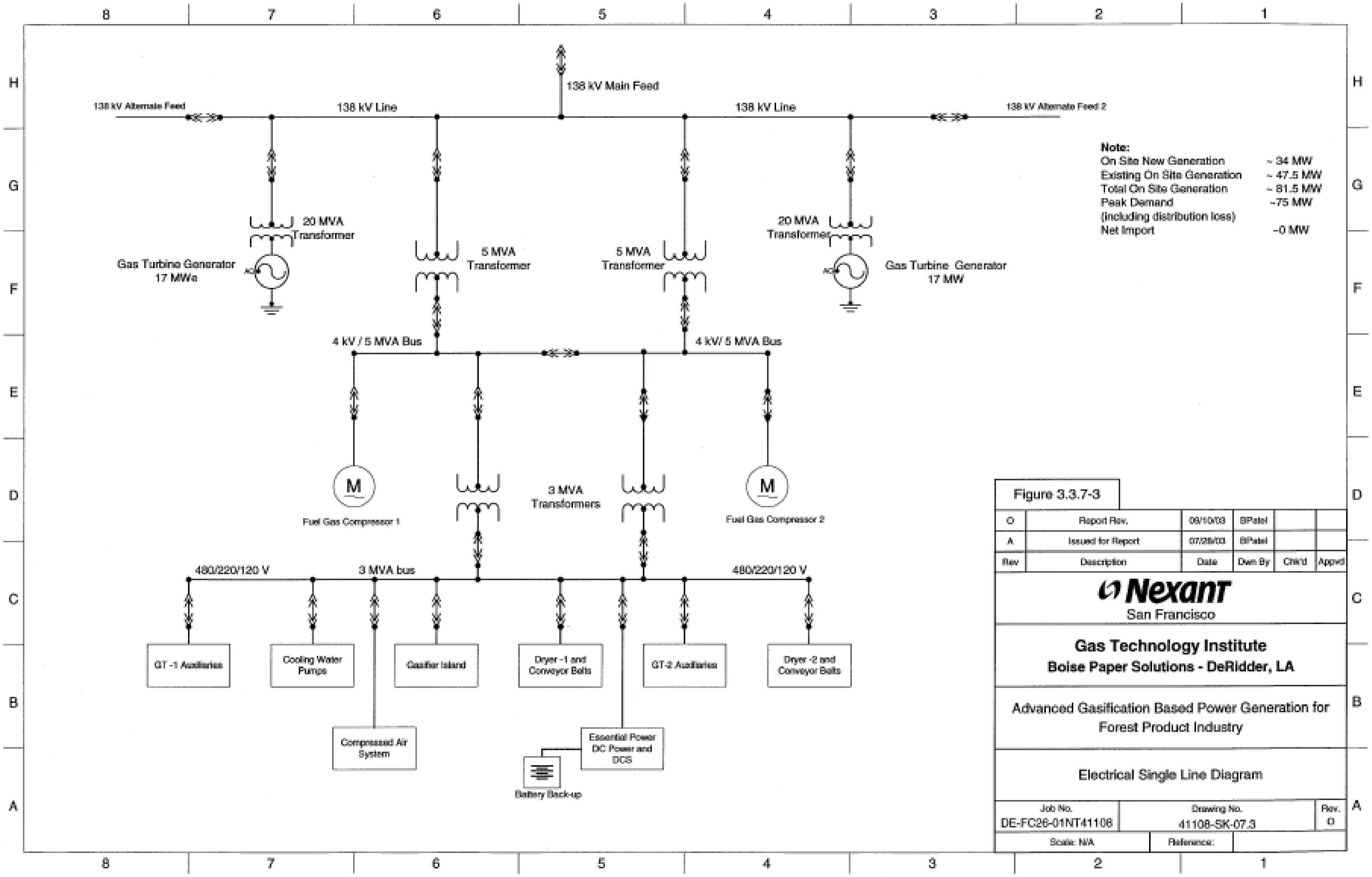


Figure 3.3.7-3 Electrical Single Line Diagram

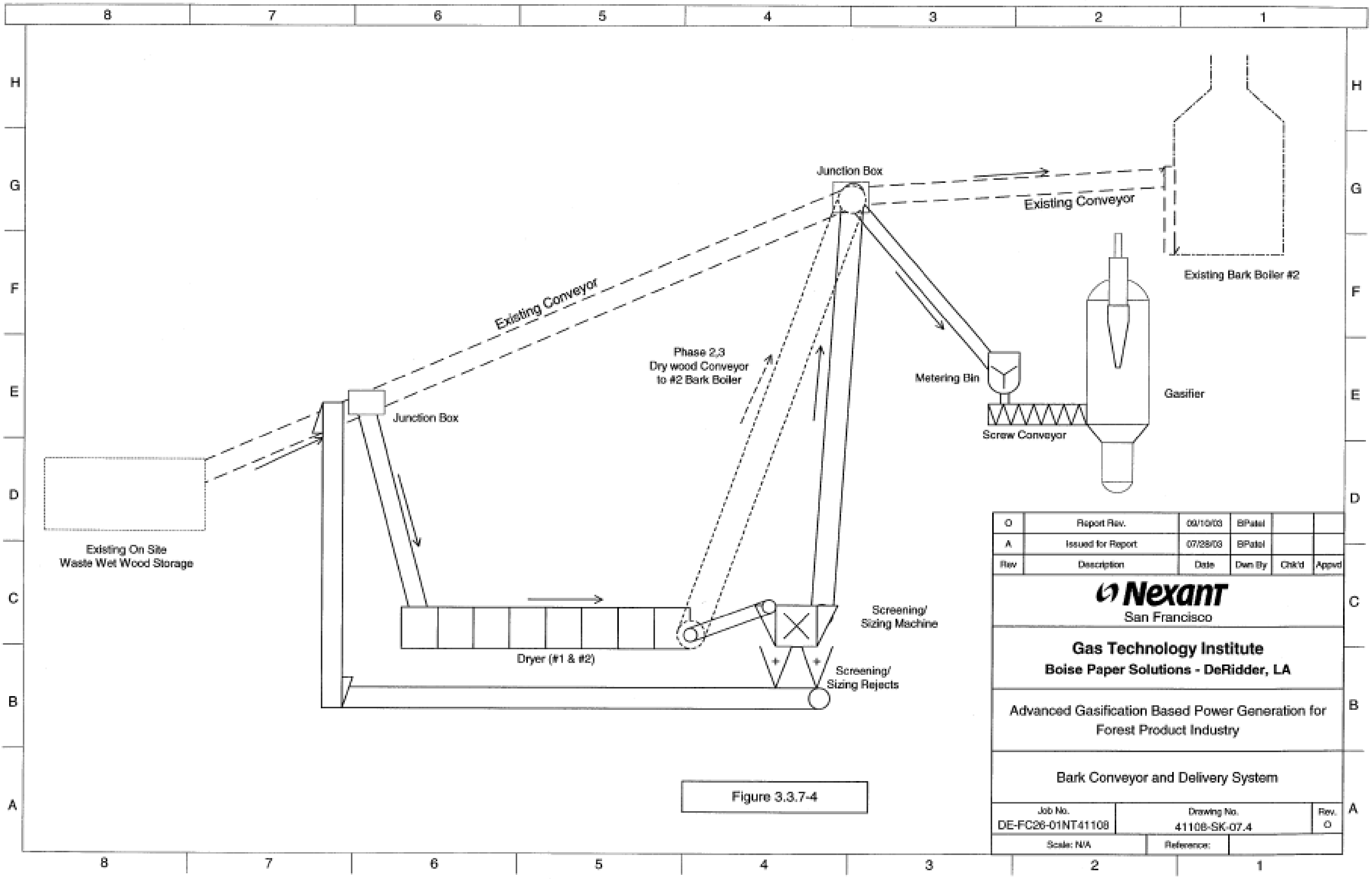


Figure 3.3.7-4 Bark Conveying and Delivery System

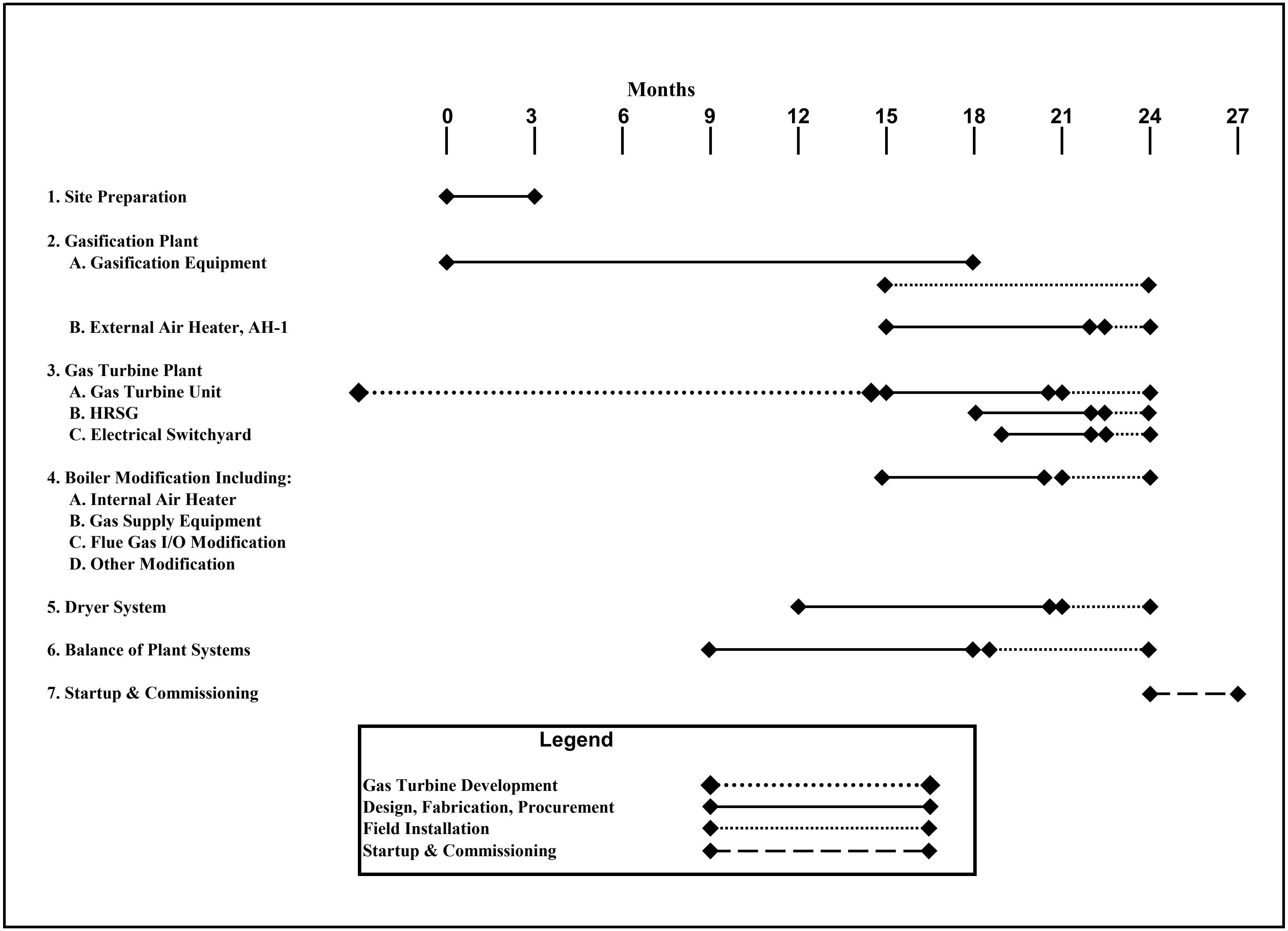


Figure 3.3.8-1 Plant Construction Schedule

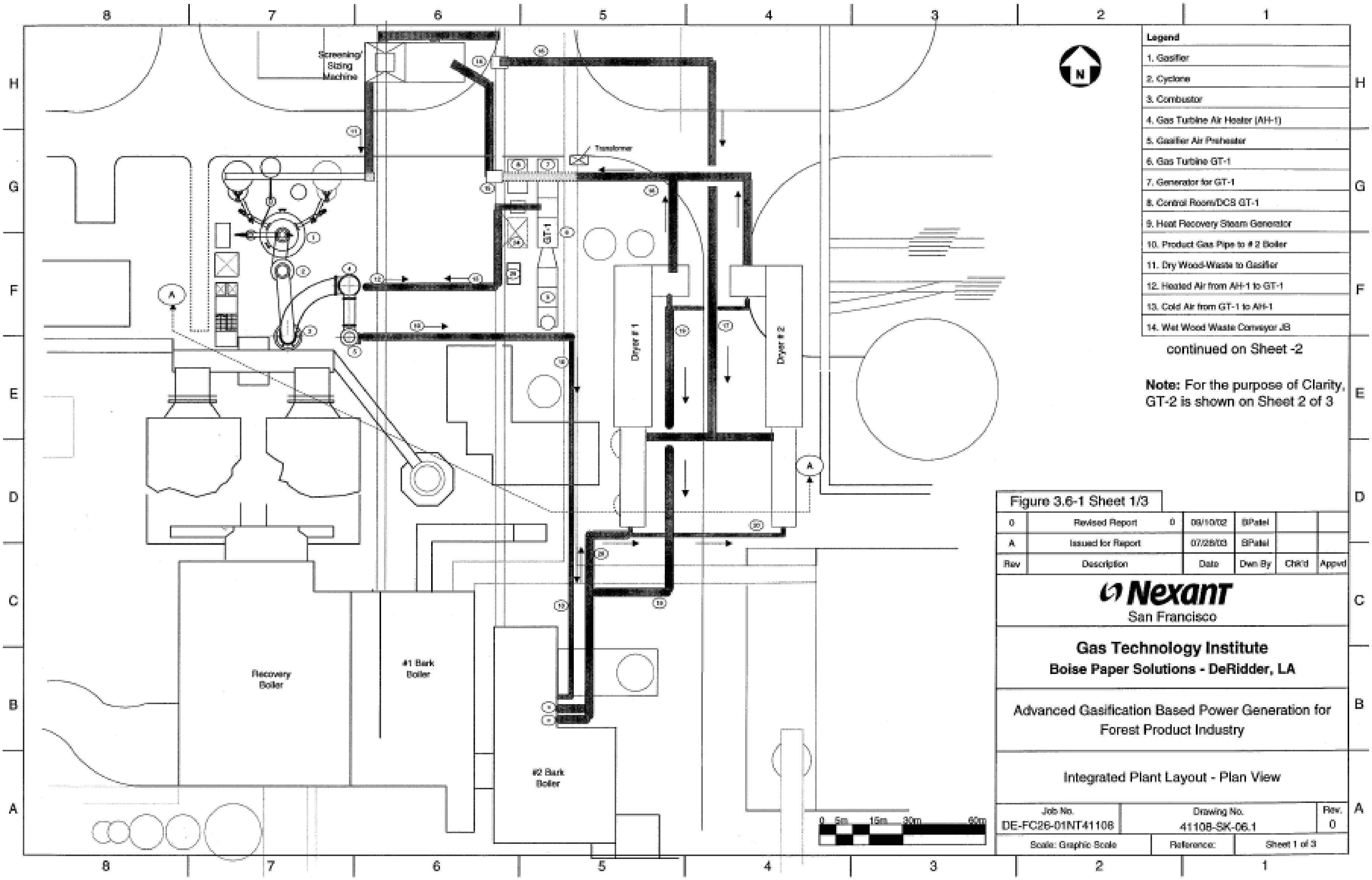


Figure 3.6-1 Integrated Plant Layout - Sheet 1

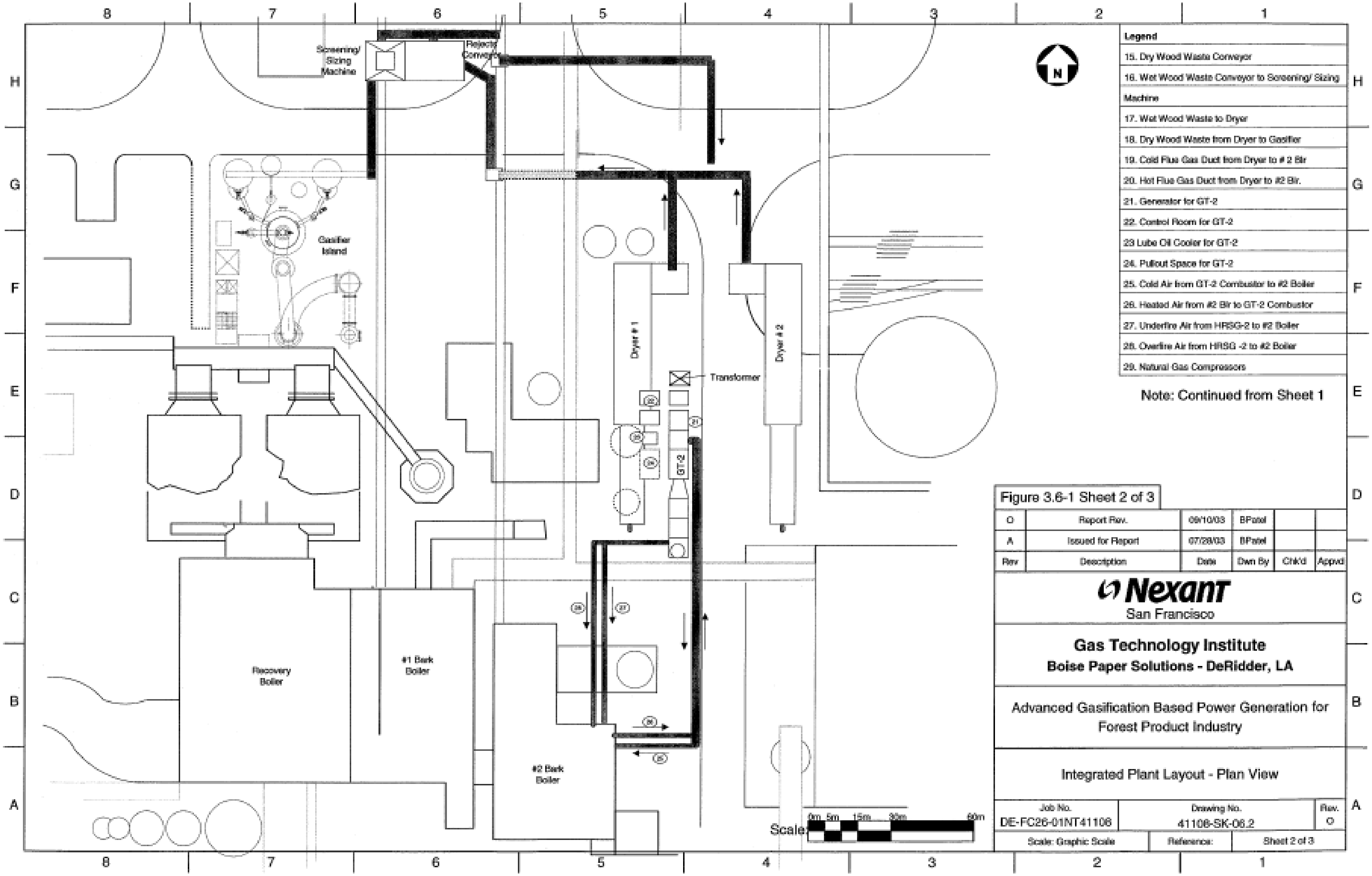


Figure 3.6-1 Integrated Plant Layout - Sheet 2

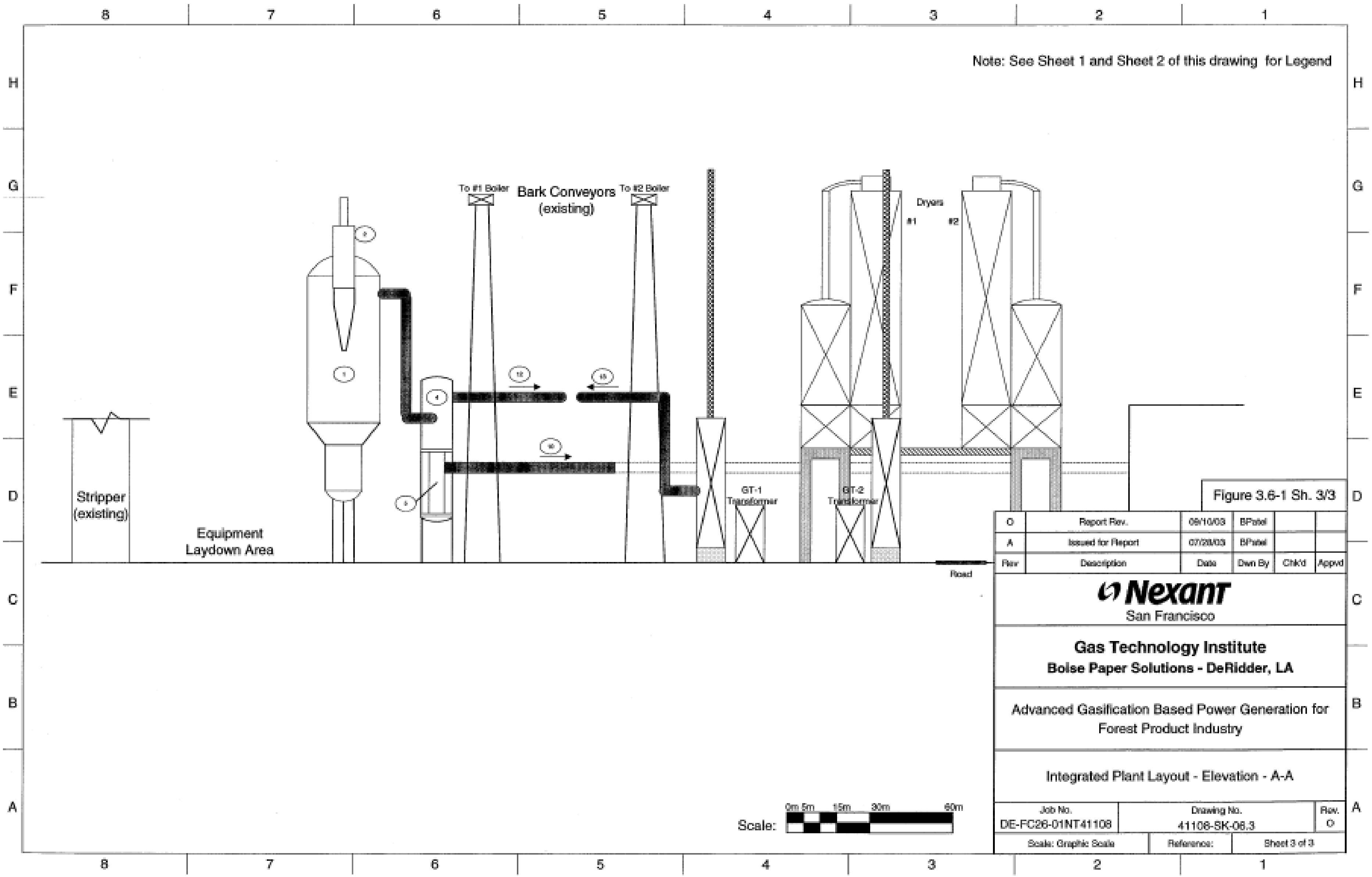


Figure 3.6-1 Integrated Plant Layout - Sheet 3

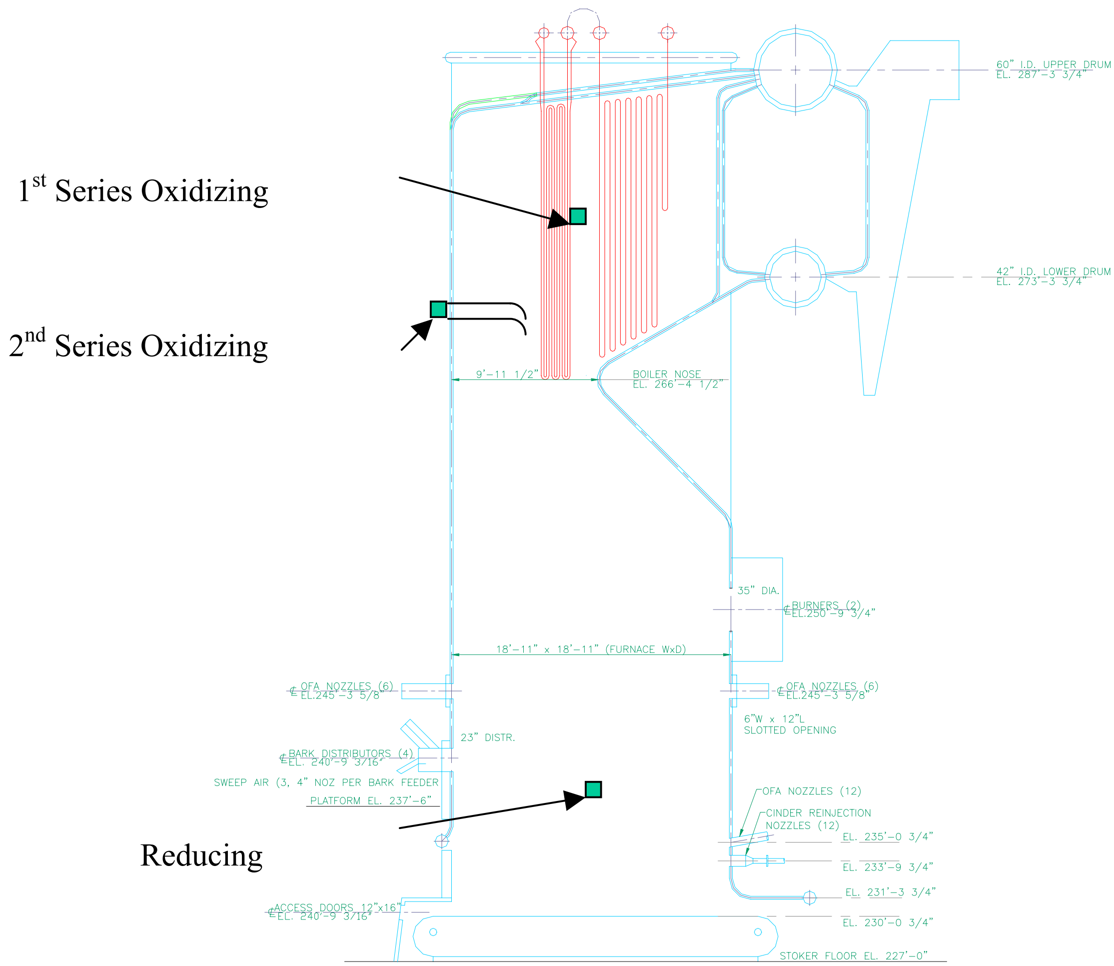


Figure 4.2-1. Locations of Tube Samples in No. 2 Bark Boiler at Boise DeRidder

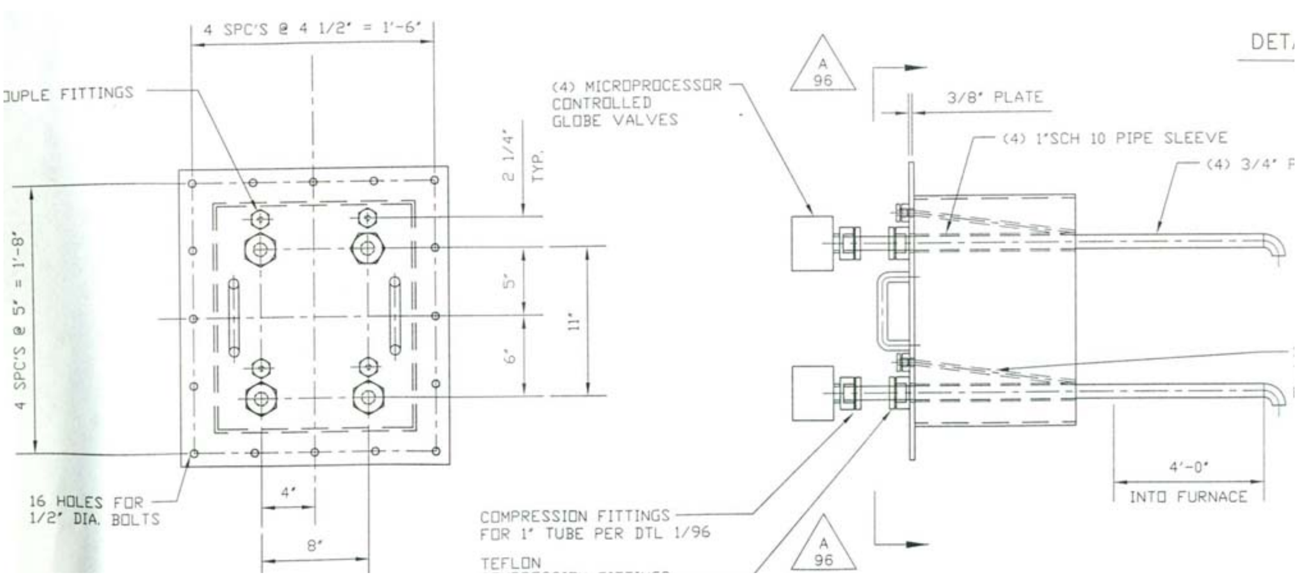
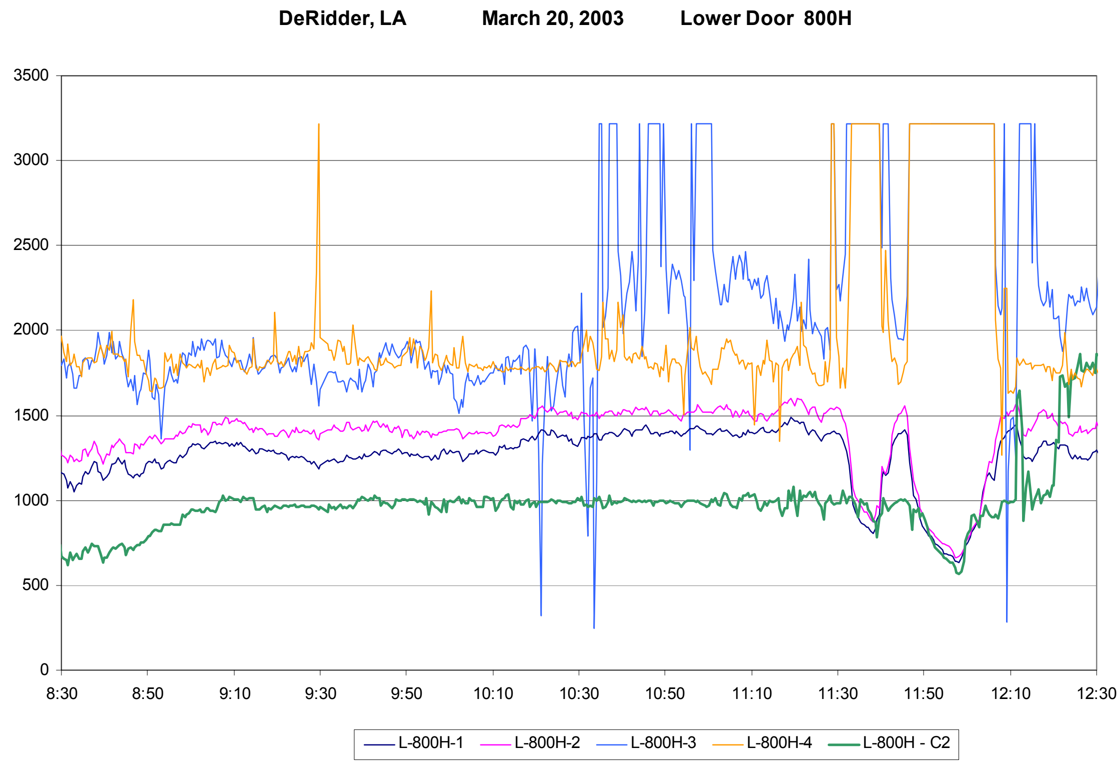


Figure 4.2-2 Test Assemblies for Tube Testing in No. 2 Bark Boiler DeRidder



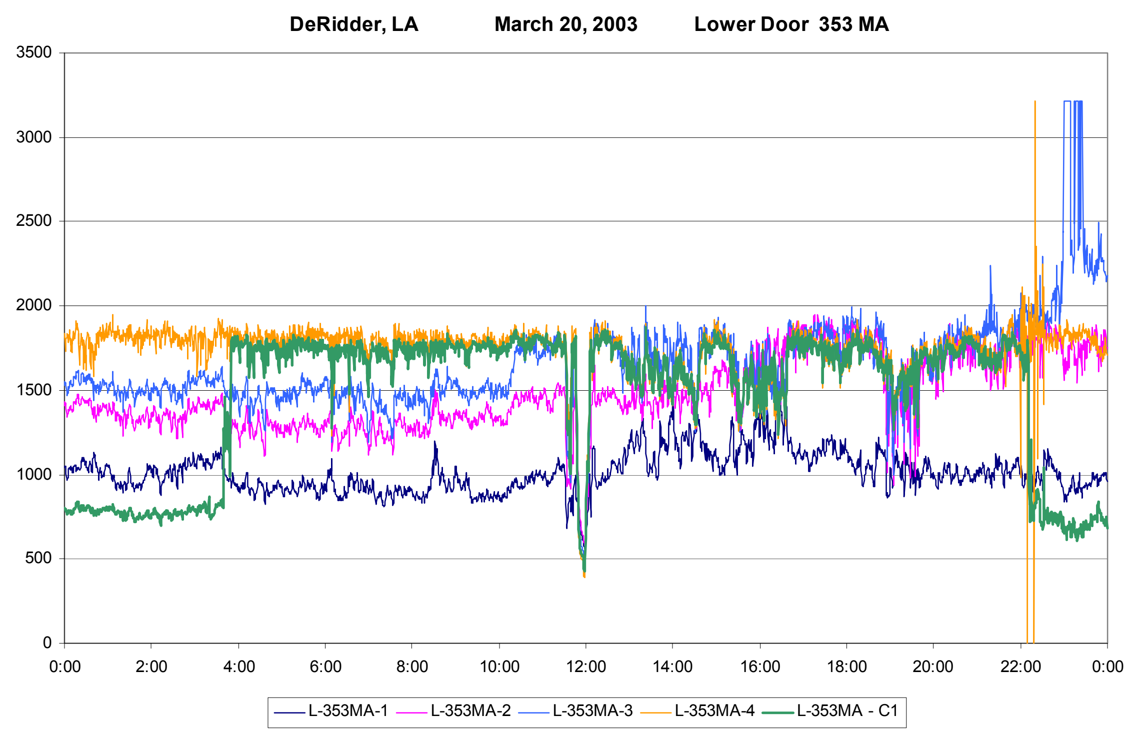


Figure 4.2-3. Typical Temperature Profile for Test Tube Coupons with air temperature control (800 H) and skin temperature control (353 MA)

APPENDIX A
EQUIPMENT LIST

This appendix lists the major equipment and the associated physical and capacity/rating parameters. The operating and performance parameters are presented in the respective descriptive sections in Sections 3.1 through 3.5.

APPENDIX A: PLANT EQUIPMENT LIST

	Equipment	Capacity/Rating
1.0	**Gasification Plant**	
	Manufacturer	Carbona Corp., Finland
	Gasifier Output	146 MMBtu/hr
	Gasifier Input	180 MMBtu/hr
	Cyclone	--
	Start-up Heater Burner	1.57 MMBtu/hr
	Start-up Heater Furnace	--
	Fuel Feeder (2 trains)	65.8 m^3/hr, each train
	Limestone Feeder	1.2 m^3/hr
	Gas Feeder Steam Trap	--
	Process Air Compressor	7.2 kg/hr air
	Ash Discharge Equipment	1.1 m^3/hr
	Nitrogen Generation Equipment	
	- Nitrogen Generator	65 g/sec
	- Air Compressor	0.21 kg/sec
	Flare Equipment – Burner & Stack	9.8 kg/sec
2.0	**Bark Dryer**	
	Manufacturer	MEC Company, USA
	No. of Dryers	2
	Drying Capacity	24,500 kg/hr, each
	Other Equipment include Wet Bark Feed Valve, Inlet Air Damper, Flue Gas Inlet Damper, Natural Gas Burner, Drying Drum, Drop-out Box, Bark Discharge Valve	
	Overall Dimensions	36.9m Lx10.2m Wx20.1m H
3.0	**External Air Heater (AH-1)**	
	Type	Shell and U-Tube
	No. of Heater	1
	Heat Transfer Area	743 m^2
	Type	Pure Counter Flow
	Tube Outside Diameter	63.5 mm
	Overall Dimensions	2.9m int. dia. x 11.9m high

APPENDIX A: PLANT EQUIPMENT LIST (Cont'd)

Equipment	Capacity/Rating
4.0 Boiler Modification	
Internal HT HP Air Heater	
Manufacturer	Babcock Power Services, USA
No. of Heater	1
Heat Transfer Area	437 m^2
Type	Parallel Cross Flow, Radiant
Tube Outside Diameter	70 mm
Tube Materials	SB-407, 800 HT and Haynes 230
Location	Boiler Rear Wall Below Present HMZ OFA nozzles
Syngas Injection Nozzles	
No. and size of Nozzles	12; 152.4 mm OD, each
Location	Above the Top of Auxiliary Natural Gas Burners and Below the Furnace Arch Tip
5.0 Gas Turbine Generator	
Manufacturer	Alstom
No. of Turbines	2, Each Single Shaft
Standard Rating	17 MW, each
Generators	12 kV, Air Cooled
Accessories	
Include Air Compressor, Lube Oil Cooler, Control Panel, Intake Air Filter, External Combustor	
6.0 Natural Gas Compressor for Gas Turbines	
No. of compressors	2, one for each gas turbine
Type of compressor	ARIEL JGH4 four throw double acting
Rating	6,000 std. cu. m per hr, 890 rpm, each
Motor	350 kW, 4000 V, each

APPENDIX A: PLANT EQUIPMENT LIST (Cont'd)

Equipment	Capacity/Rating
7.0 Heat Recovery Steam Generator (HRSG)	
No. of HRSGs	2
Type	unfired, two pressure, non-reheat, natural circulation, drum type with horizontal gas flow
Stack	2 Nos., 6m D, 42.7m H
Deaerator	1
Accessories	Feed Pumps, Feedwater stop and check valves, relief valves, continuous and intermittent blowdown system, and economizer bypass, chemical treatment equipment
8.0 Electrical Distribution	
Step-up Transformer	2, Oil-filled, 20 MVA
Station Service Transformer	1, Oil-filled, 5 MVA
Low Voltage Load Center Transformer	2, 500/750 kVA

Other Equipment include: 4.1 kV Switchgear; 400 V MCCs; 400 V Load Switchgear, Protective Relay Panel, Cathodic Protection, Lightning Protection

9.0 Balance-of-Plant

9.1 Cooling Water System

Equipment	Capacity/Rating
Cooling tower and fan	1 Cell, Mechanical Draft 4.5×10^6 kJ/hr
Circulating Water Pumps	2-100%; 22,500 kg/hr each Motor – 5 kW, 1,500 RPM
Cooling Water Pumps (Closed Loop)	2-100%; 22,500 kg/hr each Motor – 10 kW, 1,500 RPM

Other Equipment include: 2-100% Cooling Water Heat Exchangers and 1Surge Tank

9.2 Compressed Air System

Equipment	Capacity/Rating
Air compressor	2 – 100% capacity 1,700 scfm /hr, 7.0 bar each 200 kW, 380 V Motor each

Other Equipment includes: air receiver; accumulator; pre-filter; after-filter; dryer

9.3 Fire Protection System

Equipment	Capacity/Rating
Fire Suppression and Extinguishing System	FM-200 total system or fire extinguisher High pressure CO_2 system

Other Equipment includes: Stand pipes, fire hydrants, fire hose, sprinklers

APPENDIX A: PLANT EQUIPMENT LIST (Cont'd)

	Equipment	Capacity/Rating
9.4	**Conveying, Air Piping, Ducting**	
	Bark Conveying	
	Screening/Sizing Machine	145 m^3/hr
	From Junction Box #14 to Dryers	1.2m W x 110m L
	From Dryers to Junction Box #15	1.2m x 91m L
	From Junction Box #15 to Screening Machine	1.0m x 37m L
	From Screening Machine to Gasifier	1.0m W x 30m L
	From Screening Machine to Existing Conveyor	0.5m x 60m L
	External Air Heater (AH-1) Piping	
	Air piping from GT-1 to AH-1	250mm ID; 45m long
	Air piping from AH-1 to GT-1	300mm ID; 50m long
	Internal Air Heater (AH-2) Piping	
	Air piping from GT-2 to AH-2	250mm ID; 28m long
	Air piping from AH-2 to GT-2	300mm ID; 33m long
	Product gas piping from gasifier plant to boiler	122m long
	Flue gas ducting	
	From boiler to dryers	55m
	From dryers to boiler	91m
	From HRSGs to boiler	137m

APPENDIX B
SOLAR TURBINES INCORPORATED
FINAL REPORT

Advanced Gasification Based Fuel Conversion and Electricity Production System for Forest Product Industry
Power Island Pre-design Study

GTI Subcontract: PF 13524

DOE/GTI Contract: DE-FC26-01NT41108

Final Report

October 31, 2003

Submitted To:

GAS TECHNOLOGY INSTITUTE

Submitted By:

SOLAR TURBINES INCORPORATED
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Executive Summary

Project Background

Boise Paper Solutions (Boise) and the Gas Technology Institute (GTI) have teamed to develop, demonstrate, and place in continuous operation an advanced biomass gasification-based, power generation system suitable for near-term commercial deployment in the forest products industry. The program aims to develop and install a system that will be used in conjunction with, rather than in place of, existing wood waste fired boilers and flue gas cleanup systems. The main objective of the initial development phase of this program is to define a system that avoids the major hurdles of high-pressure gasification (i.e., high-pressure fuel feeding, ash removal, and hot gas cleaning) that are typical for conventional IGCC power generation. The specific system under study uses an atmospheric pressure biomass gasifier and an externally recuperated gas turbine.

Introduction

As part of the pre-design evaluation study, Solar Turbines Incorporated (Solar) completed the following tasks using a Titan 130 gas turbine as a baseline:

i. Propose and analyze different candidate cycles for an Externally Recuperated Gas Turbine (ERGT)

ii. Assess the viability of modifying existing Solar gas turbine designs to the ERGT cycle.

iii. Provide GTI with information required to install and commission a standard Titan 130 package so as to enable GTI to assess the viability of installing an ERGT in an existing forest products industrial site.

iv. Support an assessment of the market potential for an ERGT in the forest products industry.

Technical Summary

Solar considered six different ERGT cycles for an ERGT system comprised of two Titan 130 sized turbines. Of these six options two were selected as the most suitable based on the practical operational constraints. These two options were thermodynamically analyzed over a range of operating parameters. Potential operating cost savings and mechanical design feasibility was also evaluated.

Three different ERGT development parameters were considered in defining system configuration. The scenarios reflected three potential development strategies

- Development for a near-term application and requiring only moderate modification
- A long-term engine development program requiring a significant developmental effort and considerable modification to the present engine designs.
- Modifications and developmental effort required specifically for Solar engines to operate in the ERGT configuration with current operating constraints (temperatures and material considerations).

Solar provided information on the layout, installation and commissioning of Solar's standard Titan 130 gas turbine package. This data included the arrangement and layout drawing of a standard T130 package and information regarding the various pumps, compression systems and cooling systems. This information was provided to help GTI assess the feasibility and the economics of modifying an existing facility at the Boise DeRidder paper mill to accommodate an ERGT.

Finally Solar assessed the market potential for an ERGT from a gas turbine manufacturer's perspective.

Conclusions and Recommendations

Based on the results of the thermodynamic analyses it is concluded that increasing the high pressure/high temp (HP/HT) heat exchanger exit temperature and reducing the pressure drop across the heat exchanger will help maximize the fuel cost savings. Of the various options considered, the most beneficial one in terms of fuel cost savings and the net power output was the ERGT cycle that used steam as the cooling medium for its turbine and had a humidifier installed at its inlet. However, in view of the major redesign effort required to develop a steam cooled turbine it is recommended that presently available air-cooled turbines be used with an inlet fogger/humidifier when the ambient conditions warrant its use.

Based on an analysis of the current operating constraints on the various components of an ERGT, it is recommended that an ERGT be developed by modifying existing gas turbines to accommodate a heat exchanger outlet temperature (combustor inlet temperature) of no more than 1450°F. In addition, an HP/HT air heater that works with a pressure drop at or below 15 psi is recommended so as to minimize changes to present gas turbine designs and keep development costs low.

From a long-term perspective, the ERGT cycle potential can be realized if significant improvements can be made to effectively sustain air temperatures of up to 1800ºF. Critical components of the cycle include: high-pressure air heater (having a low pressure-drop), boost combustor, scroll and engine casings.

1 Introduction

Boise Paper Solutions (Boise) and the Gas Technology Institute (GTI) have teamed up to develop, demonstrate, and place in continuous operation an advanced biomass gasification-based power generation system suitable for near-term commercial deployment in the forest products industry. The program is being funded by the US Department of Energy and Gas Research Institute. The program aims to develop and install a system that will be used in conjunction with, rather than in place of, existing wood waste fired boilers and flue gas cleanup systems. The novel system is expected to include three advanced technological components based on GTI's RENUGAS® and METHANE de-NOX® technologies, and a concept used in the HIPPS program. The main objective of the development phase of this program is to design a system that avoids the major hurdles of high-pressure gasification (i.e., high-pressure fuel feeding and ash removal, and hot gas cleaning) that are typical for conventional IGCC power generation. It aims to also minimize capital intensity and technology risks. The system shall meet the immediate needs of the forest products industry for highly efficient and environmentally friendly electricity and steam generation systems utilizing existing wood waste as fuel resources.

As part of the pre-design evaluation study phase, Solar Turbines Incorporated (Solar) accepted a sub-contract from GTI (contract number PF13524) to propose and analyze different possible cycles for the operation of an Externally Recuperated Gas Turbine (ERGT) and then assess the viability of modifying existing gas turbine designs to suit the ERGT cycle. It was mutually agreed that the analysis be based on a Titan 130 size gas turbine.

As part of the subcontract, Solar agreed to provide GTI with information required to install and commission a standard Titan 130 package so as to enable GTI to assess the viability of installing an ERGT in an existing forest products industrial site. Finally Solar also agreed to participate (from a gas turbine manufacturer's perspective), in an estimate of the market potential for an ERGT in the forest product industry.

2 Project Approach

For this study Solar evaluated the Titan 130 turbine since GTI, required a power generation system that provides 27 MW of electrical energy output. Two Titan 130 size gas turbines are required to meet the electrical demand. Based on these evaluations Solar proposed various cycle options.

In the next stage Solar, with inputs from GTI, narrowed the proposed cycle options to the two most promising ones and conducted thermodynamic cycle analyses of these two options for various engine parameters including but not limited to turbine rotor inlet temperature (TRIT), inlet air humidity and different turbine cooling scenarios.

Based on the results of these cycle analyses, Solar has provided recommendations on

- The feasible operating conditions with respect to
    - Thermodynamic performance
    - Engine design

- Engine layout and configuration

Solar has also classified the range of parameters, which are thermodynamically feasible, as

- Appropriate for the near-term application and will require moderate modification,
- Appropriate for a long-term engine development program that would require a significant developmental effort and considerable modification to the present engine designs,
- Modifications and developmental effort required on Solar engines to achieve the parametric ranges that are feasible from both thermodynamic and mechanical design aspects.

In a parallel effort Solar provided GTI with information regarding Solar's existing Titan 130 package layout and installation and commissioning. Included were the arrangement and layout drawing of a Standard T130 package and information regarding the various pumps, compression systems and cooling systems. This information was intended to help GTI assess the feasibility and the economics of modifying an existing facility at one of Boise's paper mills to accommodate an ERGT.

Finally, Solar assessed the market potential of an ERGT from the gas turbine manufacturer's perspective.

3 Project Outcomes

3.1 Design Information Definition and Inquiry

3.1.1 Proposed Cycle Concepts

In the initial phase of this study Solar evaluated the gas turbine requirements of the program with an understanding that GTI required a power generation turbine system that provides 27 MW of electrical energy output. Configurations based on two Titan 130 size gas turbines were studied. Based on the information provided, Solar with input from GTI proposed the different cycle options detailed below.

<u>Option 1</u>

Option 1 consists of two independent ERGTs, each having the heat exchanger and boost combustor in series. A schematic layout of the cycle is shown in *Figure 3.1*.

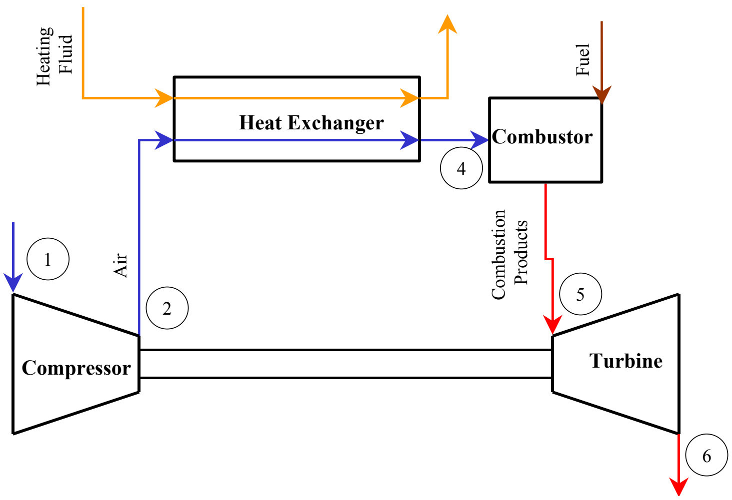

Figure 3.1 ERGT With Boost Combustor in Series With The Heat Exchanger

Option 2

Option 2 consists of two independent ERGTs, each having the heat exchanger and boost combustor in parallel. *Figure 3.2* shows a schematic layout of the cycle.

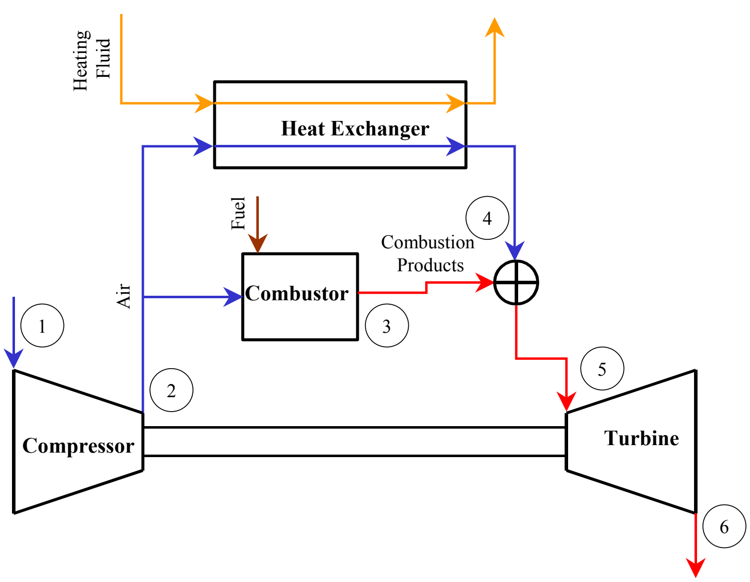

Figure 3.2 ERGT With Boost Combustor in Parallel With The Heat Exchanger

Option 3

This option consists of two ERGTs, each of which is connected to an independent generator. However, the compressed air flow from the compressors of the two ERGTs is combined into a single flow stream and fed through a single heat exchanger and boost combustor (in series). The products of combustion are then split into two streams and fed to the respective turbines of the two ERGTs. A schematic layout of this cycle is shown in *Figure 3.3*.

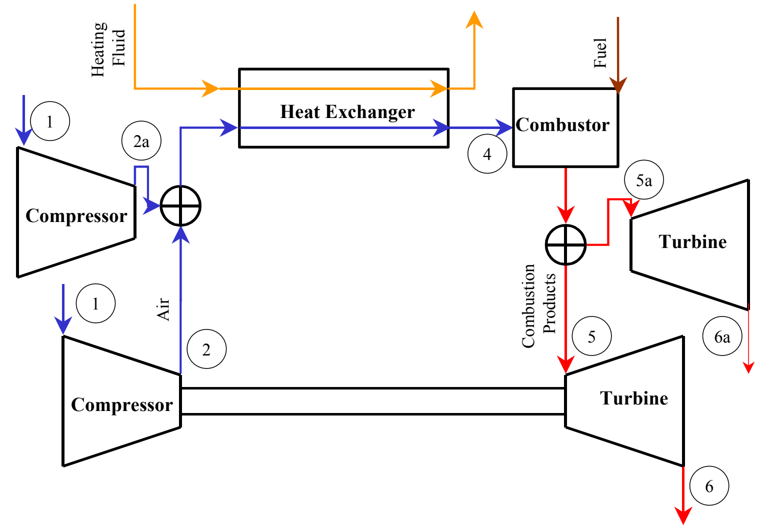

Figure 3.3 *System of Two ERGT With Single Boost Combustor in Series With The Heat Exchanger*

Option 4

Option 4 is similar to Option 3 with the only difference being the layout of the common heat exchanger and the boost combustor. Here the common heat exchanger and the boost combustor are piped in parallel as opposed to being in series. A schematic layout of this cycle is shown in *Figure 3.4*.

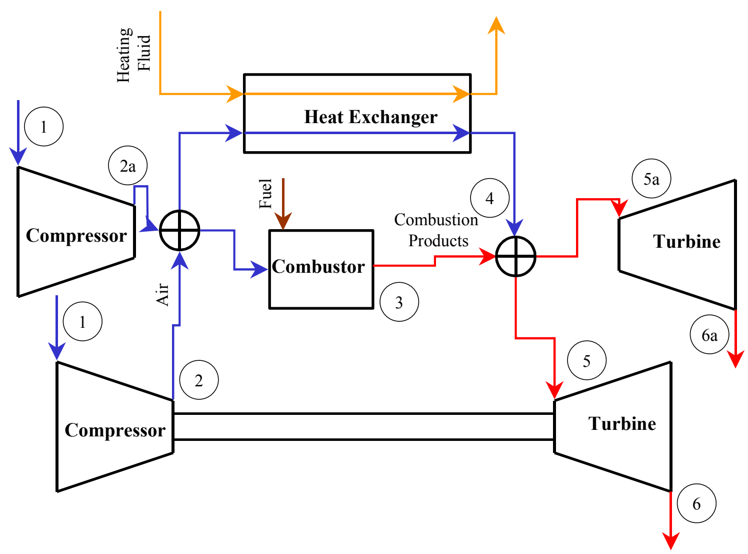

Figure 3.4 *System of Two ERGT With Single Boost Combustor in Parallel With The Heat Exchanger*

Option 5

This option consists of two ERGTs, each of which is connected to an independent generator. Each ERGT has its own compressor, boost combustor and turbine. The compressed air flow from the individual compressors is combined into a single flow stream and fed through a single heat exchanger. Downstream of the heat exchanger the air stream is split into two streams, each of which feeds into the boost combustor of the two turbines. A schematic layout of this cycle is shown in *Figure 3.5*.

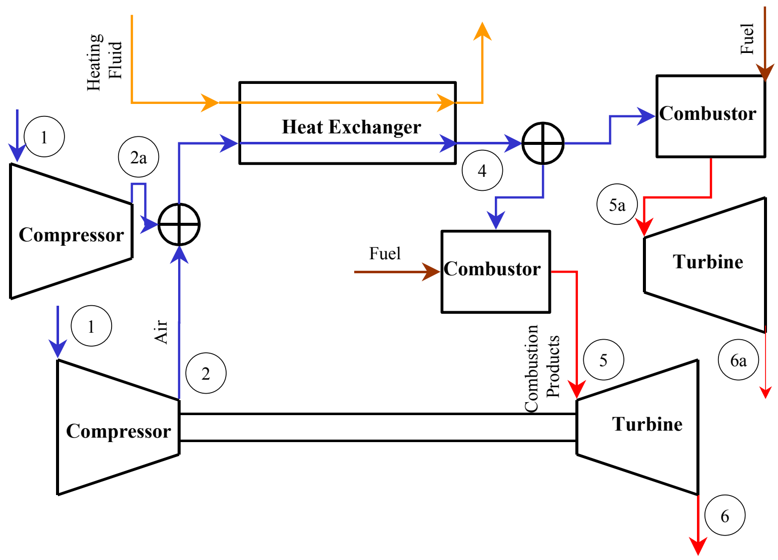

Figure 3.5 System of Two ERGT With Two Boost Combustors in Series With a Single Heat Exchanger

Option 6

Option 6 is similar to Option 5 with the only difference being the layout of the boost combustors. Here the boost combustors are piped in parallel to the heat exchanger, as opposed to being in series. A schematic layout of this cycle is shown in *Figure 3.6*.

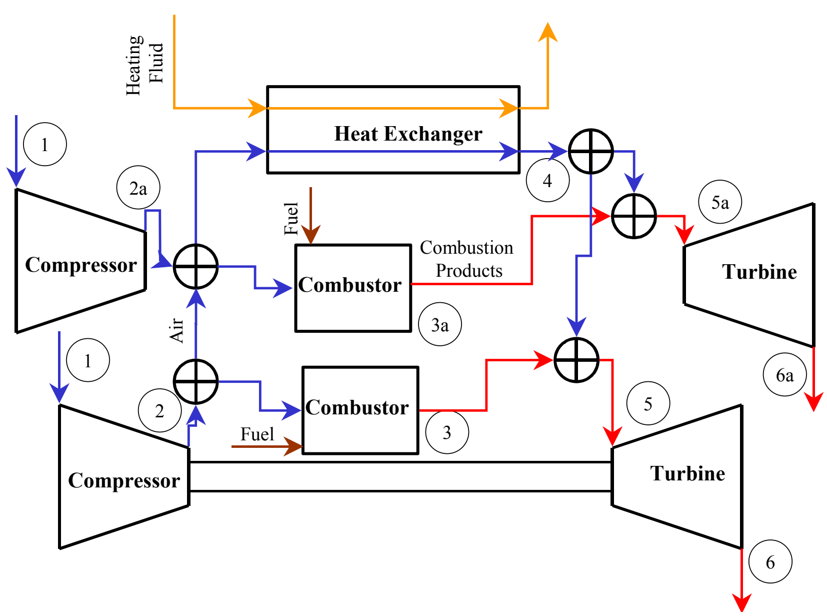

Figure 3.6 System of Two ERGT With Two Boost Combustors in Parallel With a Single Heat Exchanger

3.1.2 Cycle Selection

Based upon the desire for system and controls simplicity, *Option 3* through *Option 6* were discarded. These options add controls and synchronization complexities that would increase the capital and operating cost of the equipment. Based on the space available at the pilot plant test site at DeRidder Paper Mill, DeRidder, Louisiana, it was determined that it would be possible to install two independent ERGT units of Titan 130 size. Therefore it was decided to limit the cycle analysis for this feasibility study to *Option 1* and *Option 2.*

3.1.3 Standard Titan 130 Cycle Data

The cycle analysis and the design study presented in this report used the Titan 130 engine performance as the baseline. The performance data for this baseline single shaft Titan 130 gas turbine under ISO conditions is detailed in *Table 3.1*, while the composition of Natural gas used for the cycle analysis is detailed in *Table 3.2.*

Table 3.1 Gas Turbine Data

Parameter	**Units**
Number of compressor stages	14
Number of turbine stages	3
Pressure at compressor inlet	14.7 psia
Pressure at compressor outlet pressure	230.25 psig
Pressure ratio (outlet/inlet)	16.663
Ambient air temperature	60ºF
Air flow rate through the compressor	6478.9 lb/min
Gas composition	Natural Gas*
Fuel flow rate ‡	112.02 lb/min
Temperature at turbine outlet	908.34ºF
Pressure at turbine outlet	14.7 psia
Flow rate out the turbine	6590.92 lb/min

**Note 1: The composition of natural gas assumed is detailed in Table 4*
‡Note 2: The lower heating value of the fuel is 20167.86 Btu/lb

Table 3.2 Fuel Composition

Parameter	**Volume Percent**
CH_4	92.7899
C_2H_6	4.16
C_3H_8	0.84
C_4	0.18
C_5	0.04
C_6	0.04
CO_2	0.44
N_2	1.51
H_2S	0.0001

3.1.4 Parameters for ERGT Cycle study

After discussions with GTI, it was decided to conduct the cycle analysis for a range of parameters that is broader than those acceptable for on the current Titan 130 designs. The ranges of parameters being considered for this are detailed in *Table 3.3*

Table 3.3 Ranges of Parameters Considered for this Study

Turbine rotor inlet temperature	1900°F, 2100°F
Pressure drop across the HP/HT air heater	10, 20, 30 psi
Exit temperature of the HP/HT air heater:	1300°F through 1800°F in increments of 50ºF
HP/HT air heater leakage	0%
Pressure drop across the combustor	3.5% of compressor outlet
Ambient air temperature	80°F
Ambient air RH	60%, humidified using a fogger at inlet
Combustor and turbine cooling	Air, Steam to replace air wherever feasible (no steam injection)
Engine Load Conditions	Full Load

3.2 Cycle Analysis for ERGT cycle

Table 3.4 Inlet Conditions for Performance Code Benchmarking

Inlet Air Temperature	60°F, 80°F
Relative Humidity	60 %
Inlet Air Pressure	Sea level
Pressure Ratio	16.6
Nominal Net Turbine Output	14 MW
Turbine Exhaust Pressure	14.7 psia

A computer code was developed for the purpose of thermodynamic cycle analysis of the various ERGT configurations. The code was benchmarked with results from an in-house proprietary code (simple cycle GT) for inlet conditions detailed in *Table 3.4*. The comparison was made for a standard gas turbine cycle. Based on the comparison it was concluded that the newly developed code could be considered reasonably accurate for purpose of this study. Using this new code, parametric thermodynamic simulations were conducted to map out all of the parametric variations detailed in *Table 3.3*. The results of the analyses were used to assess the cycle configurations and assist in identifying the most cost effective one. The evaluation presented here includes the effects of the various parameters on system performance and fuel costs.

Simulations were performed at full load conditions for fixed TRIT, and ambient conditions, and a fixed turbine-cooling scheme. The pressure drop across the HP/HT air heater was varied between 10 psi and 30 psi in increments of 10 psi, while the exit temperature of the HP/HT air heater was varied between 1300°F and 1800°F in increments of 50°F

For the purpose of evaluating the fuel cost (per KW-hr) the following assumptions were made

i. Full load operation for 8000 hours per year
ii. Natural gas price of $4/MBTU
iii. Wood waste price of $1.78/MBTU
iv. Only 50% of the wood waste used is purchased. The other 50% is generated from an in-house paper manufacturing process so is considered free.
v. Efficiency of the boiler that uses wood waste as fuel and houses the HP/HT heat exchanger is 80%.

3.2.1 Comparison of Option 1 and Option 2 to Baseline

Detailed thermodynamic analyses of Option 1 and Option 2 (schematically shown in *Figure 3.1* and *Figure 3.2*) were conducted for the parameters detailed in *Table 3.5*. Two cycle variants were investigated based on Option 2. Option 2a employs a combustor that is cooled using compressed air at the compressor discharge temperature. Option 2b reflects a combustor cooled using either heated air from the air heater or steam, assuming there is no leakage of the steam into the combustor. For Option 1, the combustor cooling was achieved using heated air from the air heater

Table 3.5 The Parameters Ranges used in the Evaluation of Option 2

Turbine rotor inlet temperature	2100°F
System pressure drop	10 psi
HP/HT air heater exit temperature	1300°F to 1800°F in increments of 50°F
HP/HT air heater leakage	0%
Ambient air temperature	80°F
Ambient air RH	60%
Combustor cooling (Option 1) (Option 2a) (Option 2b)	Air from HP/HT air heater Air at compressor discharge temperature Steam (no steam injection) or air from HP/HT air heater
Turbine cooling	Air at compressor discharge temperature

While conducting the analyses, the flame temperature was restricted to levels at which current gas turbines can operate in a low emissions mode. This restriction defined the amount of air that could bypass the combustor and be heated by the air heater in Option 2. The percentage of air that can be allowed to pass through the HP/HT air heater for the various conditions used in the evaluation of Option 2 is detailed in *Table 3.6.*

Table 3.6 Percentage of Air Allowed to Pass Through the Air Heater as a Function of Air Heater Exit Temperature

Configuration	**Percentage of air passed through the air heater for different Heater Exit Temperatures**					
	1300°F	**1400°F**	**1500°F**	**1600°F**	**1700°F**	**1800°F**
Option 2a	7.57	8.33	9.27	10.48	12.07	14.25
Option 2b	35.88	38.1	40.64	43.58	47.01	51.06

The results of the cycle analyses are shown in *Figure 3.7* through *Figure 3.9*. *Figure 3.7* compares the net electrical power output of the three cases with that of the baseline standard gas turbine (no heat exchanger and directly fired). *Figure 3.8* compares the natural gas consumption for the three cases with that of the baseline standard gas turbine, and *Figure 3.9* compares the fuel costs ($/KW-hr) of operating an ERGT based on the three cases with that of a baseline gas turbine. *Figure 3.10* shows the percentage of fuel cost savings achieved while operating an ERGT in comparison with a standard engine.

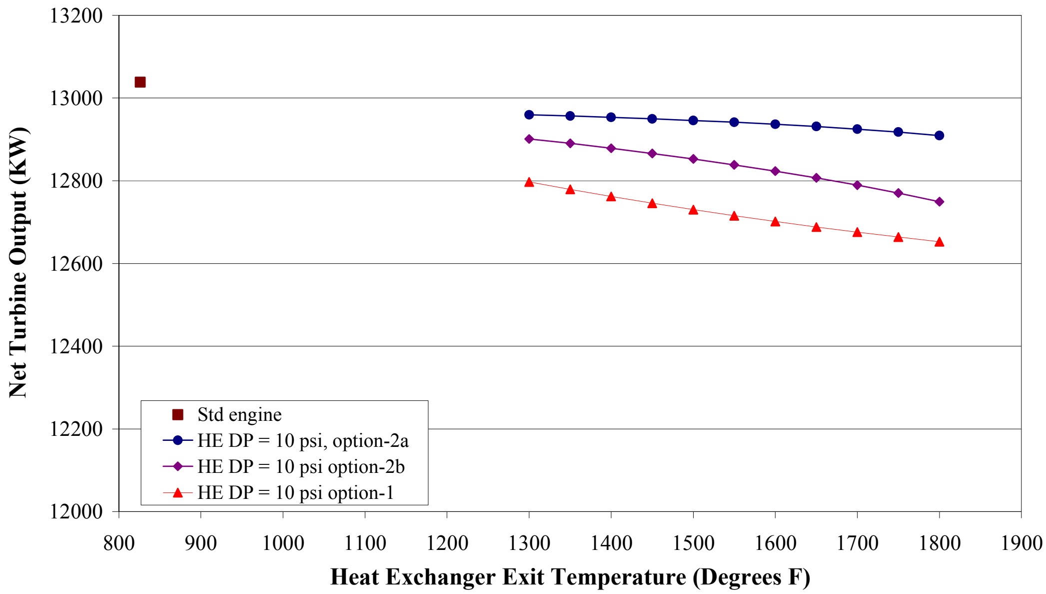

Figure 3.7 Net Turbine Output as a Function of HP/HT Air Heater Exit Temperature for Option 1 and Options 2a & 2b

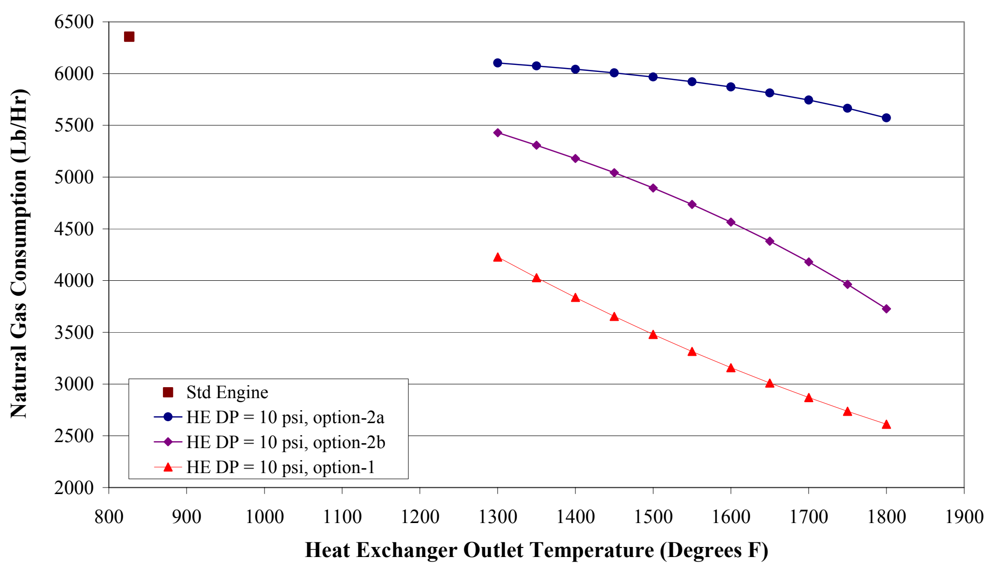

Figure 3.8 Natural Gas Consumption as a Function of HP/HT Air Heater Exit Temperature for Option 1 and Option 2a & 2b

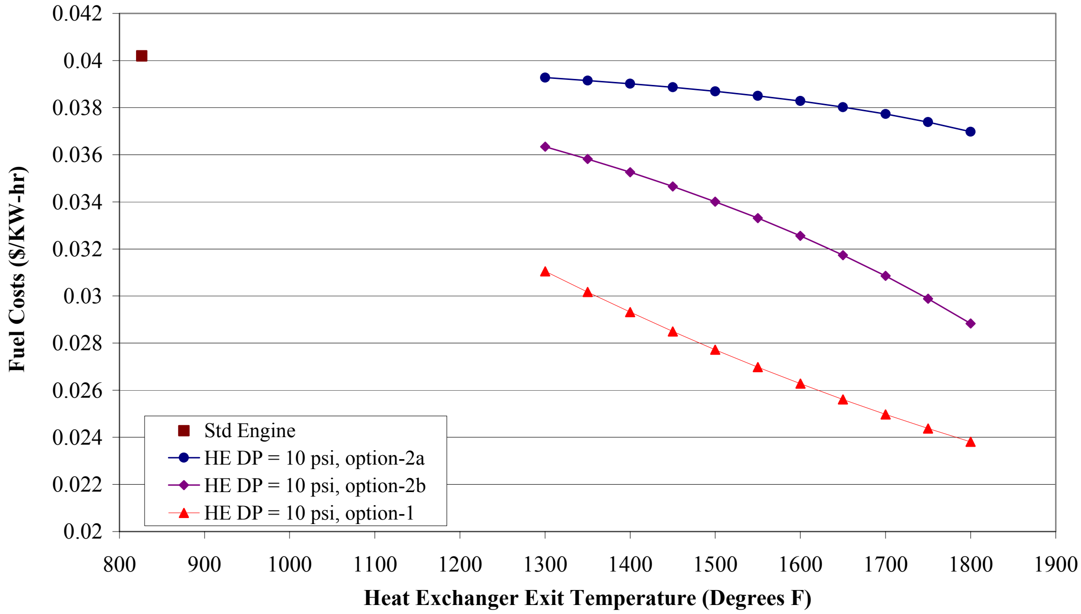

Figure 3.9 Fuel Cost as a Function of HP/HT Air Heater Exit Temperature for Option 1 and Option 2a & 2b

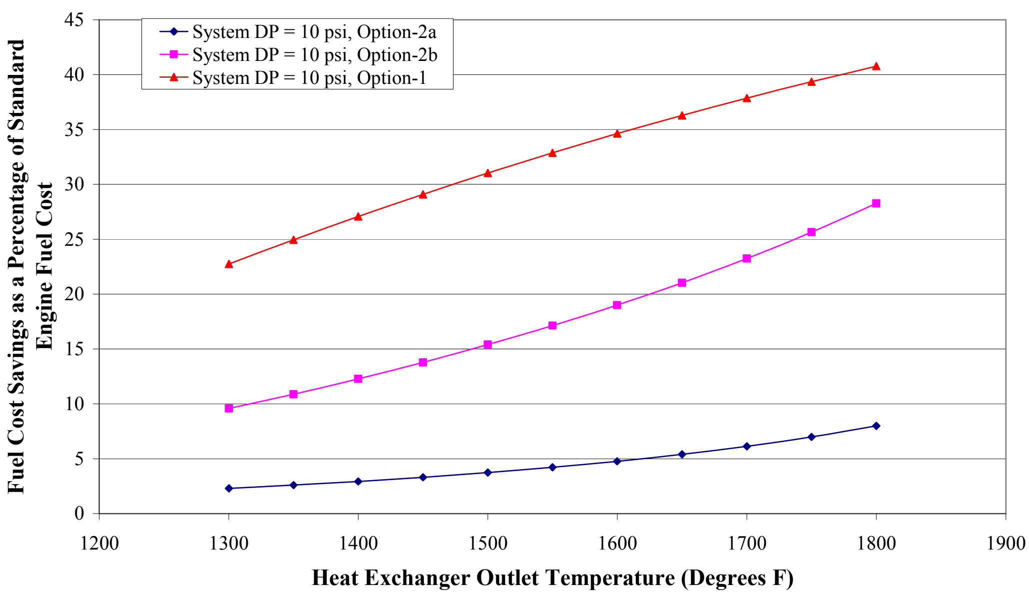

Figure 3.10 Percentage Fuel Cost Savings using the Standard Engine Fuel Costs as a Baseline

Evaluating these results, it is seen that both Option 2a and Option 2b provide a slightly more power than Option 1. However, their savings in fuel cost is small when compared to a standard gas turbine and much smaller than that of Option 1. Since the initial pilot installation will most likely be with air heater exhaust temperatures in the range of 1400ºF to 1500ºF, Option 2 does not provide significant fuel cost savings to warrant development.

Therefore, it was recommended that Option 1 be the preferred cycle for this study and the rest of the analysis presented on this report is based on Option 1.

3.2.2 Performance of ERGT based on Option 1

Detailed parametric, thermodynamic analyses of Option 1, *Figure 3.1*, were conducted for two TRITs, to determine the effect of HP/HT air heater pressure drop and exit temperature on the net turbine power output, natural gas consumption, turbine exhaust temperature and fuel cost. In these simulations, compressed air was used as the cooling medium for the turbine. The results of these analyses were compared with those of the baseline engine and are graphically shown in *Figure 3.11* through *Figure 3.14*, while *Figure 3.15* shows the fuel savings while operating an ERGT as a percentage of the fuel cost needed to run a standard engine.

The TRITs chosen for comparison were 2100ºF and 1900ºF. The TRIT of 2100ºF is currently state of the art for small industrial turbines. The lower TRIT was adopted based on the assumptions that mechanical design of some of the ERGT components (such as the scroll) might force the system to run at a lower TRIT and that operation at lower TIRT could be cost effective when natural gas costs rise.

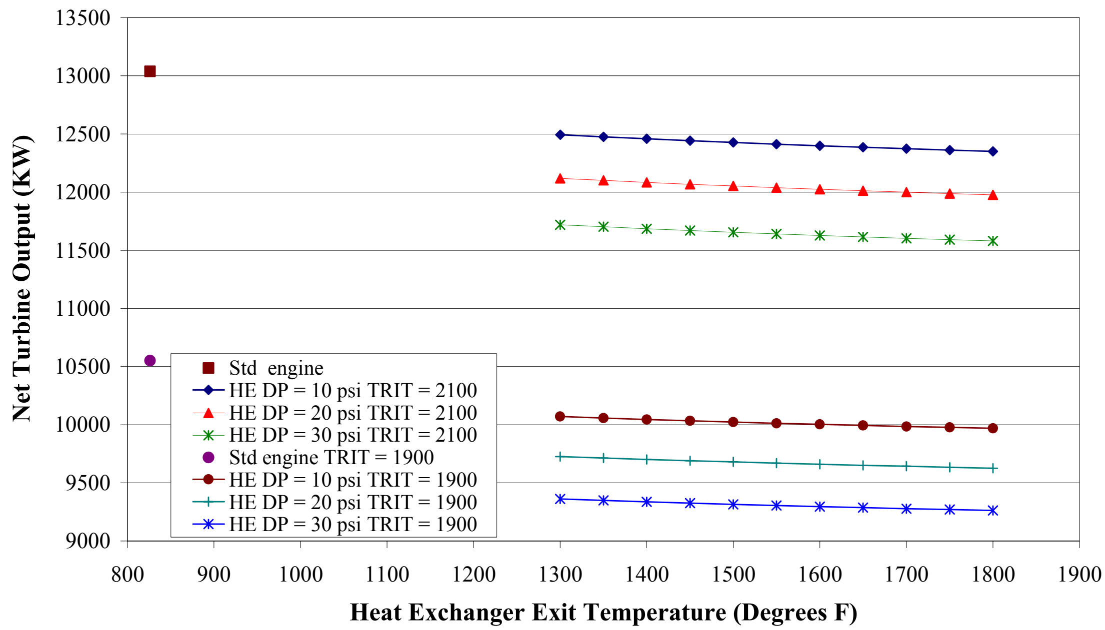

Figure 3.11 Net Turbine output as a Function of HP/HT Air Heater Pressure Drop and HP/HT Air Heater Exit Temperature

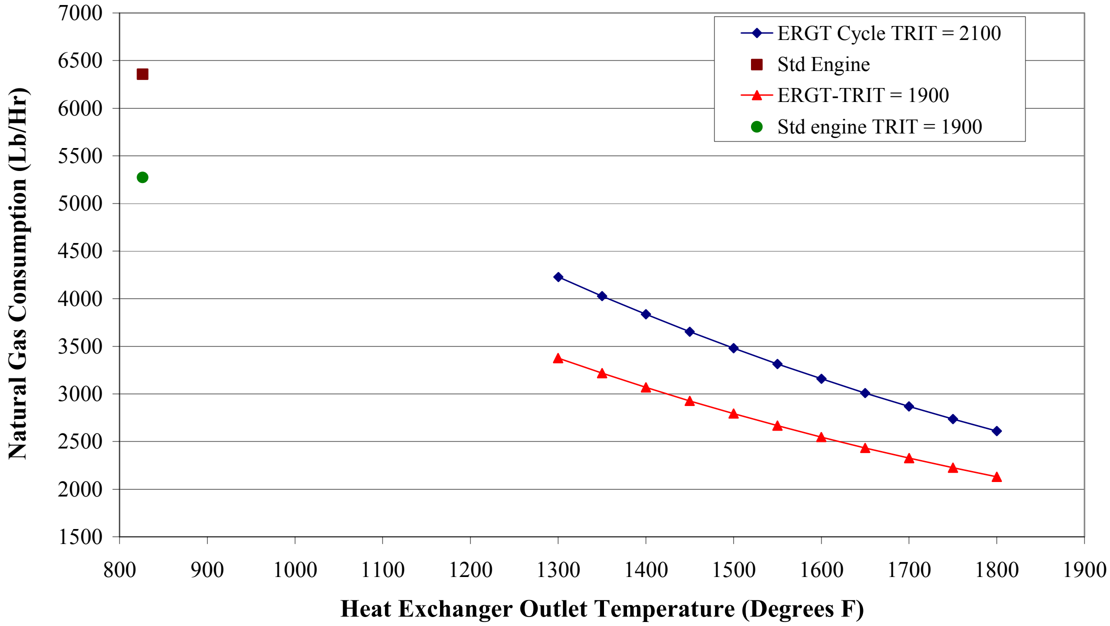

Figure 3.12 Natural Gas Consumption as a Function of HP/HT Air Heater Exit Temperature

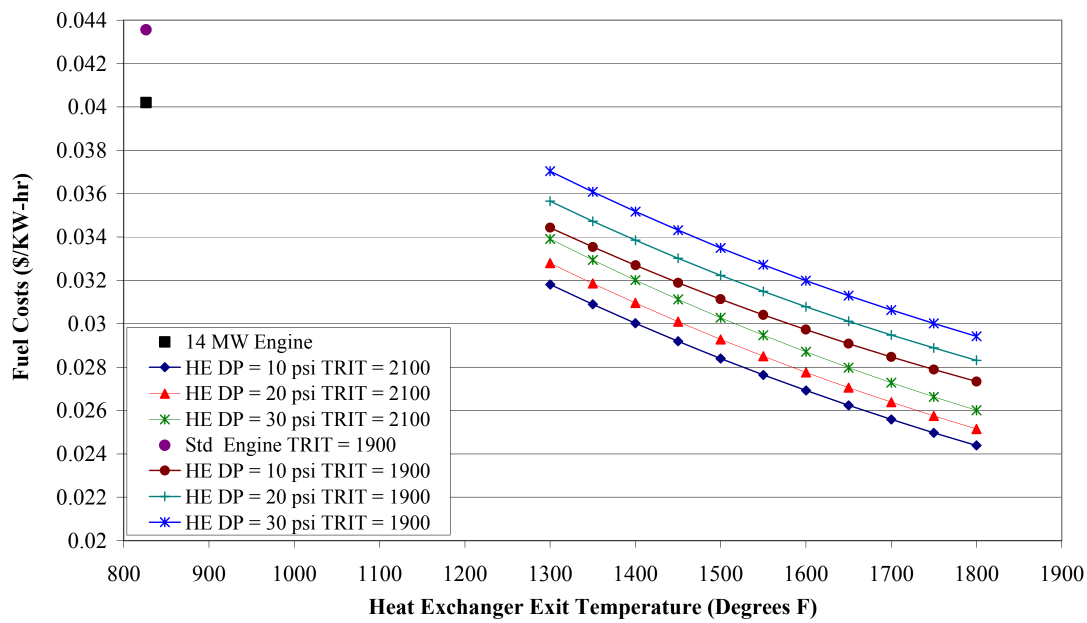

Figure 3.13 Fuel Cost as a Function of HP/HT Air Heater Pressure Drop and HP/HT Air Heater Exit Temperature

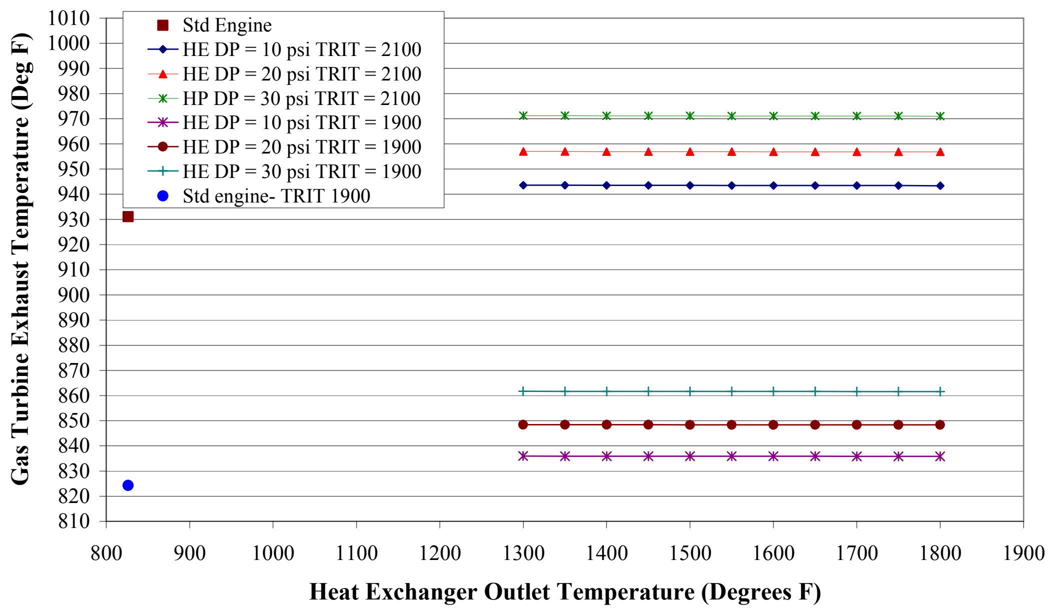

Figure 3.14 Gas Turbine Exhaust Temperature as a Function of HP/HT Air Heater Pressure Drop

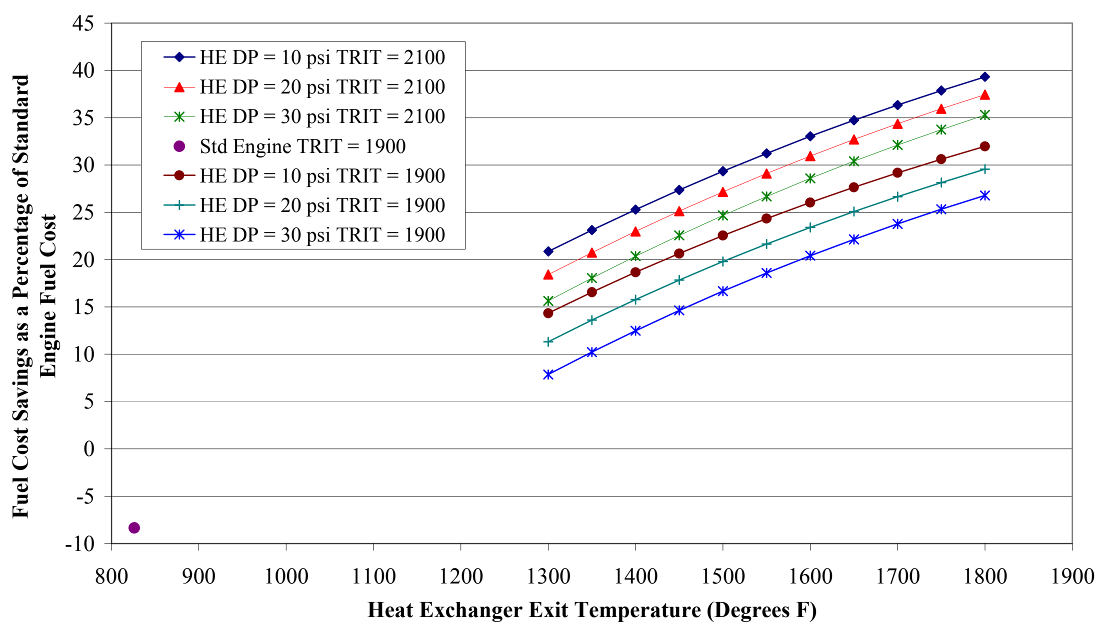

Figure 3.15 Percentage Fuel Cost Savings while Operating an ERGT Using the Standard Engine Fuel Costs as a Baseline

These results indicate that increasing the heat exchanger exit temperature and lowering its pressure drop, while increasing the TRIT, will maximize the ERGT savings and make the cycle more attractive. Note that unless the TRIT is increased beyond 2100ºF, there will always be a small penalty on the net turbine KW by using an ERGT. However, the ERGT will provide higher exhaust heat relative to the standard gas turbine when both are compared at the same TRIT.

3.2.3 ERGT with Steam Cooled Turbines

Additional thermodynamic analysis of Option 1 was conducted to evaluate the effect of using steam as the cooling medium for the turbine. Using steam allows more air to flow through the power turbine, increasing output. These simulations were performed for a TRIT of 2100ºF. The variables involved in this analysis were the HP/HT air heater pressure drop and exit temperature, while the parameters used for the comparison were the net turbine power output, natural gas consumption, turbine exhaust temperature, cost of fuel per KW-hr and the percentage of savings in fuel cost. The results of this analysis are graphically shown in *Figure 3.16* through *Figure 3.19*, while *Figure 3.20* shows the percentage of fuel cost saved when operating an ERGT. The basis of the cost savings is taken to be the fuel cost needed to run a standard engine.

This analysis assumes that no steam enters the turbine flow stream and that the steam is available in abundance. In addition the analysis is not evaluating a combined cycle. Therefore some leakage of steam can be permitted in the actual design. Thus the design constraints on the turbine cooling circuit leakage can be made less restrictive.

Due to the pressure drop across the HP/HT heat exchanger, an ERGT with an air-cooled turbine will always have a net electrical output lower than a standard gas turbine. However an ERGT with a steam-cooled turbine can provide an electrical output greater than that of a standard gas turbine for an added pressure drop of up to 20 psi. The results further show that an ERGT can produce an increase in electrical output of about 8% by switching the cooling medium of the turbine from compressed air to steam (comparisons being made for the same pressure drop across the HP/HT heat exchanger).

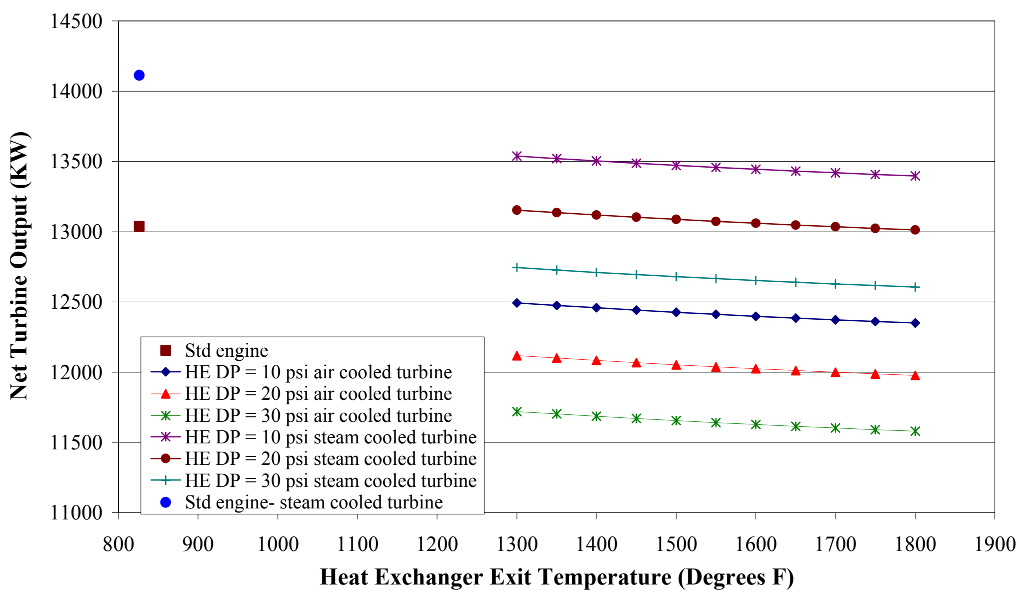

Figure 3.16 Net Turbine Output as a Function of HP/HT Air Heater Pressure Drop and HP/HT Air Heater Exit Temperature (TRIT = 2100ºF)

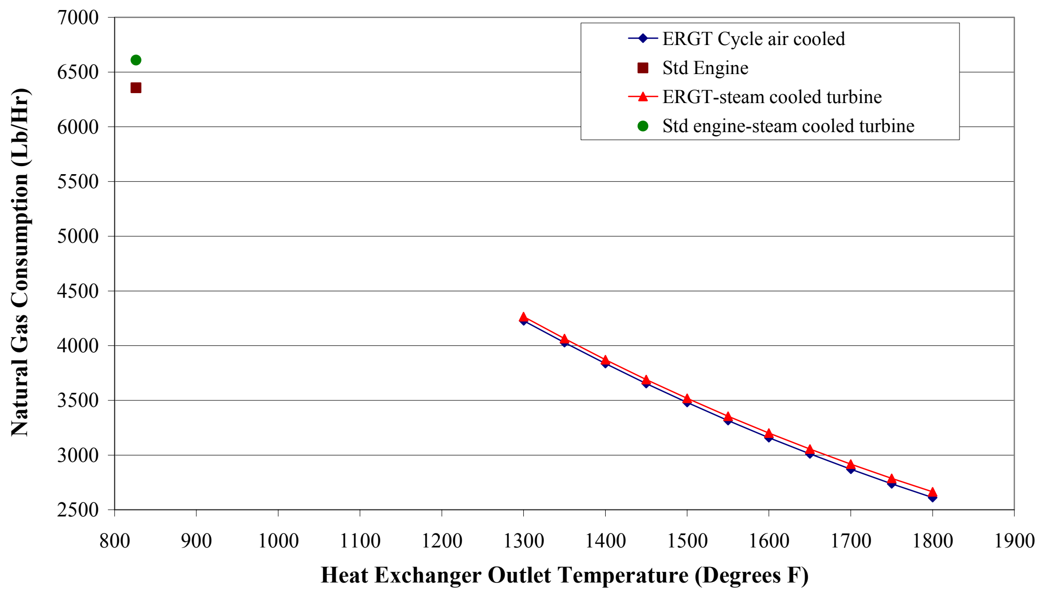

Figure 3.17 Natural Gas Consumption as a Function of HP/HT Air Heater Exit Temperature (TRIT = 2100ºF)

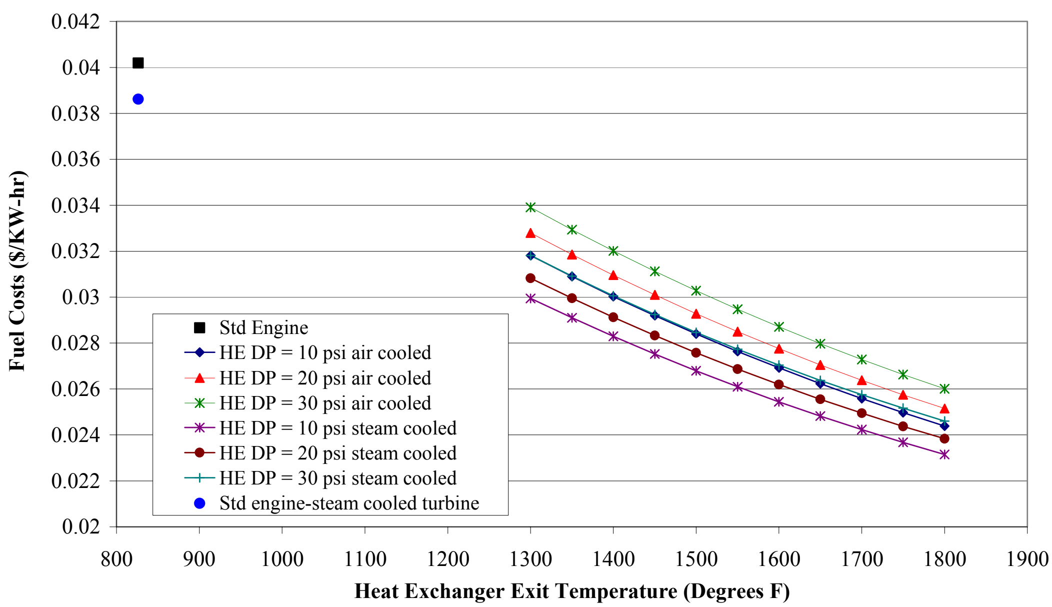

Figure 3.18 Fuel Cost as a Function of HP/HT Air Heater Pressure Drop and HP/HT Air Heater Exit Temperature (TRIT = 2100ºF)

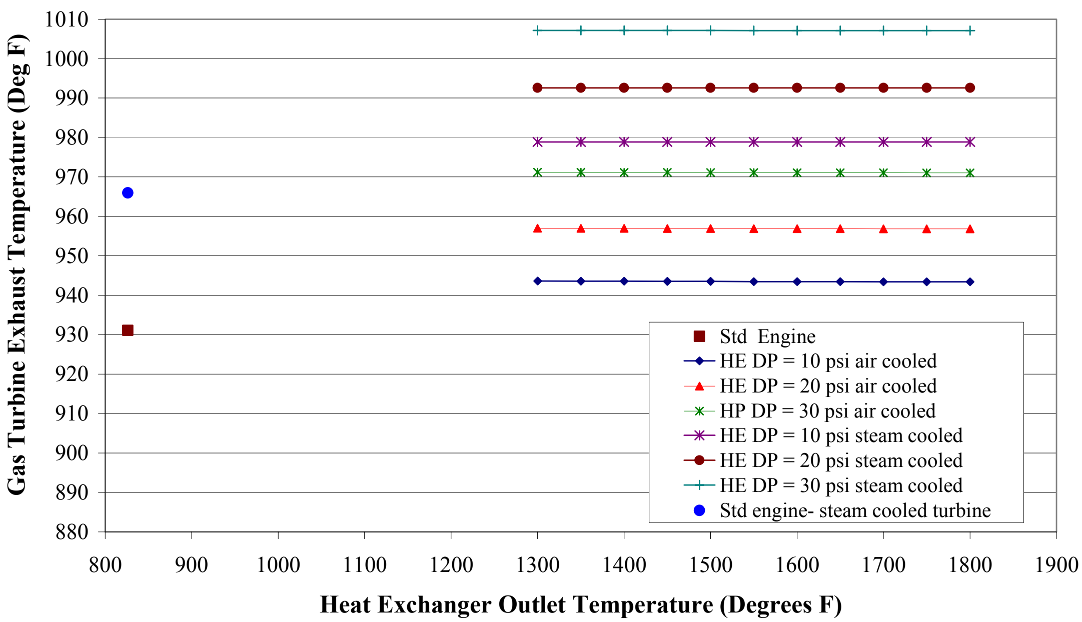

Figure 3.19 Gas Turbine Exhaust Temperature as a Function of HP/HT Air Heater Pressure Drop (TRIT = 2100ºF)

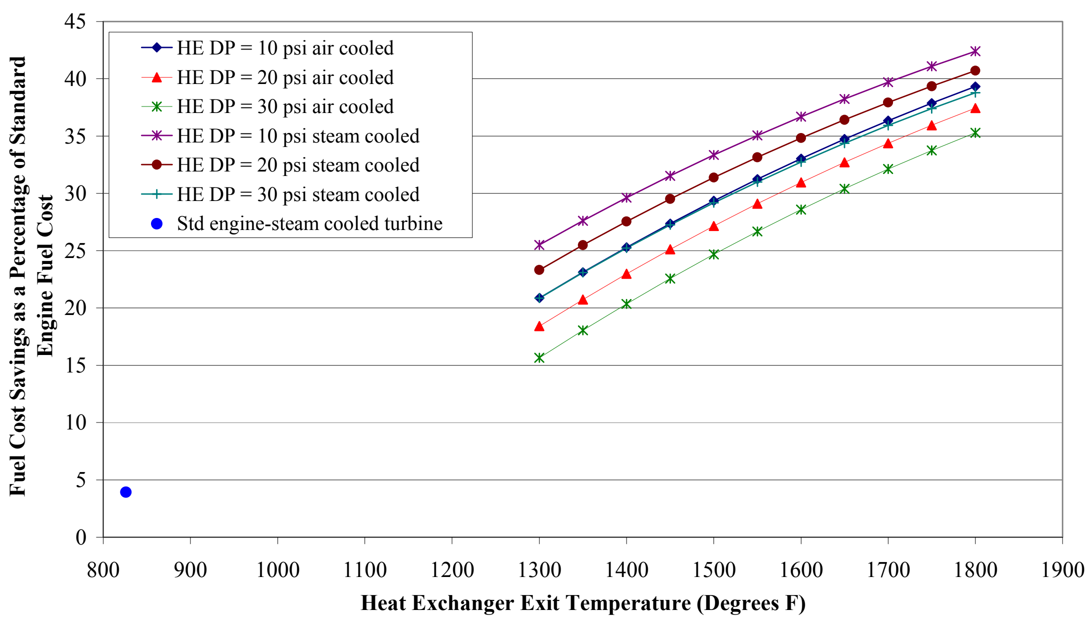

Figure 3.20 Percentage Fuel Cost Savings while Operating an ERGT Using the Standard Engine Fuel Costs as a Baseline

Along with a higher electrical output, the ERGT with steam-cooled turbine also provides a higher exhaust temperature than an ERGT with air-cooled turbine. This aids in co-generation. Further, the ERGT with steam cooled turbine generates a fuel cost savings of at least 23% over a standard gas turbine, while an ERGT with compressed air cooled turbine blades generates a fuel cost savings of at least 15%. Thus using steam as a turbine cooling medium results in an additional 8% of fuel cost savings.

Based on the above discussion, it can therefore be concluded that thermodynamically an ERGT with a steam-cooled turbine is a preferred option.

3.2.4 Effects of Humidification of Inlet Air

Thermodynamic analysis of Option 1 was extended to encompass the effects of using a fogger/humidifier at the inlet of the gas turbine. The use of the humidifier changed the inlet air condition from a temperature of 80ºF and a RH of 60% to a temperature of 70ºF and a RH of 100%. The analysis was conducted to study the effect of the humidifier on an ERGT with an air-cooled turbine (Option 1a) and an ERGT with a steam-cooled turbine (Option 1b). *Figure 3.21* through *Figure 3.25* detail the results of the simulations of Option 1a, while *Figure 3.26* through *Figure 3.30* detail the results of the simulations of Option 1b. The analysis of Option 1b assumes that there is no steam flow into the turbine flow stream and the discussion with respect to steam cooling that is detailed in *Section 3.2.3* is valid.

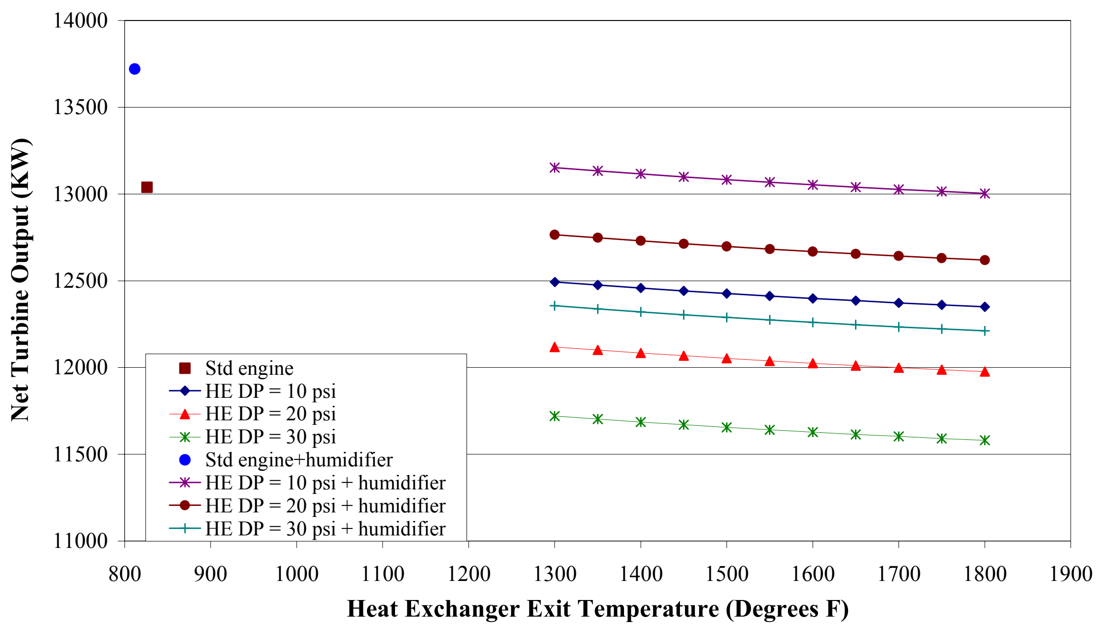

Figure 3.21 Net Turbine Output as a Function of HP/HT Air Heater Pressure Drop and HP/HT Air Heater Exit Temperature for Air-Cooled Turbine with and without a Humidifier at Inlet of Compressor

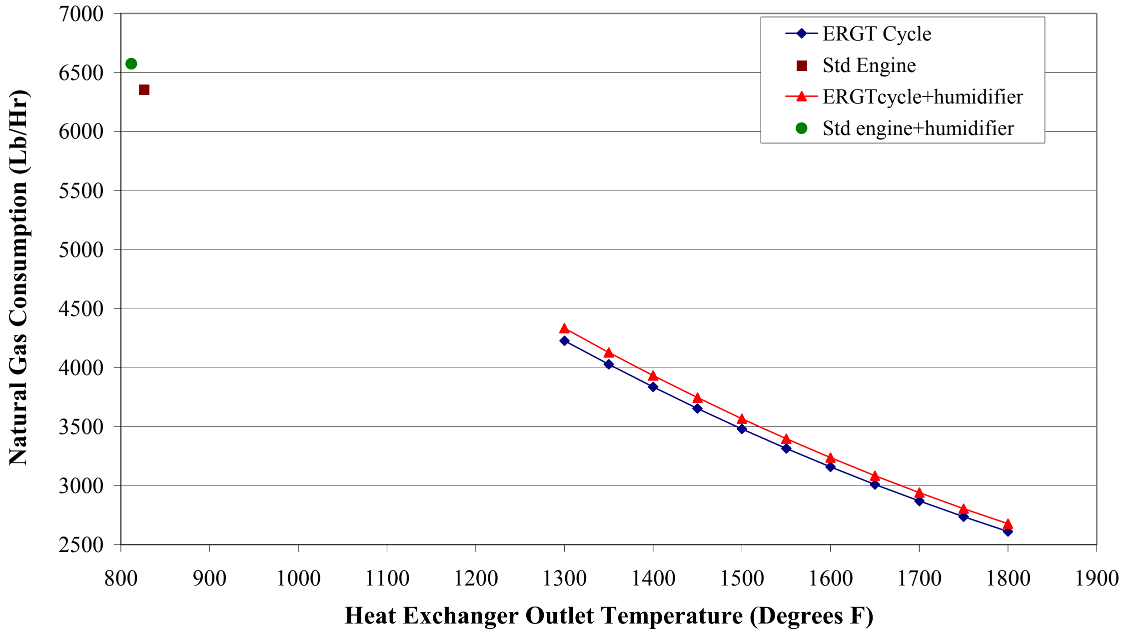

Figure 3.22 Natural Gas Consumption as a Function of HP/HT Air Heater Exit Temperature for Air-cooled Turbine with and without a Humidifier at inlet of Compressor

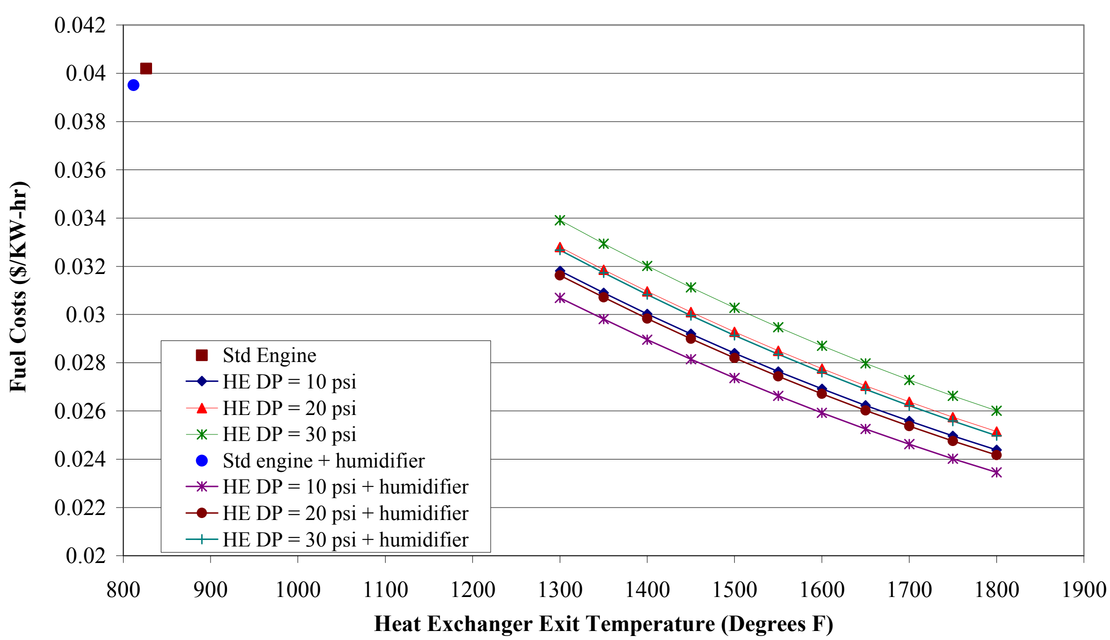

Figure 3.23 Fuel Cost as a Function of HP/HT Air Heater Pressure Drop and HP/HT Air Heater Exit Temperature for Air-Cooled Turbine with and without a Humidifier at Inlet of Compressor

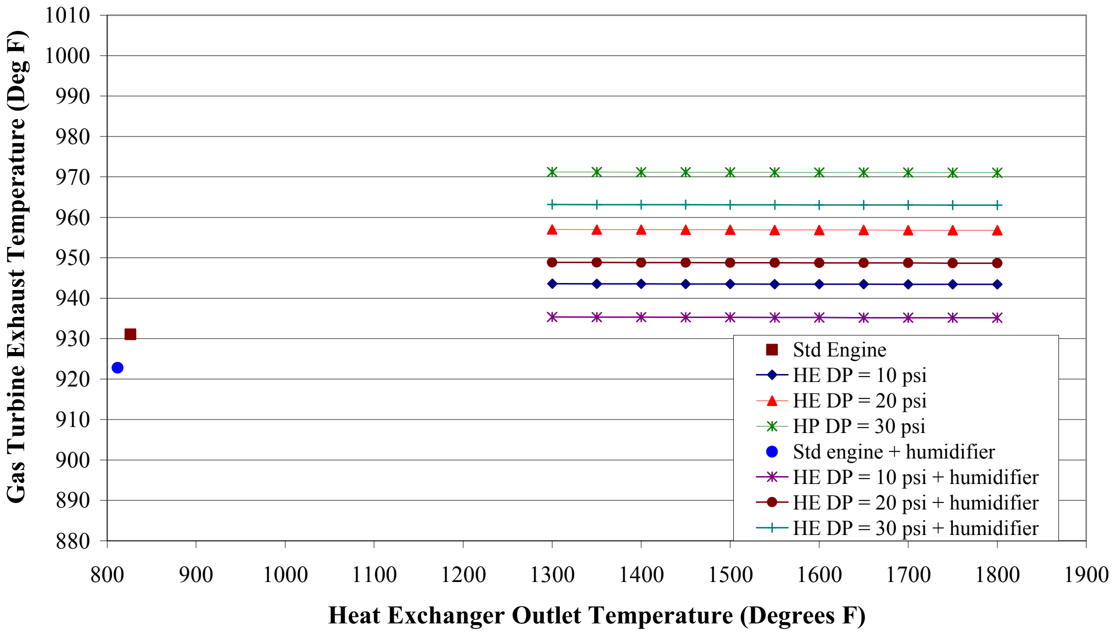

Figure 3.24 Gas Turbine Exhaust Temperature as a Function of HP/HT Air Heater Pressure Drop for Air-Cooled Turbine with and without a Humidifier at Inlet of Compressor

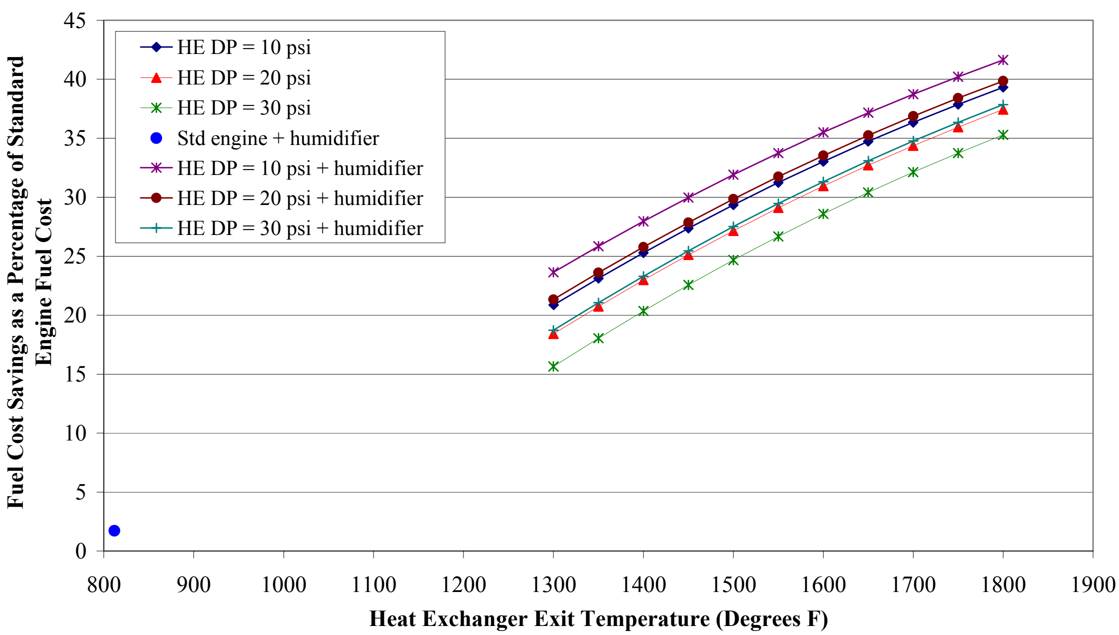

Figure 3.25 Percentage Fuel Cost Savings while Operating an ERGT using the Standard Engine Fuel Costs as a Baseline

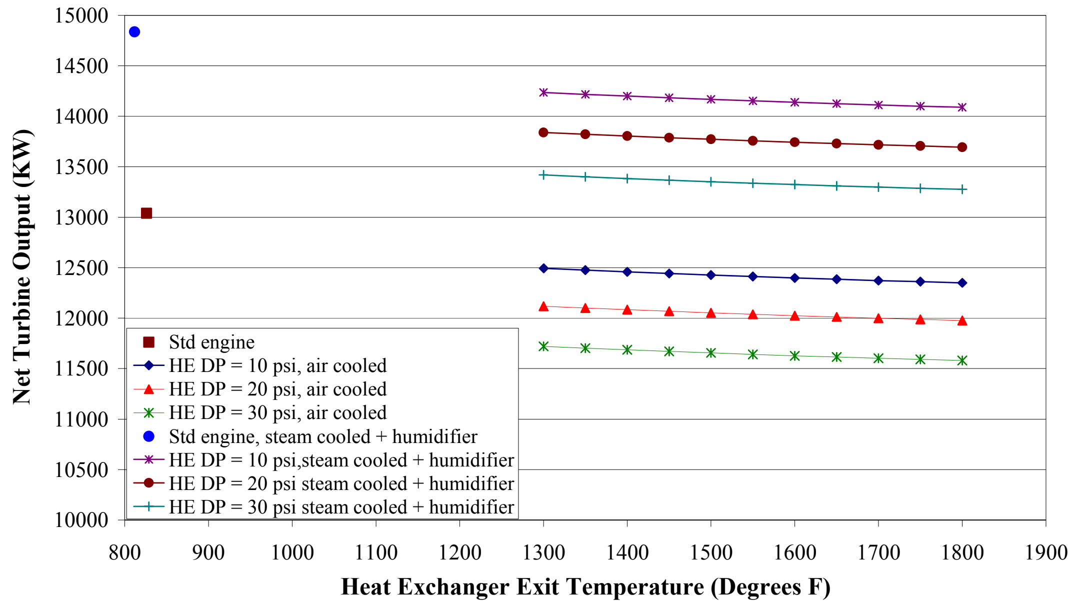

Figure 3.26 Net Turbine Output as a Function of HP/HT Air Heater Pressure Drop and HP/HT Air Heater Exit Temperature for Steam-Cooled Turbine with a Humidifier at Inlet of Compressor Compared to Baseline Cycle using Air-Cooled Turbine without a Humidifier at Inlet of Compressor

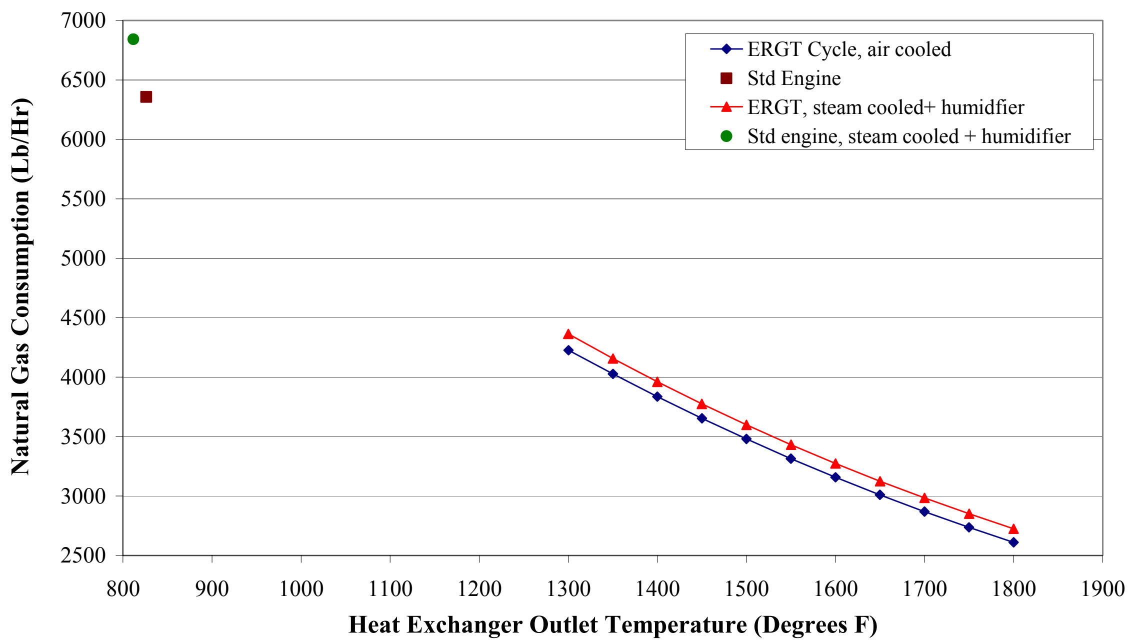

Figure 3.27 Natural Gas Consumption as a Function of HP/HT Air Heater Exit Temperature for Steam-Cooled Turbine with a Humidifier at Inlet of Compressor Compared to Baseline Cycle using Air-Cooled Turbine without a Humidifier at Inlet of Compressor

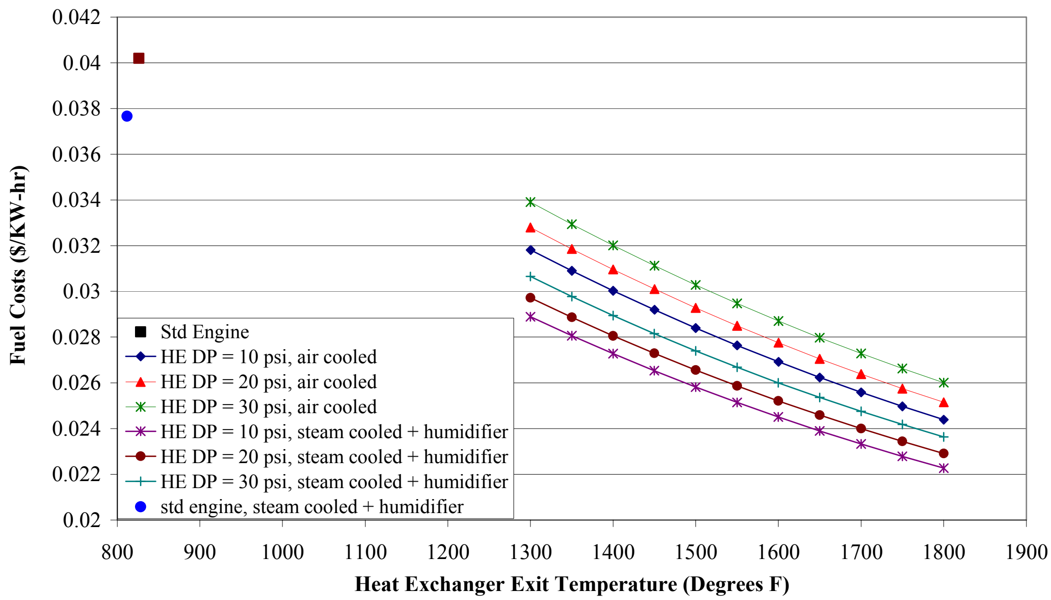

Figure 3.28 Fuel Cost as a Function of HP/HT Air Heater Pressure Drop and HP/HT Air Heater Exit Temperature for Steam-Cooled Turbine with a Humidifier at Inlet of Compressor compared to Baseline Cycle using Air-Cooled Turbine without a Humidifier at Inlet of Compressor

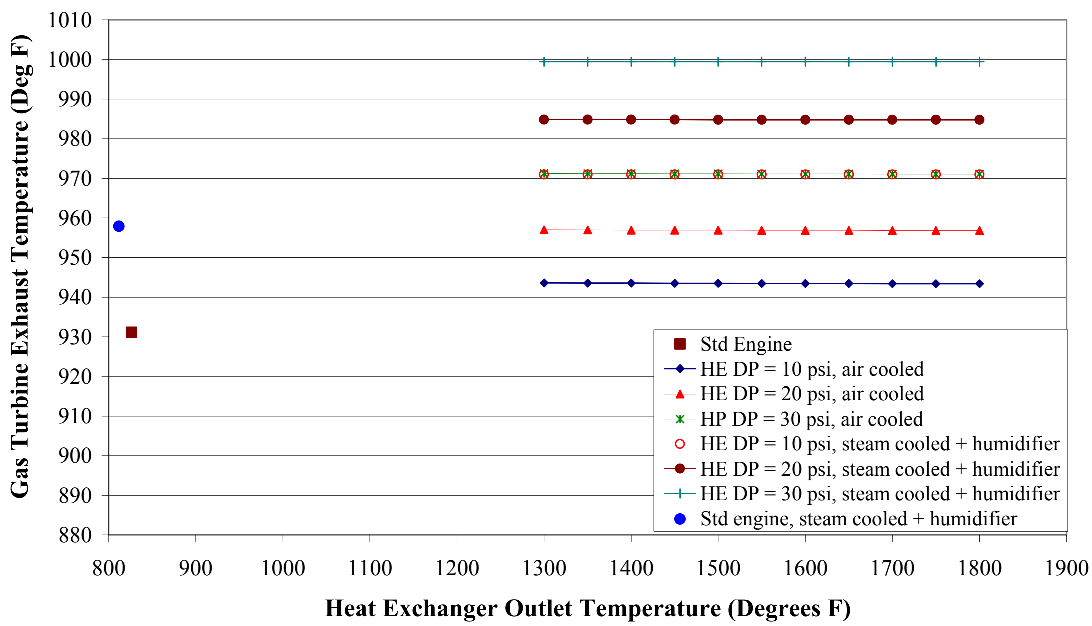

Figure 3.29 Gas Turbine Exhaust Temperature as a Function of HP/HT Air Heater Pressure Drop for Steam-Cooled Turbine with a Humidifier at Inlet of Compressor Compared to Baseline Cycle using Air-Cooled Turbine without a Humidifier at Inlet of Compressor

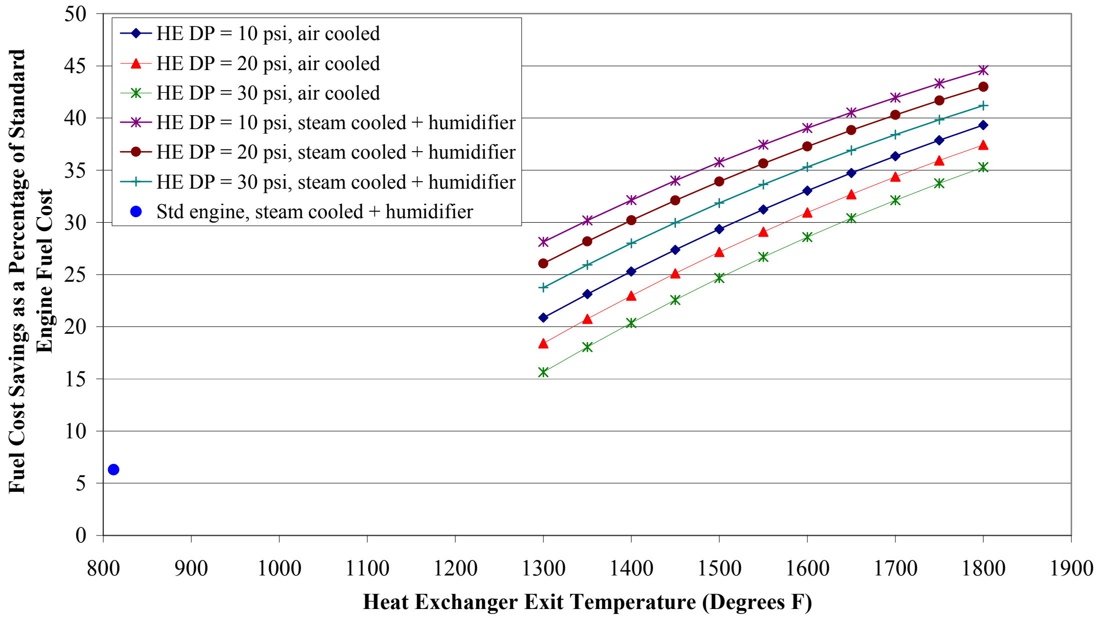

Figure 3.30 Percentage Fuel Cost Savings while Operating an ERGT using the Standard Engine Fuel Costs as a Baseline

The results shown above indicate that installation of the humidifier at the inlet of the ERGT using an air-cooled turbine increases the electrical output by about 5% while the exhaust temperature is decreased by less than 1%. In addition, the installation of the humidifier also increases the savings in fuel cost by about 3%. All these comparisons are made for the same HP/HT heat exchanger pressure drop.

Similar comparisons made for an ERGT using a steam cooled turbine reveal that using a humidifier at the inlet of the ERGT increases the electrical output by 5%, while the exhaust temperature drops by less than 1% and the fuel cost savings is increased by about 3%.

Thus over all, installing a humidifier at the inlet of an ERGT and using steam as the turbine cooling medium increases the electrical output by at least 14% and the fuel cost savings by at least 8% when compared to an ERGT that uses air as a turbine cooling medium and does not have humidification equipment installed at the inlet.

When compared to a standard engine, an ERGT that uses air as a turbine cooling medium and does not have humidification equipment installed at the inlet, saves approximately 18% in fuel cost. A similar comparison between the standard engine and an ERGT that has a humidifier installed at the inlet and uses steam as the turbine cooling medium results in a fuel cost saving of approximately 26%

Based on the above discussion, it can therefore be concluded that thermodynamically an ERGT with a steam-cooled turbine and a humidifier at the compressor inlet is a preferred option.

3.3 Conclusions

The results of the cycle analysis in terms of percentage increase in electrical output and percentage increase in fuel cost savings are summarized in *Figure 3.31* through *Figure 3.36*.

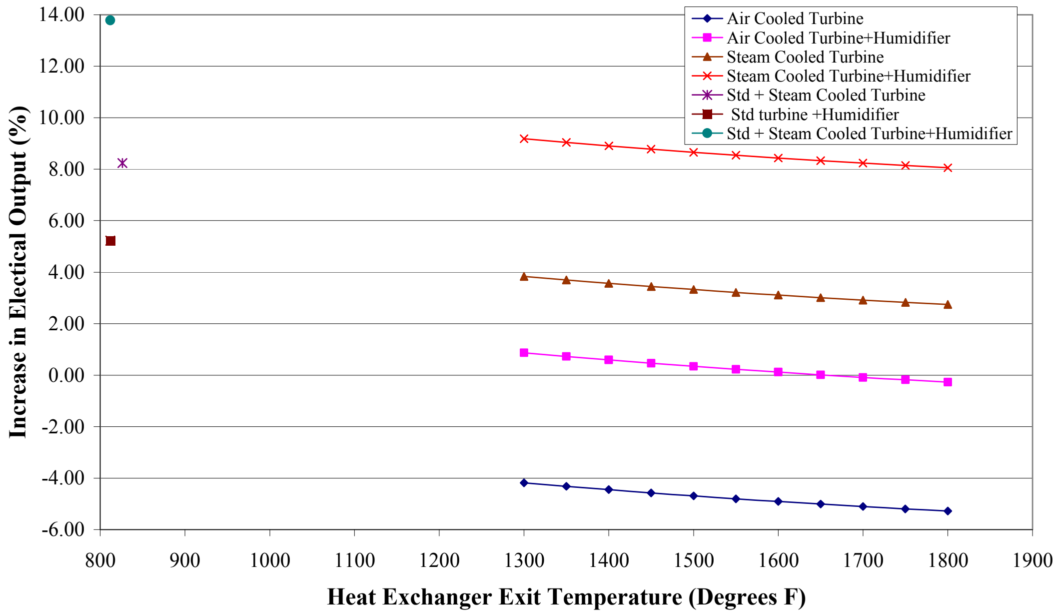

Figure 3.31 Percentage Increase in Electrical Output (with a Standard Engine as the Baseline) for various ERGT Cycles using a HP/HT Heat Exchanger that has a 10psi Pressure Drop

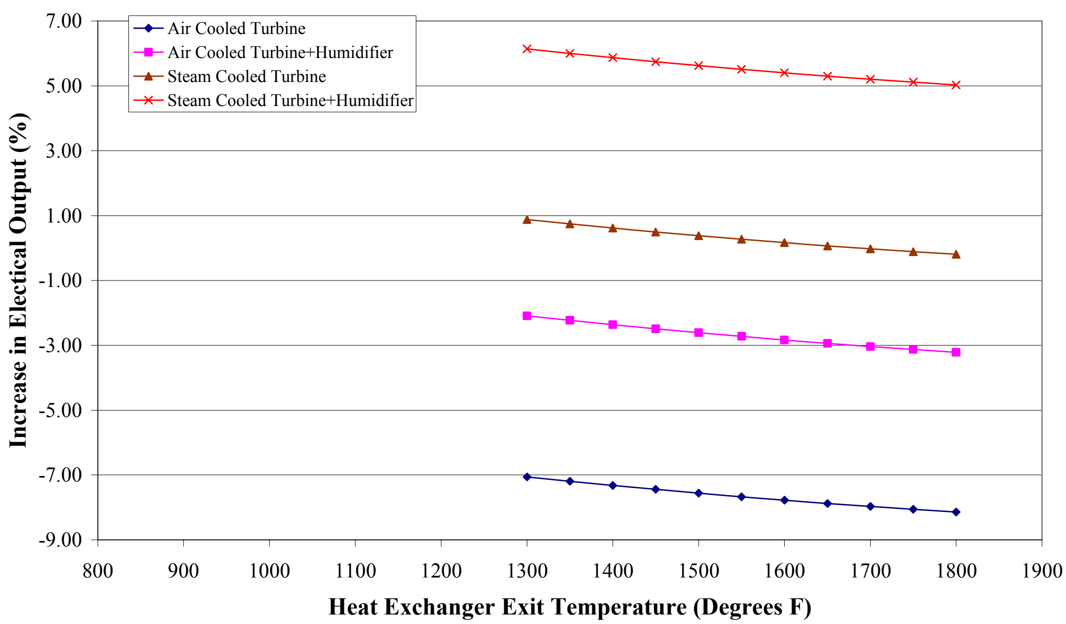

Figure 3.32 Percentage Increase in Electrical Output (with a Standard Engine as the Baseline) for various ERGT Cycles using a HP/HT Heat Exchanger that has a 20psi Pressure Drop

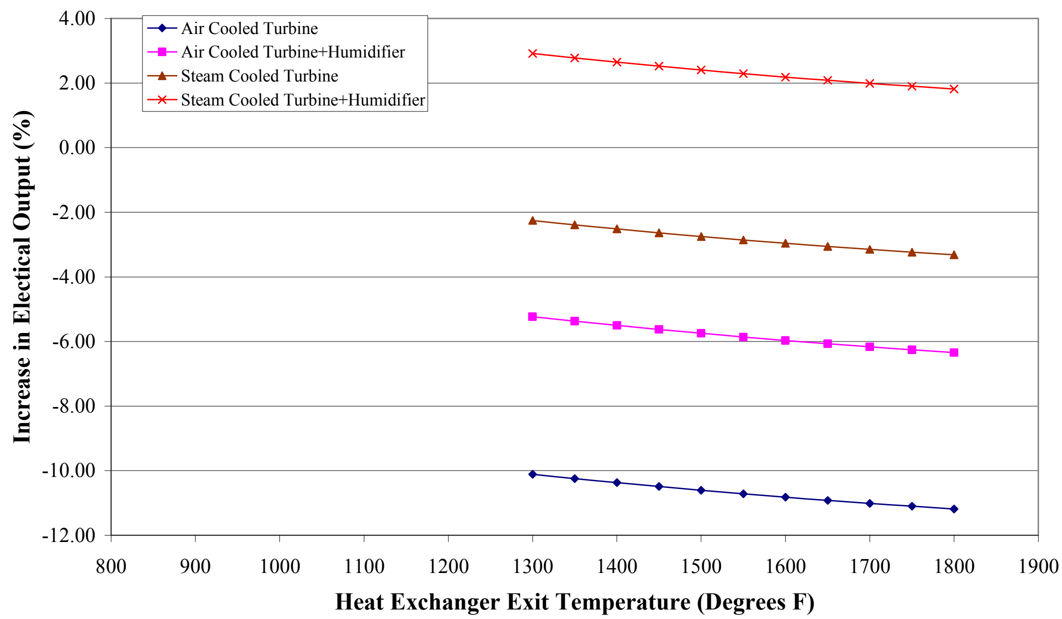

Figure 3.33 Percentage Increase in Electrical Output (with a Standard Engine as the Baseline) for various ERGT Cycles using a HP/HT Heat Exchanger that has a 30psi Pressure Drop

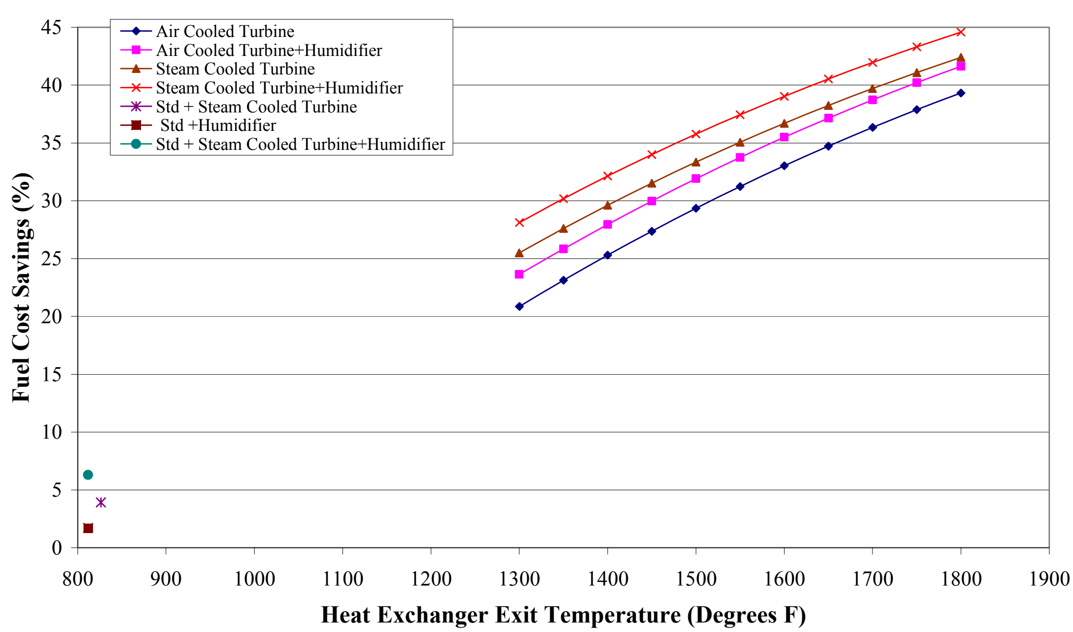

Figure 3.34 Fuel Cost Savings as a Percentage of the Fuel Costs of a Standard Engine for various ERGT Cycles Using a HP/HT Heat Exchanger that has a 10psi Pressure Drop

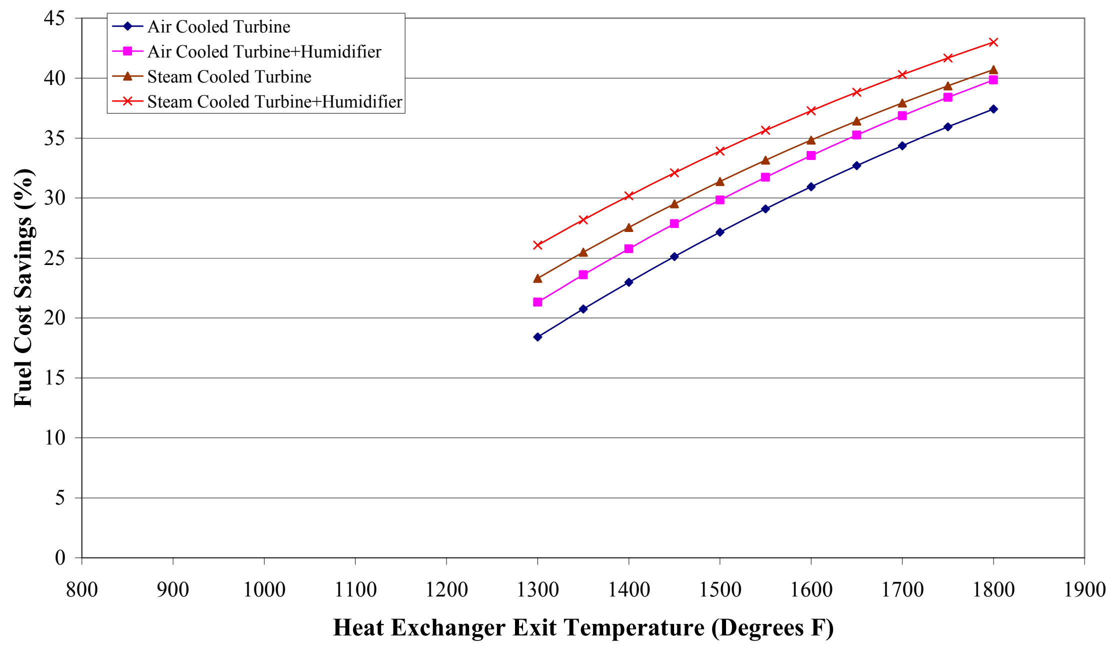

Figure 3.35 *Fuel Cost Savings as a Percentage of the Fuel Costs of a Standard Engine for various ERGT Cycles Using a HP/HT Heat Exchanger that has a 20psi Pressure Drop*

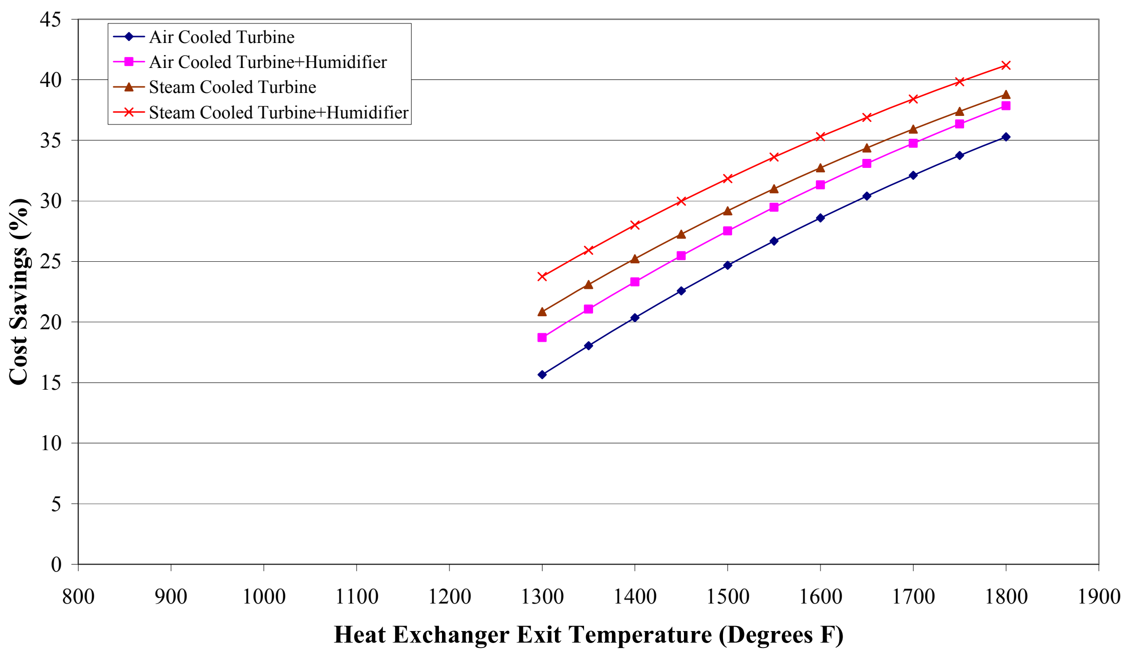

Figure 3.36 *Fuel Cost Savings as a Percentage of the Fuel Costs of a Standard Engine for various ERGT Cycles using a HP/HT Heat Exchanger that has a 30psi Pressure Drop*

Based on the results of the thermodynamic analysis summarized above, it is noted that increasing the HP/HT Heat Exchanger exit temperature and reducing the pressure drop across the heat exchanger will help maximize the fuel cost savings. The reduction in pressure drop also increases the net electrical output, but an increase in the heat exchanger exit temperature causes the power output to reduce slightly. Of the various options considered, the most beneficial in terms of fuel cost savings and the net power output is the ERGT cycle that uses steam as the cooling medium for its turbine and has a humidifier installed at the inlet. However, there are physical design constraints for the present gas turbine design that will limit the extent of savings that can be derived from this ERGT cycle. Potential limits for various ERGT cycles are discussed in *Section 3.5.*

3.4 Arrangement Drawings, PFD and PI&D for Recommended Gas Turbine System

3.4.1 Titan 130 Package

Solar has provided information regarding the package layout and installation and commissioning. This includes the arrangement and layout drawings of a standard T130 package (*Figure 3.37* through *Figure 3.40*), and information regarding the various pumps, compression systems and cooling systems used on a standard Titan 130 package. It should be noted that all data provided relate to Solar's present Titan 130 package and do not reflect modifications engineered for the ERGT program.

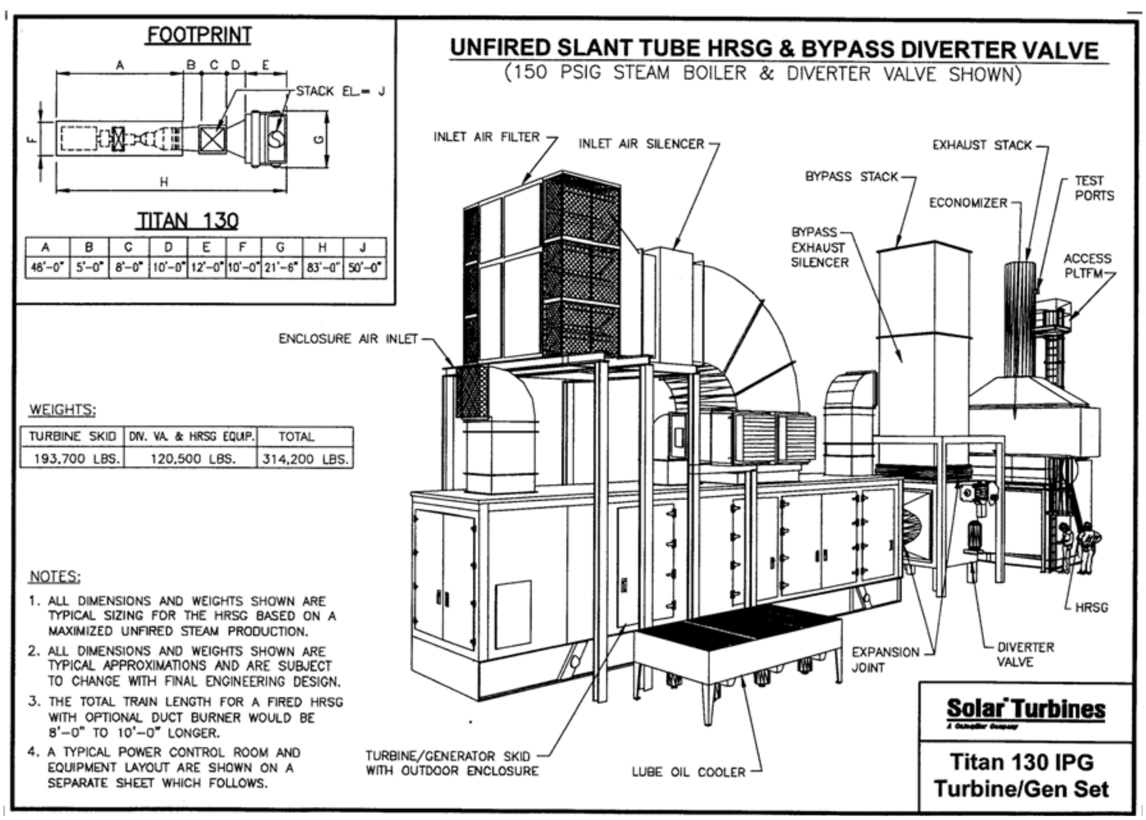

Figure 3.37 Titan 130 IPG Turbine/Gen Set

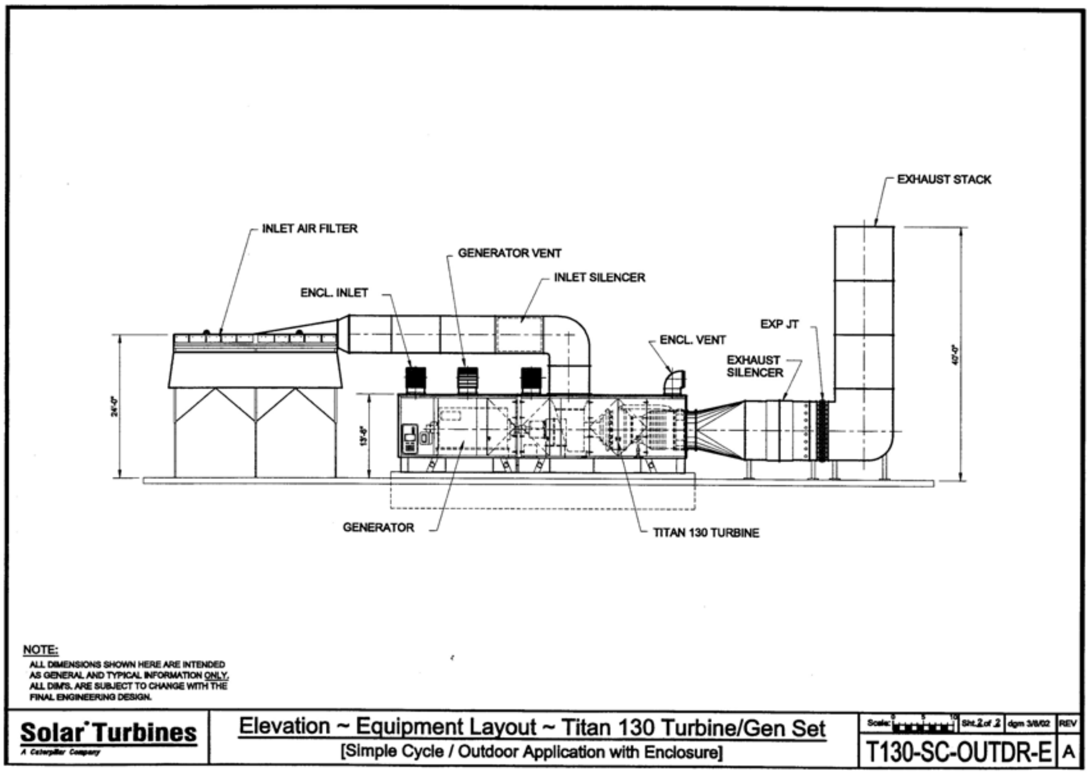

Figure 3.38 Equipment Layout – Titan 130 Turbine /Gen Set

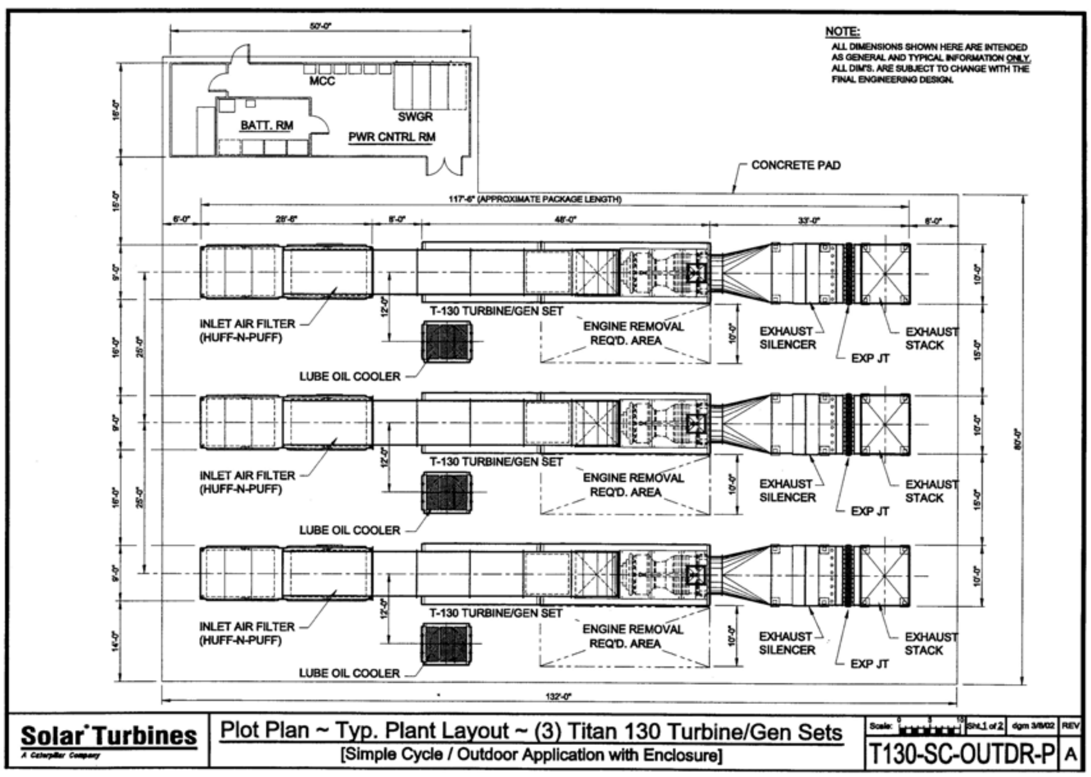

Figure 3.39 Typical plant Layout of Three Titan 130 Turbine/Gen Set

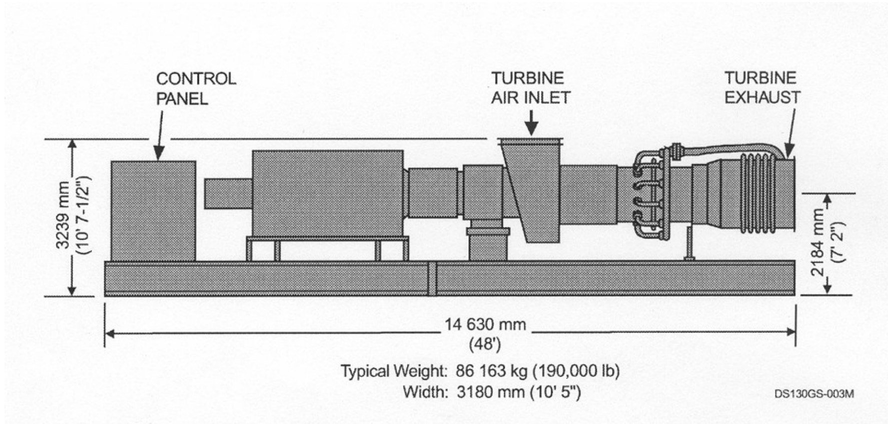

Figure 3.40 Titan 130 Package Dimensions

To minimize the development cost of an ERGT based on one of the cycles discussed in *Section 3.1*, it is recommended that an ERGT be developed by modifying an existing gas turbine. Minimum modifications would include modifying the combustor and providing a passage for the airflow to and from the HP/HT heat exchanger. The possible candidates for such a modification are:

i. A gas turbine that has a side mounted can combustor as shown in *Figure 3.41*
ii. A gas turbine that has a silo combustor, similar to the one shown in *Figure 3.42*
iii. A gas turbine that has been designed to run on a recuperated cycle, similar to the one shown in *Figure 3.43*

Each of these three configurations can accommodate airflows to and from the HP/HT heat exchanger.

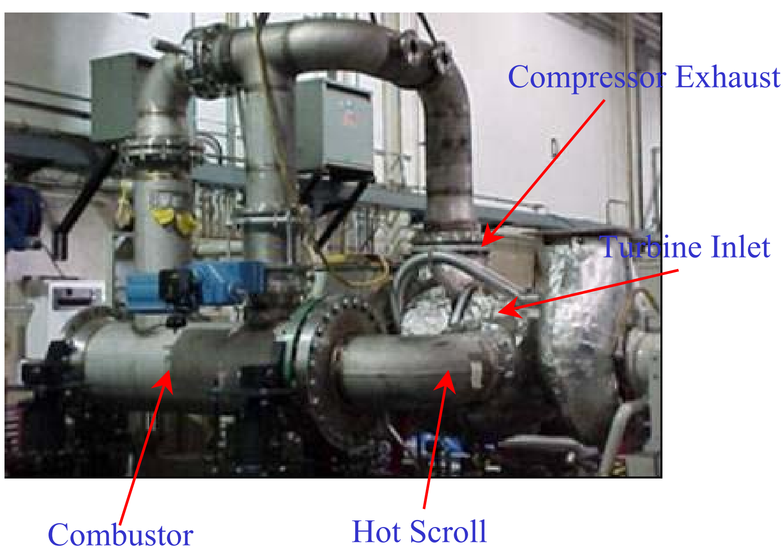

Figure 3.41 Gas Turbine with a Side Mounted Can Configuration

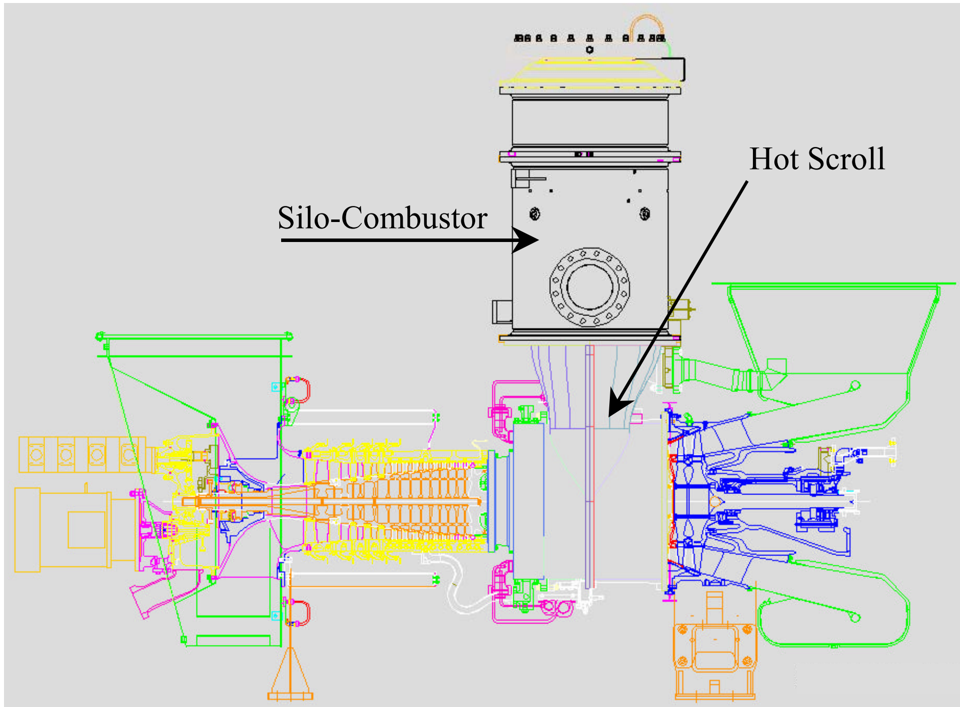

Figure 3.42 A Gas Turbine with a Silo-Combustor and a Hot Scroll

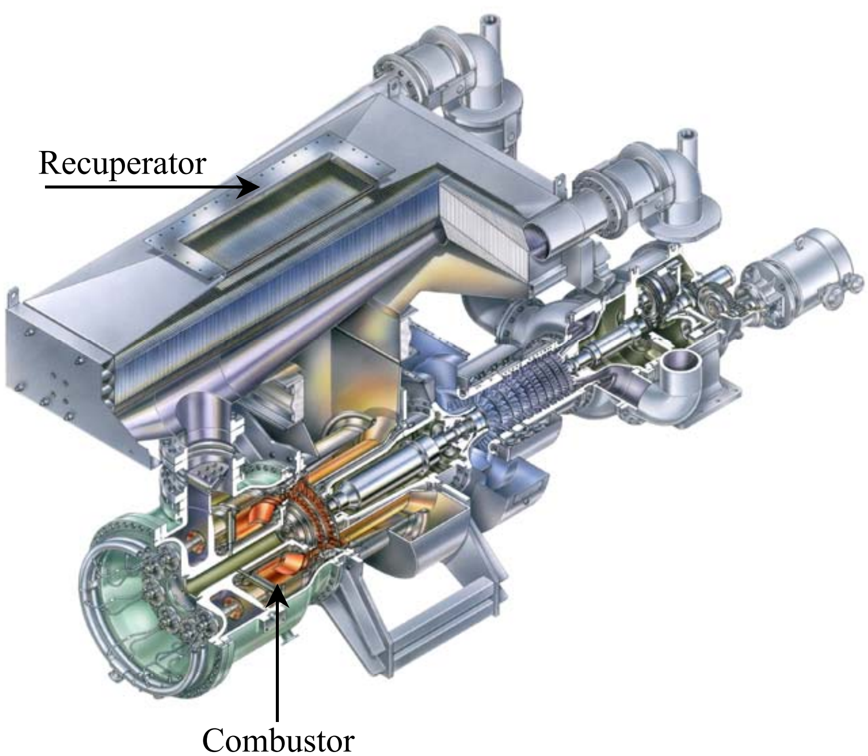

Figure 3.43 A Recuperated Gas Turbine

The production Titan 130 gas turbine that Solar currently offers does not reflect any of the above geometries and is unsuitable for the ERGT application. The effort required to modify this gas turbine to the preferred ERGT cycle is equivalent to developing a new gas turbine. Further, given the present heat exchanger exit temperature limit of 1450°F (see *Section 3.5*), the developmental cost will more than offset any fuel cost savings unless there is a significant market for the ERGT.

3.5 Equipment Modifications Design Study

Within the program an assessment was made of the technologies reflected in the ERGT cycle to define near-term improvements. The conclusions drawn from these assessments are presented here.

3.5.1 Combustor

For production engines at Solar combustor inlet temperatures range from approximately 690ºF to 1250ºF. Present materials and premixed combustion technology limitations hinder the design of gas turbine combustors with inlet temperatures greater than 1450°F. Further, the design effort required to modify present gas turbines to accommodate combustor inlet temperatures of 1450°F will be considerable.

3.5.2 HP/HT Air Heater

The present study has assumed that the HP/HT air heater will be operated at a pressure drop of 30 psi and an exit air temperature of 1400°F. With future materials advancement the exit temperature most likely can be increased to 1500°F. However, the high-pressure drop (30 psi) is incompatible with most small to medium industrial gas turbines. A high-pressure drop will increase the risk of compressor surge. To overcome this hurdle either

i. A new compressor and turbine need to be designed and developed,

Or

ii. The heat exchanger pressure drop has to be reduced below 15 psi.

Reduction of the heat exchanger pressure drop poses technical, cost and practical difficulties as such a reduction would require increased surface area and volume of the air heater (while there is normally a limited space within the furnaces to fit an air heater). Further, this reduction in pressure drop will adversely affect the convective heat transfer coefficient, causing the heat exchanger temperatures to rise. Higher metal temperatures may reduce the life and operational safety of the heat exchanger.

3.5.3 Steam Cooled Turbine

Although the ERGT using a steam cooled turbine looks attractive, developing such a turbine cooling system is prohibitively expensive for Solar. This issue will effect most smaller gas turbines where first cost is a critical buying criterion. If such a turbine was developed its capital cost will be substantially higher than current air-cooled gas turbines due to the complexities of the cooling circuits and control systems. It is expected that the high development and manufacturing cost of such a system will nullify the fuel cost savings seen in the cycle analysis, especially as the combustor and heat exchanger designs limit the combustor inlet temperature to 1450°F.

3.6 Market Potential Estimate

After evaluating the present technologies and the cycle requirements for ERGT in the forest products industry (FPI) applications, it was concluded that on a long-term basis, the FPI could be a potential market. However, to realize this potential the present technologies related to critical components of the cycle including high-pressure air heaters (with low pressure-drop) need significant development. Further materials for gas turbine components such as boost combustor, scroll, etc., need to be developed to effectively sustain air temperatures of up to 1800ºF without any significant rise in component cost from present levels.

It was also concluded that Solar's present range of products is unsuitable both in size and configuration for the FPI application. However, this conclusion may warrant re-evaluation if in the future Solar adds to its product family an engine configuration that is more adaptable to the FPI application.

4 Conclusions and Recommendations

Based on the results of the thermodynamic analysis, it is concluded that increasing the HP/HT heat exchanger exit temperature and reducing the pressure drop across the heat exchanger will help maximize the fuel cost savings. The reduction in pressure drop also increases the net electrical output, but increase in heat exchanger exit temperature causes the power output to reduce slightly and raises material durability issues for the ERGT engine casings. Of the various options considered, the most beneficial one in terms of fuel cost savings and net power output is the ERGT cycle that uses steam as the cooling medium for its turbine and has a humidifier installed at its inlet. However, in view of the major redesign effort required for Solar to develop a steam-cooled turbine, it is recommended that presently available air-cooled turbines be used with an inlet fogger/humidifier (when the ambient conditions warrant its use).

Based on the analysis of the present physical design constraints on the various components of an ERGT, it is recommended that an ERGT be developed by modifying existing gas turbines to accommodate a heat exchanger outlet temperature (combustor inlet temperature) of up to 1450°F. Efforts need to be made to develop an HP/HT air heater that works with a maximum pressure drop of 15 psi so as to minimize changes to present gas turbine designs and keep development costs low.

From a long-term perspective, the ERGT cycle potential can be realized in the coming years if significant improvements can be made to the present technologies related to critical components of the cycle including high-pressure air heaters (having a low pressure-drop). Further materials for gas turbine components such as boost combustor, scroll, etc., need to be developed to effectively sustain air temperatures of up to 1800ºF without any significant rise in component cost from present levels.