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Thermoeconomic Design Optimization of a KRW-Based IGCC Power Plant

Final Report

G. Tsatsaronis
L. Lin
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November 1991

Work Performed Under Contract No.: DE-FC21-89MC26019

For
U.S. Department of Energy
Office of Fossil Energy
Morgantown Energy Technology Center
Morgantown, West Virginia

By
Southern Company Services, Inc.
Research and Environmental Affairs
Birmingham, Alabama
and
Tennessee Technological University
Center for Electric Power
Cookeville, Tennessee

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P.O. Box 880
Morgantown, West Virginia 26507-0880

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November 1991

ABSTRACT

This report discusses the cost and efficiency optimization of an integrated gasification-combined-cycle (IGCC) power plant design and the effects of important design options and parameters. Advanced thermoeconomic techniques were used to evaluate and optimize a given IGCC concept which uses Illinois No. 6 bituminous coal, air-blown KRW coal gasifiers, a hot gas cleanup system, and GE MS7001F gas turbines.

The study was sponsored by the U.S. Department of Energy Morgantown Energy Technology Center and was conducted by the Tennessee Technological University Center for Electric Power. Southern Company Services, Inc. managed the project. Other participants were The M. W. Kellogg Company and General Electric Company.

Three optimal design concepts are presented and discussed in the report. Two of the concepts are characterized by minimum cost of electricity at two different values of the steam high pressure. The third concept represents the thermodynamic optimum. This study identified several differences between the original design and the design of the optimized cases. Compared with the original concept, significant annual savings are achieved in the cost optimal cases.

Comparisons were made between results obtained using both the old and the new performance data for the MS7001F gas turbine. This report discusses the effects of gasification temperature, steam high pressure, coal moisture, and various design options on the overall plant efficiency and cost of electricity. Cost sensitivity studies were conducted and recommendations for future studies were made.

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NOMENCLATURE

Symbol	Meaning [units]
c	cost per exergy unit [$/GJ or $/MMBtu]
d	relative cost difference between average cost per exergy unit of product and average cost per exergy unit of fuel
D	cost of exergy destruction [$]
$\dot{E}$	exergy flow rate [MW or MMBtu/hr]
f	thermoeconomic factor denoting the contribution of the capital costs, Z, to the relative cost difference, d, between fuel and product in a plant component
F	variable expressing the criterion of thermoeconomic similarity, Equations 3-12 and 3-17
g	constant in Equations 3-10 and 3-13 expressing the dependence of total net investment for a plant component on the efficiency and capacity of the component
$\dot{H}$	enthalpy flow rate [MW or MMBtu/hr]
Δh^0	higher heating value [kJ/kg or Btu/lbm]
I	investment cost [$]
m	mass [kg or lbm]
n	exponent in cost Equations 3-10 and 3-13
P	pressure [bar or psia]
Q	heat [MJ or MBtu]

Symbol	Meaning [units]
r	ratio of capital costs to product exergy in a heat exchanger [$/MJ or $/MBtu]
$\dot{S}$	entropy flow rate [kW/K or MBtu/hr·R]
T	temperature [°C or °F]
$\dot{W}$	power [MW]
x	mole fraction [%]
Z	capital costs associated with a plant component and considered in cost equations [$]
Greek letters:	
γ	coefficient expressing the part of the fixed operating and maintenance costs which depends on the net investment expenditure for a plant component [%]
ϵ	capital recovery factor [%]
ζ	exergetic efficiency [%]
η	thermal efficiency [%]
θ	exergy destruction ratio [%]
τ	annual number of hours of plant operation at the nominal capacity [hr]
Subscripts:	
D	exergy destruction
F	fuel (according to the definition of exergetic efficiency)

<u>Symbol</u>	<u>Meaning</u> [units]
i	stream
in	input
j	output stream
k	plant component
L	exergy losses
LOSS	heat losses
P	product (according to the definition of exergetic efficiency)
tot	total plant
Superscripts:	
OPT	optimum
•	time rate of the corresponding variable

LIST OF ACRONYMS AND ABBREVIATIONS

ACRS	Accelerated Cost Recovery System
AFDC	Allowance for Funds During Construction
ASPEN	Computer Simulation Software
BFW	Boiler Feedwater
BLDWN	Blowdown
CM	Coal Moisture at the Gasifier Inlet
COE	Cost of Electricity
CT	Combustion Turbine
CTG	Combustion Turbine Generator
DOE	U.S. Department of Energy
DOE/METC	U.S. Department of Energy/Morgantown Energy Technology Center
EPRI	Electric Power Research Institute
EXD	Exergy Destruction
FGD	Flue Gas Desulfurization
FWH	Feedwater Heater
GE	General Electric Company
GI	Gasification Island
GPC	Georgia Power Company
GSC	Gland Seal Condenser
GT	Gasification Temperature
HHV	Higher Heating Value
HP	High Pressure
HRSD	Heat Recovery Steam Drum
HRSG	Heat Recovery Steam Generator
IGCC	Integrated Gasification Combined Cycle
IP	Intermediate Pressure

LIST OF ACRONYMS AND ABBREVIATIONS (Continued)

KRW	KRW Energy Systems
LASH	Limestone-Containing Ash
LHV	Lower Heating Value
LP	Low Pressure
MWK	The M. W. Kellogg Company
O&M	Operation and Maintenance
PC	Pulverized-Coal
PH	Preheater
PI	Power Island
PPC	Process Plant Cost
PROMOD	Computer Simulation Model
RAM	Reliability Availability Maintainability
RGC	Recycle Gas Cooler
SCS	Southern Company Services, Inc.
SHP	Steam High Pressure
SN	Stream Number
SSR	Steam Seal Regulator
STG	Steam Turbine-Generator
TAG	Technology Assessment Guide (EPRI)
TCR	Total Capital Requirement
THESIS	Thermodynamic and Economic Simulation System (computer software)
TPC	Total Plant Cost
TPI	Total Plant Investment
TTU	Tennessee Technological University
UNIRAM	Software Application for Reliability, Availability, and Maintainability

LIST OF CASES MENTIONED IN THIS REPORT

Case	Explanation
1	Original Case 1 (see section 2.3). This case is identical with Case 1 in Reference [1] (GT=1900°F; CM=4.98%; SHP=1600 psia)
1A	Same design configuration as in the original Case 1. (GT=1800°F; CM=4.98%; SHP=1600 psia)
1B	Same design configuration as in the original Case 1. (GT=2000°F; CM=4.98%; SHP=1600 psia)
1C	Same design configuration in the gasification island as in the cost optimal Cases 1CO1 and 1CO2. (GT=1800°F; CM=11.12%; SHP=1815 psia)
1CO1	Cost Optimal Case 1 (see section 4.1) at a relatively high steam pressure. (GT=1920°F; CM=11.12%; SHP=2055 psia)
1CO2	Cost Optimal Case 1 (see section 4.2) at a steam pressure value close to the original Case 1 value. Same design configuration in the gasification island as in the Case 1CO1. (GT=1920°F; CM=11.12%; SHP=1515 psia)
1TO1	Thermodynamically optimal case 1 (see section 4.3). (GT=1800°F; CM=11.12%; SHP=2200 psia)

Abbreviations

GT	Gasification temperature
CM	Coal moisture at the gasifier inlet (weight percent)
SHP	Steam high pressure

UNITS CONVERSION FACTORS

Temperature	$T[^\circ F] = \frac{9}{5} T[^\circ C] + 32$
Pressure	1 bar = 14.504 psia 1 bar = 29.530 inches Hg 1 bar = 0.1 MPa
Mass	1 kg = 2.2046 lbm
Mass flow rate	1 kg/s = 7936.6 lbm/hr
Energy (or Exergy)	1 kWh = 3.6 MJ 1 MJ = 10^6J = 0.9478 MBtu 1 GJ = 10^9J = 0.9478 MMBtu
Specific energy (or exergy)	$1 \frac{\text{MJ}}{\text{kg}} = 0.4299 \frac{\text{MBtu}}{\text{lbm}}$
Power (energy flow rate)	$1\ \text{MW} = 10^6\text{W} = 3.412 \frac{\text{MMBtu}}{\text{hr}}$
Entropy flow rate	$1 \frac{\text{MW}}{\text{K}} = 1.8956 \frac{\text{MMBtu}}{\text{hr}\cdot{}^\circ\text{F}}$
Cost per exergy unit	$1 \frac{\$}{\text{GJ}} = 1.0551 \frac{\$}{\text{MMBtu}}$

Meaning of Prefixes

SI System of Units

G = 10^9; k = 10^3; M = 10^6

English System of Units

M = 10^3; MM = 10^6

EXECUTIVE SUMMARY

Several integrated gasification-combined-cycle (IGCC) power plants are currently under development to provide a clean, efficient and cost effective option for generating electric power from coal. Through a joint site-specific project with the U.S. Department of Energy (DOE) Morgantown Energy Technology Center (METC), Southern Company Services, Inc. (SCS) conducted a comprehensive study to determine the characteristics of IGCC power plants with respect to thermal and environmental performance, capital costs and electricity costs. The results of that study are summarized in Reference [1].

Among the power plant configurations developed and compared in that study, the so-called Case 1 design configuration was found to be an attractive IGCC power plant configuration. This plant uses four air-blown KRW coal gasifiers, a hot gas cleanup system, and two GE MS7001F gas turbines. However, no attempt was made to minimize the cost of electricity (COE) in the results presented in Reference [1]. Indeed, final conclusions from the comparison of different IGCC design configurations should only be drawn after these configurations have been optimized from the economic viewpoint, or at least after the potential for further cost and efficiency improvements has been estimated.

To optimize the design of Case 1 using advanced thermoeconomic optimization techniques, SCS entered into a supplementary project with DOE/METC. The simulation and optimization studies reported here were conducted by the Tennessee Technological University (TTU) Center for Electric Power under a subcontract to SCS. SCS was responsible for overall project management and cost estimates in the power island. The M. W. Kellogg Company provided TTU with material and energy balances in addition to cost estimates for the gasification island. Some other companies (General Electric, Industrial Filter and Pump Manufacturing, and the Henry Vogt Machine Company) provided performance and cost data for specific equipment items.

The major objectives of this supplementary project were to:

- Study the effect of various design options on the plant efficiency and the cost of electricity generated by an IGCC power plant using Illinois No. 6 bituminous coal, air-blown KRW coal gasifiers, a hot gas cleanup system, and industrial gas turbines.

- Develop an IGCC power plant concept characterized by minimum cost of electricity.

Several design concepts were developed and evaluated on the same basis as those in Reference [1]. For each case, a process design was developed; capital costs, O&M costs and cost of electricity were estimated; detailed thermoeconomic analyses were conducted; and suggestions for improving the cost effectiveness were developed. In addition to the original Case 1, three optimal cases were selected for presentation and discussion in this report. Two of the cases (1CO1 and 1CO2) are cost optimal cases while the third case (1TO1) represents the thermodynamic optimum. Figures 1 and 2 show the simplified flow diagram of Case 1CO1 as an example of the design configurations optimized in the present study.

The major differences among the cases discussed here refer to the

- gasification temperature
- coal moisture at the gasifier inlet
- steam high pressure
- use of the heat of the flue gas from the sulfation area
- design of the heat recovery steam generator, product gas cooler, recycle gas cooler, exit gas cooler, and feedwater preheaters
- design of the HP section of the steam turbine, and
- steam extractions from the steam turbine.

Table 1 compares the primary features and performance of the cases addressed in this study. The cost optimal cases 1CO1 and 1CO2 have identical configurations of the gasification island. The major difference between these cases is in the value of the steam high pressure which is 2055 psia in Case 1CO1 and 1515 psia in Case 1CO2. Case 1TO1 represents the thermodynamically optimal case and demonstrates the potential for improving the overall plant efficiency.

The differences in the cold gas efficiency of the gasification island in Table 1 can be explained by the differences in the gasification temperature and the coal moisture at the gasifier inlet. The sulfur removal in the gasifier is 91.4 percent for the three optimized cases and 86.5 percent for the original Case 1. The sulfur removal in the gasification island is the same, 99.4 percent, for all four cases.

The net power output varies between 458.4 MW (original Case 1) and 482.5 MW (Case 1CO1). As expected, Case 1TO1 possesses the lowest net plant heat rate (8,181 Btu/kWh). Case 1CO1 is the next lowest (8,351 Btu/kWh), while the original Case 1 has the highest net plant heat rate (8,595 Btu/kWh). The efficiency difference between Cases 1CO1 and 1CO2 is mainly caused by the difference in the steam high pressure value.

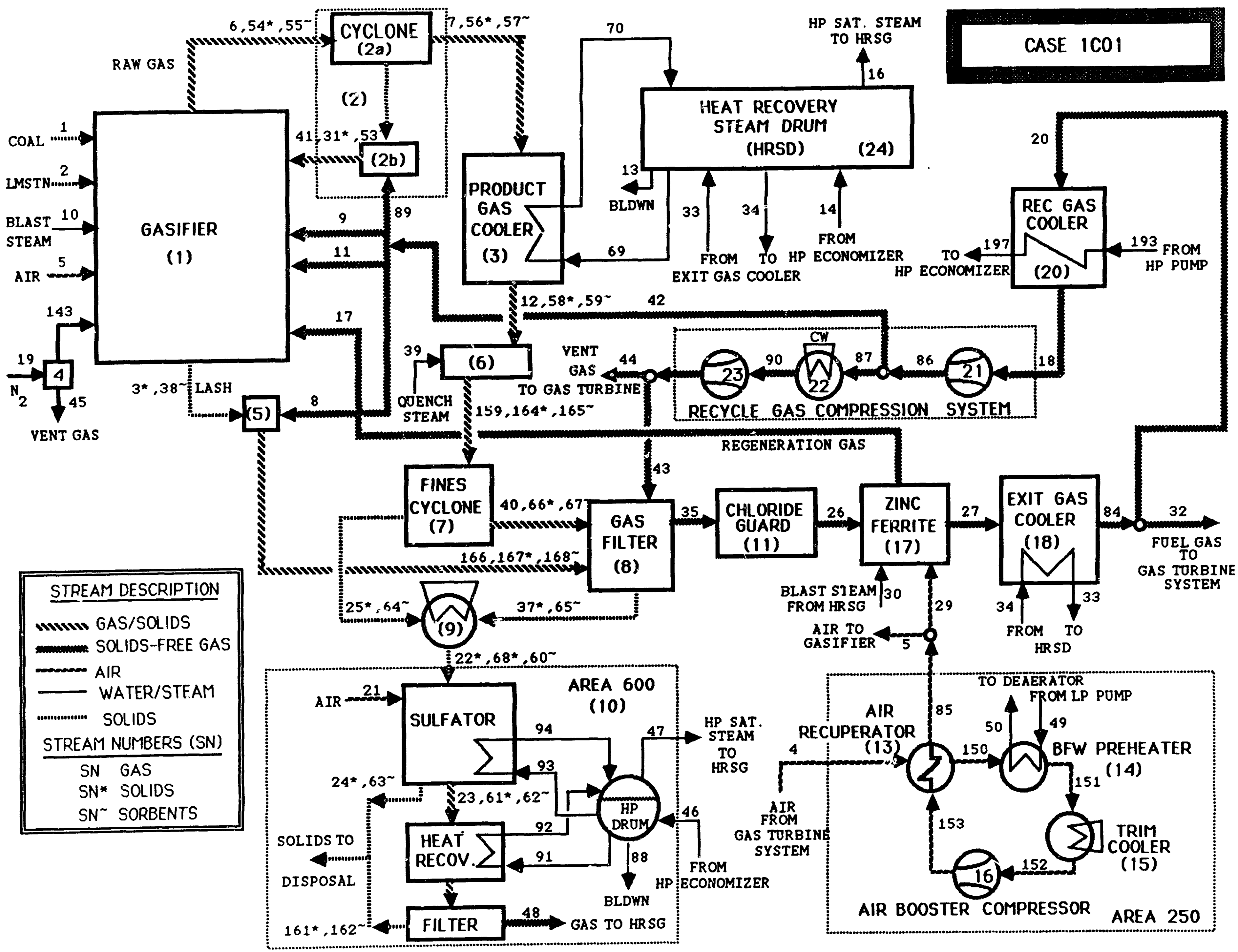

Figure 1. Flow Diagram of the Gasification Island in Case 1C01

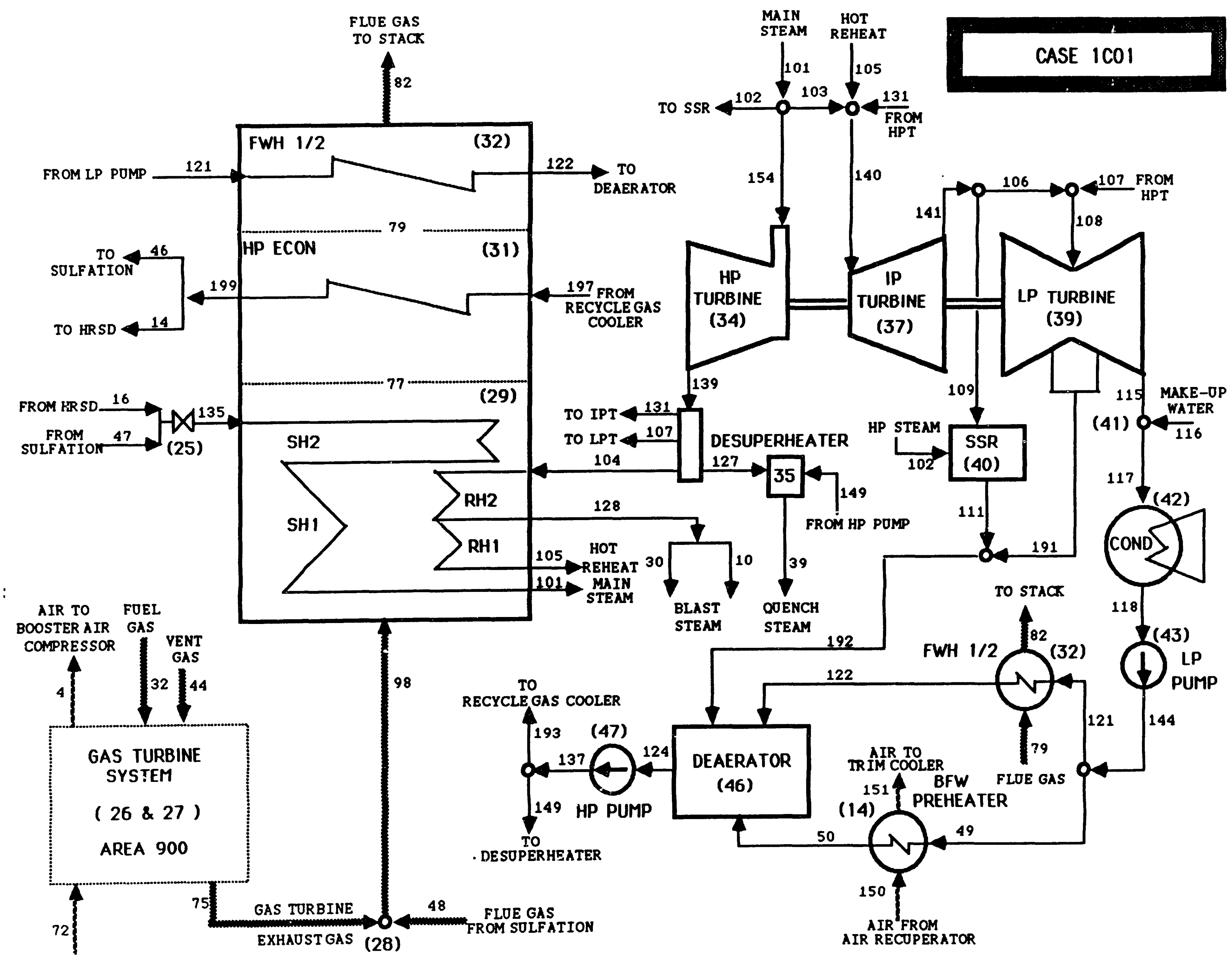

Figure 2. Flow Diagram of the Power Island in Case 1C01

TABLE 1

Summary of Gasification Power Plant Performance

Case	1	1CO1	1CO2	1TO1
Gasification Temperature [°F]	1900	1920	1920	1800
Coal Moisture [wt%]	4.98	11.12	11.12	11.12
Steam High Pressure [psia]	1600	2055	1515	2200
<u>Gasifier Feeds:</u>				
Coal (t/day) (as prepared)	3,792	4,146	4,146	4,035
Limestone (t/day)	1,053	1,077	1,077	1,048
Air (t/day)	12,888	13,682	13,682	12,760
Steam (MMlbm/day)	10,714	10,500	10,500	10,336
<u>Gasification Island Products:</u>				
Fuel Gas (LHV, MMBtu/day)	63,336	63,300	63,300	63,300
Steam to Power Island (Mlbm/day)	24,894	38,436	32,426	37,414
Solid Waste (t/day)	1,231	1,259	1,259	1,226
<u>Gasification Island Performance:</u>				
Cold Gas Efficiency - LHV (%)	69.7	68.0	68.0	69.9
Carbon Conversion (%)				
Gasifier Only	96.5	96.5	96.5	96.5
Gasification Island	99.9	99.9	99.9	99.9
Sulfur Removal (%)				
Gasifier Only	86.5	91.4	91.4	91.4
Gasification Island	99.4	99.4	99.4	99.4
<u>Overall Plant Performance:</u>				
Coal Energy Input (MMBtu/hr)	3,940	4,030	4,030	3,922
Gross Power Output (MW)				
Combustion Turbine(s)	311.6	311.6	311.6	311.6
Steam Turbine(s)	182.2	210.0	201.6	205.5
Total Power	493.8	521.6	513.1	517.1
Station Service (MW)	35.4	39.0	37.7	37.8
Net Power Output (MW)	458.4	482.5	475.4	479.3
Net Heat Rate - HHV (Btu/kWh)	8595	8351	8475	8181
Thermal Efficiency - HHV (%)	39.7	40.9	40.2	41.7

TABLE 2

Summary of Gasification Power Plant Costs

Case	1		1CO1		1CO2		1TO1	
	$MM	$/kW	$MM	$/kW	$MM	$/kW	$MM	$/kW
Net Power Output, MW	458.4		482.5		475.5		479.3	
CAPITAL COST								
Plant Construction Cost	378	825	402	833	392	825	423	882
Process Contingency	20	43	20	41	19	41	21	44
Project Contingency	80	174	84	175	82	173	89	185
Total Plant Cost	477	1042	506	1049	494	1039	533	1111
AFDC	94	204	99	206	97	204	104	218
Total Plant Investment	571	1246	605	1254	591	1242	637	1329
Owner's Costs	44	96	44	91	44	93	44	92
Total Capital Requirement	615	1342	649	1346	635	1335	681	1421
O&M COST*								
Fixed O&M Costs, $/kW-Yr	47.08		46.48		46.41		48.43	
Variable O&M, mills/kWh	2.41		2.11		2.14		2.08	
Fuel Costs, mills/kWh	12.90		12.53		12.72		12.27	
COST OF ELECTRICITY								
Current $, mills/kWh**	67.4		66.6		66.5		68.8	
Constant $, mills/kWh**	49.1		48.4		48.4		49.9	
Current $, mills/kWh***	60.3		59.2		59.3		60.6	
Constant $, mills/kWh***	39.0		38.3		38.4		39.2	

*First-Year O&M Cost, at 85% Capacity Factor
**Ten-Year Levelized Cost at 65% Capacity Factor
***Thirty-Year Levelized Cost at 85% Capacity Factor

Table 2 provides a cost summary. The differences in the total capital requirement among the Cases 1, 1CO1 and 1CO2 (1335-1346 \$/kW) are less than one percent. The highest total capital requirement is for Case 1TO1 at 1421 \$/kW, primarily the result of the optimization according to efficiency criteria.

Case 1CO1 possesses the lowest cost of electricity (COE) with 38.3 mills/kWh in constant dollars, 1.83 percent lower than the COE of the original Case 1(39.0 mills/kWh). Case 1CO2 is the next lowest with a COE of 38.4 mills/kWh, which is 1.61 percent lower than the COE of the original Case 1. The highest COE is for Case 1TO1 at 39.2 mills/kWh, 2.30 percent greater than for Case 1CO1, but only 0.44 percent greater than the COE for the original Case 1.

The difference in COE between Case 1CO1 (1CO2) and the original Case 1 results in savings of over 2.4 (2.2) million constant (mid-1990) dollars per year of plant operation. Compared with the original Case 1, the 30-year pre-tax present value of the cost savings in Cases 1CO1 and 1CO2 are 30.0 and 27.7 million constant mid-1990 dollars, respectively.

This study identified several design changes which improve the cost effectiveness of the Case 1 IGCC concept. These changes include the following:

- The coal should be supplied to the gasifier with the "as received" moisture of 11.12 weight percent, or with the highest moisture content allowed by the reliability of the coal feeding process.

- The gasification temperature and the steam high pressure should be optimized using the thermoeconomic variables discussed in section 3. This optimization refers to the interaction between gasification island and power island.

- The heat of the flue gas from the sulfation area should be used in the HRSG.

- The mass flow rate of the quench steam should be decreased and its temperature increased through adjustments in the steam turbine and elimination of the desuperheating process.

- The BFW preheater in area 250 (air booster compression) and the recycle gas cooler should preheat LP (or low-temperature) instead of HP (or high-temperature) feedwater.

- The recycle gas should be extracted from the clean gas after the exit gas cooler instead of after the zinc-ferrite unit.

- The size of the product gas cooler and the exit gas cooler should be increased to accommodate some of the above changes.

- The design of the HP steam turbine should be adjusted to the new steam high pressure values.

- The deaerator should be operated at the lowest possible pressure.

- At least one LP steam extraction should be used to preheat feedwater.

These recommendations refer only to the design of Case 1 and should not be used automatically in conjunction with other IGCC concepts. The cost optimal Cases 1CO1 and 1CO2 are certainly not unique. Other design configurations can be found with comparable cost of electricity values.

General Electric supplied new performance data for the MS7001F gas turbine. These data have significant impact on the overall efficiency and the cost of electricity. Use of the new data results in (a) an increase in thermal efficiency by 0.7-0.8 percentage points in the optimized cases, and (b) a decrease in the cost of electricity by more than 2.5 percent.

The parametric study conducted to investigate the impact of major design parameters on the efficiency and cost of electricity established the importance of the steam high pressure and the gasification temperature for the cost optimization process. The cost sensitivity studies confirmed that the plant capacity factor is the most important variable for cost-effective plant operation.

The thermoeconomic analysis and optimization techniques were very useful tools in conducting this study. Some optimization techniques were refined during the investigations and will be applied to future design optimizations of IGCC plants and other energy systems.

Future studies should include investigation of the economic feasibility of design options aimed at eliminating or modifying the external desulfurization step and the gas recycling process. Significant performance and cost benefits should be expected from the elimination or modification of these processes.

Reference

1. Southern Company Services, "Assessment of Coal Gasification/Hot Gas Cleanup Based Advanced Gas Turbine Systems," Final Report prepared for the U.S. Department of Energy, Morgantown Energy Technology Center, Contract No. DE-FC21-89MC26019 December 1990.

1.0 INTRODUCTION

Several studies of Integrated Gasification-Combined-Cycle (IGCC) power plants have indicated that these plants have the potential for providing performance and cost improvements over conventional coal-fired steam power plants with flue gas desulfurization, assuming equal plant capacity factors. Generally, IGCC power plants have a higher energy-conversion efficiency, require less water, conform with existing environmental standards at lower cost, and are expected to convert coal to electricity at lower costs than coal-fired steam power plants.

Most IGCC power plant designs currently under development require cooling of the hot gas from the coal gasifier to near-ambient temperatures to facilitate cleanup of sulfur, particulates, and other contaminants. The gas must then be reheated before being burned in a combustion turbine. The volume of gas which must be cleaned is reduced by using oxygen from an air separation plant as the primary fuel oxidant in the gasifier. However, the requirements of gas cooling/reheating and air separation result in capital cost and thermal efficiency penalties which adversely affect the economics of IGCC power plant designs using an oxygen-blown gasifier and a cold gas cleanup process.

Hot gas cleanup, in which sulfur and particulates are removed from a gas stream at high temperatures, has the potential to increase the cost competitiveness of IGCC plants. Using air as the coal oxidant rather than oxygen may yield further cost advantages.

Southern Company Services (SCS), Inc., under a cooperative agreement with the U.S. Department of Energy, Morgantown Energy Technology Center (DOE/METC), conducted a study entitled "Assessment of Coal Gasification/Hot Gas Cleanup Based Advanced Gas Turbine Systems" [1]. The objective of that study was to compare the estimated costs, performance, and reliability of several Kellogg-Rust-Westinghouse (KRW)-based IGCC power plant configurations. The power plants were assumed to be located at the Plant Wansley site of Georgia Power Company. The project team consisted of SCS, the engineering and research subsidiary of The Southern Company, the M. W. Kellogg Company (MWK), the Tennessee Technological University (TTU) Center for Electric Power, and General Electric Company (GE).

Among the power plant configurations compared in that study [1], the so-called Case 1 configuration was found to be an attractive alternative for generating electricity from coal. This configuration uses four air-blown KRW coal gasifiers, a hot gas cleanup system, and two GE MS7001F gas turbines to generate 458.4 MW electric power at a net heat rate of 8,595 Btu/kWh. The Higher-Heating-Value (HHV)-based net thermal efficiency is 39.70 percent. Assuming ten-year levelized costs at 65 percent average plant capacity factor, the estimated levelized cost of electricity is 67.4 mills/kWh in current dollars, or 49.1 mills/kWh in constant mid-1990 dollars.

For the results reported in Reference [1], no attempt was made to minimize the Cost of Electricity (COE) generated in each IGCC power plant. Thus, we should expect that the potential for improving the efficiency and cost effectiveness of each IGCC configuration studied in Reference [1] is different. Reference [1] discusses the comparison of Case 1 with the Reference Case (oxygen-blown KRW gasifier and cold gas cleanup). This comparison indicates that significant differences in the potential for improving each case exist. Final conclusions from the comparison of different configurations, however, should be drawn after these configurations have been optimized from the economic viewpoint, or after at least the potential for further improvements has been estimated. The second law analysis and the thermoeconomic analysis of Case 1 reported in Reference [1] identified a number of design changes that could potentially improve the performance and reduce the Cost of Electricity (COE).

To study various design options for Case 1 and determine the effect of the most important plant parameters on performance and COE, SCS entered into a supplementary project with the U.S. DOE/METC. The focus of the study was to optimize the design of Case 1 using advanced thermoeconomic evaluation and optimization techniques. The present report summarizes the results of this study. As in the previous study [1], the IGCC power plant was assumed to be located at the Plant Wansley site of Georgia Power Company, an operating subsidiary of The Southern Company. Every effort was made to put the results of the present study on the same basis as those reported in References [1], [2] and [3].

2.0 PROJECT DESCRIPTION

2.1 PROJECT OBJECTIVES

The major objectives of this project are to:

- Study the effect of various design options on the efficiency and the cost of electricity generated by an IGCC power plant using air-blown KRW coal gasifiers, a hot gas cleanup system, and industrial gas turbines.

- Develop a power plant design characterized by minimum cost of electricity.

For this purpose, several design configurations were evaluated. This report discusses the most interesting features of each attractive configuration.

2.2 PROJECT ORGANIZATION

The simulation, evaluation, and optimization studies were conducted at Tennessee Technological University (TTU) Center for Electric Power under a subcontract to Southern Company Services (SCS). Input data were provided by several companies. SCS Research and Environmental Affairs was responsible for overall project management and cost estimates in the power island. The M. W. Kellogg Company was responsible for developing gasification island designs and costs according to TTU specifications. Some other companies (General Electric, Industrial Filter and Pump Manufacturing and the Henry Vogt Machine Company) provided performance and cost data for specific equipment items.

2.3 IGCC PLANT DESCRIPTION

The IGCC power plant configuration for Case 1, as reported in Reference [1], is the base case for the studies discussed in this report. In the following we will refer to this case as original Case 1. This section briefly describes this case. Additional details may be found in Reference [1].

Figures 2-1 and 2-2 show simplified flow diagrams of the gasification island and the power island, respectively. The gasification island converts coal to a clean combustible gas that fuels a combustion turbine. The combustion turbine exhaust heat is used in a heat recovery steam generator (HRSG) to produce steam which drives a turbine generator. The integration between the gasification island and the steam cycle mainly involves (a) generation of saturated steam in the gasification island and use of this steam in the steam cycle, and (b) supply of steam at various temperature and pressure levels by the steam cycle to cover the demands of the gasification island. Case 1 has four separate processing trains for the gasification island (two trains for the Area 250-Booster Air Compression) with two combustion turbines and heat recovery steam generators and one steam turbine for the power island. In the following, the numbers given in parentheses refer to the material streams shown in Figures 2-1 and 2-2.

The coal receiving, handling, and preparation system (Area 100) includes an unloading system, twenty-eight vibrating feeders, a magnetic separator, four coal storage silos and four fluid-bed roller mills. Flue gas from the sulfator (48) provides the heat to dry the coal, which is fed into the gasifier with a moisture content of 4.98 weight percent.

The limestone receiving, handling and preparation system includes a rail car receiving hopper, a car shaker, two belt feeders, a conveyor, two reclaim hoppers, three storage silos, and two pulverizers, where limestone is dried with the heat supplied by an oil-fired heater. After preparation, the coal and limestone are transported to separate bunkers and then fed with weigh feeders into bins where the two are mixed. Four pneumatic conveying systems (one spare) are then used to transport the coal/limestone mixture to surge bins in the gasifiers. An air compressor provides pressurization of the gasifier lockhoppers.

The air booster compression system (Area 250) supplies the air necessary for the gasifier operation (5) and for the regeneration of the zinc-ferrite desulfurizer (29). Air for this system is obtained by extraction from the compressor of the gas turbine (4). The air booster compression system includes an air recuperator, a boiler feedwater (BFW) preheater, a trim cooler, and an air compressor.

The coal is gasified in a KRW pressurized fluidized bed gasifier (Area 300). Hydrogen sulfide, produced from the sulfur in the coal, is removed from the gas phase by reacting with calcium oxide which is obtained through the decomposition of limestone. The limestone in the gasifier serves as the bulk desulfurization step in the plant. The product gas from the gasifier (6) enters a cyclone which separates most of the fine particulates that escape the gasifier bed and returns them to the bed.

The gas exiting the cyclone (7) is partially cooled in the product gas cooler where HP steam is generated. Additional cooling is provided by quenching with steam (39). The amount of quench steam is determined by the desired moisture content of the gas (30% by volume) in order to achieve a satisfactory operation of the zinc ferrite unit.

The product gas (159), now around 1015°F, then passes through a non-recycle cyclone and a ceramic candle gas filter where all of the remaining particulates are removed from the gas. The collected solids are transferred to a depressurization lockhopper from which the solids, now at atmospheric pressure, are sent through a water-cooled conveyor to the sulfation area.

Since chlorides can severely affect the structural integrity of the zinc ferrite sorbent, a chloride guard is used to remove chlorides from the fuel gas (35). This guard consists of two parallel fixed beds of calcined nahcolite sorbent. When a bed becomes loaded as indicated by outlet gas (26) analysis, it will be isolated; the sorbent will be replaced and the bed will be available for the next duty cycle.

Zinc ferrite was selected as the external bed sorbent for high-temperature coal gas desulfurization because of its effectiveness and capability for sulfur absorption combined with its regenerative characteristics. In a similar manner to the chloride guard drums, the two zinc ferrite reactors, where final product gas desulfurization is achieved, are operated in a parallel arrangement. In the design of the original case 1, 86.5% of the coal sulfur is removed in the gasifier. After the additional removal in the zinc ferrite reactors, the total sulfur removal in the gasification island is 99.4%.

The zinc ferrite is regenerated through oxidation of the zinc and iron sulfides with air. This process begins with steam pressurization to a level above that of the gasifier. The flow of steam and air to the reactor is controlled to maintain the bed temperature of the oxidation front below 1500°F to avoid sintering and destroying the sorbent. The minimum temperature is 1100°F since lower temperatures will promote the formation of zinc sulfate. The regeneration gas (17) is recycled to the gasifier and the sulfur dioxide in the gas is captured by the limestone in the gasifier bed. The major part of the desulfurized product gas (84) enters the exit-gas cooler where the temperature is controlled at the maximum fuel-supply valve temperature (1000°F) for supply to the gas turbine. This temperature is controlled by varying the flow rate ($\dot{m}_{34}$) of the high-pressure saturated steam produced in the exit-gas cooler.

A small portion of the clean product gas (20) is cooled in the recycle gas cooler, compressed, and, recycled back to the gasifier (42) and the gas filter (15). The outlet gas temperature of the recycle gas cooler is controlled by the flow of boiler feedwater to the cooler (36).

The mixture of spent (sulfided) limestone, ash and fines (22) is prepared for disposal in the fluidized-bed sulfator. This system is used to oxidize the calcium sulfide to calcium sulfate, a chemical compound similar to gypsum, which can be readily disposed of in a dry landfill. The calcium sulfide oxidation is highly exothermic. At temperatures below 1600°F, little sulfur dioxide is formed. During this process, any residual carbon remaining in the fines is combusted. The heat is recovered by in-bed heat exchanger tubes that keep the bed temperature less than 1600°F without the need for high excess air. The gas from the combustor is cooled by an additional heat exchanger to approximately 1400°F before being routed to the coal preparation section of the plant for coal drying.

Two General Electric MS7001F combustion gas turbines were assumed in the combined cycle. These advanced combustion turbines will have a firing temperature of about 2300°F. Fuel gas (32) is introduced to the gas turbine combustor along with air (73) supplied by the compressor, which is driven by the gas turbine expander. The hot gas exiting the combustor (74) is supplied to the hot gas expander which in turn drives the gas turbine generator.

The exhaust gas from each combustion turbine (75) enters a heat recovery steam generator which provides steam generation and superheating of high-pressure (HP) and intermediate-pressure (IP) steam, reheating of IP steam, and feedwater preheating. The combined-cycle steam turbine consists of a high-pressure, intermediate-pressure and low-pressure sections. The HP section accepts the 1,500 psia/1000°F steam (101) from the two HRSGs. The exhaust steam (104) from the HP turbine is reheated to 1000°F by the HRSGs and returns to the IP turbine (105). The IP exhaust steam (141) is then routed to the LP turbine from which it is condensed at a design backpressure of 3.5" Hg.

Condensate from the condenser (118) enters two vertical motor driven condensate pumps and subsequently passes through the gland seal condenser. Steam for this heating is provided by the steam seal regulator (SSR). Subsequently, the condensate (121) is heated in the low-pressure feedwater heater (FWH2) before it enters (122) the two deaerators, which are an integral part of each HRSG. Finally, the electric motor driven HRSG feed pumps supply the feedwater (124) to its HRSG feedwater inlets (125, 129) and to the gasification process (36, 46, 49).

Tables 2-1, 2-2 and 2-3 summarize the results of the simulation and the exergy analysis for the original Case 1. Some small deviations between the numbers shown in these tables and the corresponding tables of Reference [1] are due to improvements in the simulation procedure. All results presented for the various cases in this report were obtained using identical assumptions for the simulation, exergy analysis, and thermoeconomic analysis.

In addition to the original Case 1 (gasification temperature - 1900°F), Reference [1] contains performance and cost data obtained for the Cases 1A and 1B, which refer to the same flow diagram as the original Case 1 but gasification temperatures of 1800°F and 2000°F, respectively. All these data were considered in the thermoeconomic optimization of Case 1.

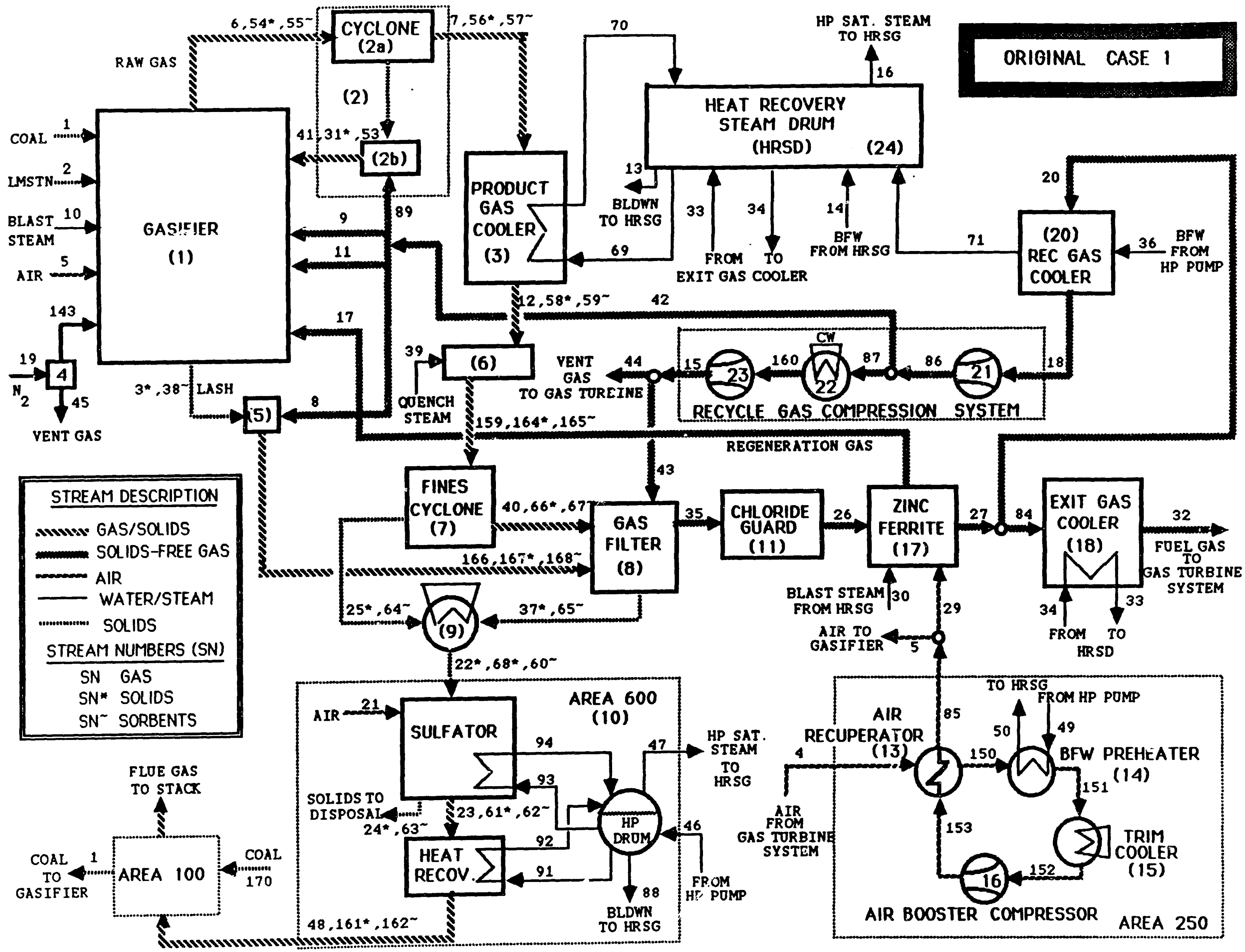

Figure 2-1. Flow Diagram of the Gasification Island in the Original Case 1.

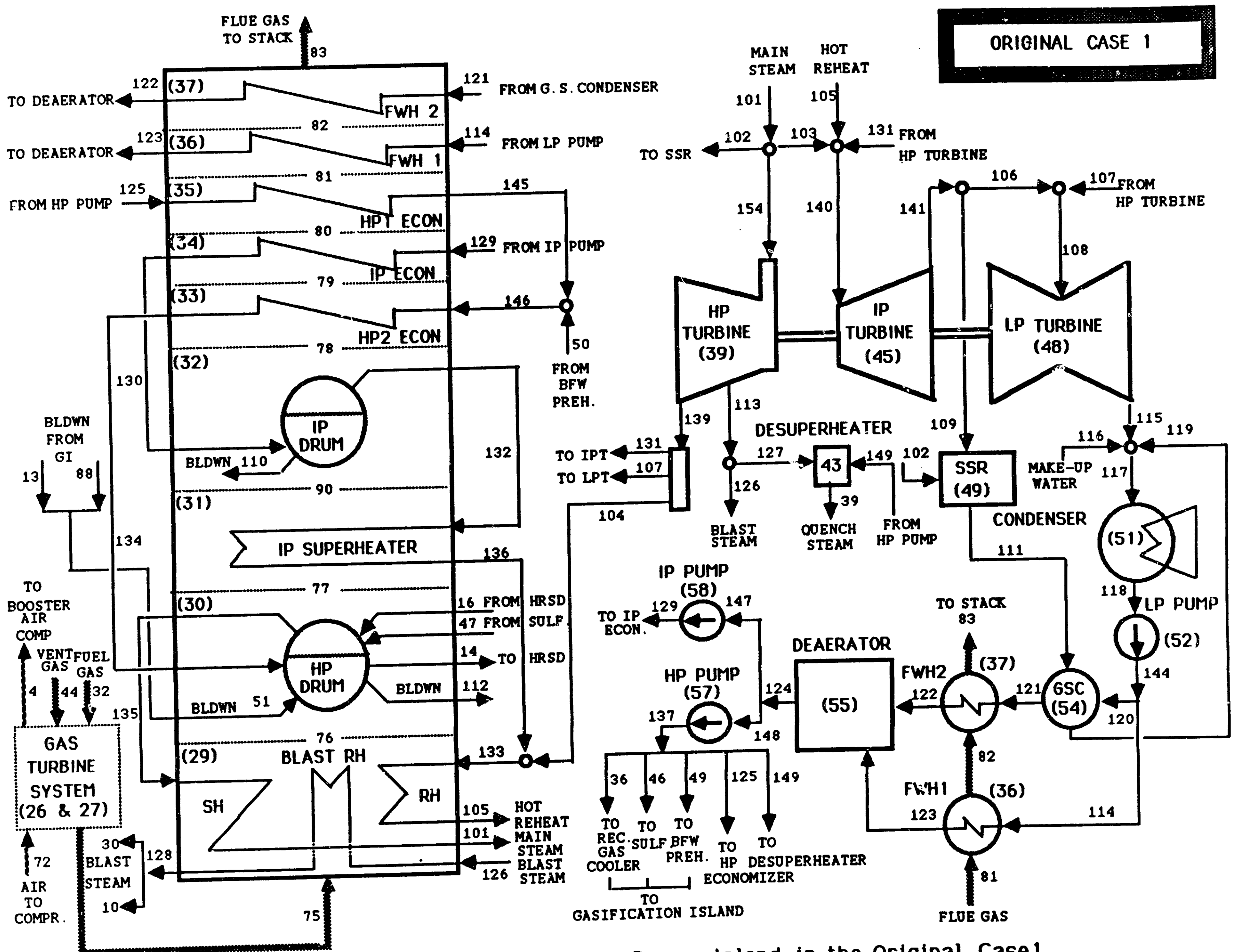

Figure 2-2. Flow Diagram of the Power Island in the Original Case1

TABLE 2-1

Mass Flow Rate, Temperature, Pressure and Flow Rates of Enthalpy, Entropy and Exergy for Each Stream of the Original Case 1 (Cont'd)

Stream No.	$\dot{m}$ [lb/hr]	T [F]	P [psia]	$\dot{H}$ [MMBtu/hr]	$\dot{S}$ [MBtu/hrR]	$\dot{E}$ [MW]
83	6736999	279.8	14.7	-6160.28	12046.16	33.51
84	1830903	1196.3	305.0	-3145.77	3943.58	920.24
85	1073962	650.0	450.0	152.20	1701.13	54.40
86	104465	411.0	450.0	-209.15	197.31	48.28
87	16342	411.0	450.0	-32.72	30.87	7.55
88	1779	604.9	1600.0	-11.13	2.95	0.09
89	7021	411.0	430.0	-14.07	13.29	3.23
90	6736999	627.1	14.9	-5529.11	12739.35	106.82
91	19343	604.9	1600.0	-121.05	32.13	1.01
92	19343	604.9	1600.0	-110.59	41.94	2.49
93	295659	604.9	1600.0	-1850.31	491.07	15.45
94	295659	604.9	1600.0	-1690.47	641.13	38.12
101	1344245	1000.0	1450.0	-7247.08	3288.01	241.82
102	568	1000.0	1450.0	-3.06	1.39	0.10
103	2313	1000.0	1450.0	-12.47	5.66	0.42
104	925235	644.7	350.0	-5131.89	2286.91	120.47
105	1025829	1000.0	329.0	-5495.88	2694.74	164.77
106	1043212	802.5	150.0	-5689.66	2755.43	135.64
107	1687	644.7	350.0	-9.36	4.17	0.22
108	1044899	802.2	150.0	-5699.02	2759.76	135.84
109	641	802.5	150.0	-3.50	1.69	0.08
110	1005	431.7	350.0	-6.51	1.45	0.02
111	1209	860.3	150.0	-6.56	3.22	0.16
112	13442	604.9	1600.0	-84.13	22.33	0.70
113	398731	727.3	500.0	-2196.97	983.42	56.55
114	110543	120.4	25.0	-751.07	111.25	0.06
115	1044899	120.6	1.7	-6039.28	2878.78	16.95
116	460852	80.0	14.7	-3149.81	430.58	0.16
117	1506960	120.6	1.7	-9197.19	3311.97	16.91
118	1506960	120.4	1.7	-10238.99	1516.52	0.83
119	1209	211.4	14.7	-8.10	1.39	0.00
120	1396417	120.4	25.0	-9487.77	1405.34	0.79
121	1396417	121.7	25.0	-9486.00	1408.38	0.82
122	1396417	160.9	24.5	-9431.26	1499.50	2.19
123	110543	240.1	24.5	-632.53	282.73	7.18
124	1506960	236.2	24.5	-10063.80	1791.66	7.85
125	853728	240.0	1650.0	-5695.19	1017.67	5.83
126	56698	727.3	500.0	-312.40	139.84	8.04
127	342034	727.3	500.0	-1884.57	843.58	48.51

TABLE 2-1

Mass Flow Rate, Temperature, Pressure and Flow Rates of Enthalpy, Entropy and Exergy for Each Stream of the Original Case 1 (Cont'd)

Stream No.	$\dot{m}$ [lb/hr]	T [F]	P [psia]	$\dot{H}$ [MMBtu/hr]	$\dot{S}$ [MBtu/hrR]	$\dot{E}$ [MW]
128	56698	950.0	450.0	-305.48	145.82	9.11
129	101600	237.0	350.0	-678.36	120.86	0.56
130	101600	391.0	350.0	-662.15	141.85	1.93
131	15711	644.7	350.0	-87.14	38.83	2.05
132	100594	431.7	350.0	-571.26	235.18	11.37
133	1025829	640.0	329.3	-5691.07	2541.02	132.33
134	1100379	568.0	1600.0	-6944.54	1772.42	49.37
135	1344245	604.9	1600.0	-7685.90	2914.98	173.31
136	100594	620.0	329.3	-559.18	248.16	12.82
137	1405360	240.0	1650.0	-9375.12	1675.24	9.60
138	1341364	992.3	1305.0	-7231.54	3295.54	238.95
139	942633	644.7	350.0	-5228.39	2329.91	122.74
140	1043853	994.5	329.0	-5595.49	2739.99	167.11
141	1043853	802.5	150.0	-5693.16	2757.12	135.73
142	1030113	604.9	1600.0	-5889.81	2233.79	132.81
143	3761	500.0	450.0	0.40	5.80	0.18
144	1506960	120.4	25.0	-10238.83	1516.59	0.86
145	853728	433.0	1625.2	-5524.08	1233.40	21.22
146	1100379	431.9	1600.0	-7121.40	1588.34	27.19
147	101600	236.2	24.5	-678.50	120.79	0.53
148	1405360	236.2	24.5	-9385.29	1670.87	7.32
149	47673	240.0	1650.0	-318.02	56.83	0.33
150	1073962	450.1	199.0	98.21	1707.59	37.54
151	1073962	266.3	198.0	49.41	1648.05	32.82
152	1073962	127.7	176.0	13.18	1601.36	29.73
153	1073962	341.9	453.0	69.39	1613.29	44.28
154	1341364	1000.0	1450.0	-7231.54	3280.96	241.30
159	1904715	1015.0	385.0	-3260.61	4036.74	973.68
160	16342	365.6	450.0	-32.98	30.56	7.53
161	67	1400.0	14.7	-0.32	0.04	0.01
162	134	1400.0	14.7	-0.59	0.06	0.01
164	13524	1015.0	385.0	-27.14	8.29	35.96
165	2115	1015.0	385.0	-8.53	0.79	0.05
166	15991	479.6	375.0	-31.62	30.89	7.40
167	28438	479.6	375.0	-139.95	9.67	7.09
168	49583	479.6	375.0	-205.42	13.91	0.27
170	337788	90.0	14.7	-343.01	111.15	1176.96

TABLE 2-2

Exergy Flow Rates of Fuel ($\dot{E}_F$), Product ($\dot{E}_P$), and Exergy Destruction ($\dot{E}_D$), Exergy Destruction Ratio (θ), Exergy Destruction to Total Exergy Input Ratio (θ^*) and Exergetic Efficiency (ζ) for Each Area and the Total Plant in the Original Case 1

Area	$\dot{E}_F$ [MW]	$\dot{E}_P$ [MW]	$\dot{E}_D$ [MW]	θ [%]	θ^* [%]	ζ [%]
Area 250: Booster Air Compressor	21.99	13.91	8.08	1.12	0.69	63.25
Area 300: KRW Gasification	1252.90	1009.09	243.82	33.81	20.67	80.54
Area 380: Recycle Gas Compression	54.22	52.02	2.20	0.31	0.19	95.94
Area 400: Gas Conditioning	196.15	174.75	21.39	2.97	1.81	89.09
Area 500: External Desulfurization	940.25	922.93	17.33	2.40	1.47	98.16
Area 600: Sulfation	33.88	21.71	12.17	1.69	1.03	64.07
Area 900: Gas Turbine System	636.13	360.89	275.23	38.17	23.33	56.73
Area 1000: HRSG	225.13	190.87	34.27	4.75	2.91	84.78
Area 1100: Steam Cycle	233.20	182.20	51.00	7.07	4.32	78.13
Total Plant Exergy Losses			45.01	6.24	3.82	
Service Station Power			10.58	1.47	0.90	
Total Plant	1179.50	458.42	721.08	100.00	61.13	38.87

TABLE 2-3

Heat Loss, Power Supplied (Generated), Exergy Destruction Flow Rate, Exergy Destruction Ratio (θ), Exergy Destruction to Total Exergy Input Ratio (θ^*) and Exergetic Efficiency (ζ) for Plant Components in the Original Case 1

	Component	$\dot{Q}_{Loss}$ [MW]	$\dot{W}$ [MW]	$\dot{E}_D$ [MW]	θ [%]	θ^* [%]	ζ [%]
1	Gasifier	9.74	1.00	216.96	30.09	18.39	82.38
2	Cyclones	1.75	0.00	1.74	0.24	0.15	99.22
3	Product Gas Cooler	1.47	0.00	24.61	3.41	2.09	65.48
4	Coal Hopper System	0.00	0.03	0.44	0.06	0.04	32.12
6	Quench Steam Mixing	0.00	0.00	17.56	2.43	1.49	98.29
7/8	Fines Cyclone & Gas Filter	0.03	0.00	1.85	0.26	0.16	98.96
9	Solids Conveyor & Cooler	0.84	0.00	0.46	0.06	0.04	98.93
10	Area 600: Sulfation	1.00	1.28	12.17	1.69	1.03	64.07
11	Chloride Guard	0.13	0.00	1.49	0.21	0.13	99.15
13	Air Recuperator	0.37	0.00	1.78	0.25	0.15	85.06
14	BFW Preheater	0.21	0.00	0.42	0.06	0.04	91.07
15	Trim Cooler	10.62	0.00	3.10	0.43	0.26	-
16	Booster Air Compressor	0.00	17.34	2.79	0.39	0.24	83.93
17	Zinc Ferrite System	0.00	0.00	9.83	1.36	0.83	99.01
18	Exit Gas Cooler	0.59	0.00	7.50	1.04	0.64	71.33
20	Recycle Gas Cooler	0.14	0.00	1.60	0.22	0.14	68.81
21	Recycle Gas Compressor I	0.44	1.42	0.53	0.07	0.04	62.89
23	Recycle Gas Compressor II	0.00	0.29	0.07	0.01	0.01	74.71
26	Gas Turbine/Air Compressor	37.07	(311.64)	82.65	11.46	7.01	79.04
27	Combustion Chamber	0.00	0.00	192.58	26.71	16.33	82.57
29	Superheater/Reheater/Blast	2.82	0.00	12.93	1.79	1.10	88.75
30	HP Steam Drum	1.00	0.00	3.46	0.48	0.29	90.12
31	IP Superheater	0.05	0.00	0.49	0.07	0.04	74.92
32	IP Steam Drum	0.37	0.00	2.82	0.39	0.24	77.01
33	HP2 Economizer	0.78	0.00	1.39	0.19	0.12	94.09
34	IP Economizer	0.07	0.00	0.78	0.11	0.07	63.65
35	HP1 Economizer	0.75	0.00	4.04	0.56	0.34	79.23
36	Feedwater Heater 1	0.52	0.00	4.18	0.58	0.35	62.99
37	Feedwater Heater 2	0.24	0.00	3.22	0.45	0.27	29.79
39	HP Turbine	0.00	(55.90)	6.12	0.85	0.52	90.13
43	Desuperheater	0.85	0.00	1.67	0.23	0.14	96.57
45	IP Turbine	0.00	(28.17)	3.22	0.45	0.27	89.75
48	LP Turbine	0.00	(98.13)	20.76	2.88	1.76	82.54
49	Steam Seal Regulator	0.00	0.00	0.02	0.00	0.00	87.94
51	Condenser	305.32	0.00	16.09	2.23	1.36	-
52	LP Pump	0.00	0.05	0.01	0.00	0.00	68.57
54	Gland Seal Condenser	0.00	0.00	0.13	0.02	0.01	17.41
55	Deaerator	0.00	0.00	1.52	0.21	0.13	83.76
57	HP Pump	0.00	3.37	1.09	0.15	0.09	67.68
58	IP Pump	0.00	0.05	0.02	0.00	0.00	64.84

2.4 RECOMMENDATIONS FROM THE PREVIOUS THERMOECONOMIC ANALYSIS

The thermoeconomic analysis reported in Reference [1] identified the following possible design changes to cost effectively improve the thermal efficiency.

- Supply the coal with the original moisture of 11.12 weight percent to the gasifier instead of the original design specification of 4.98 weight percent.
- Operate the gasifier in the temperature range of 1850°F to 1870°F versus 1900°F as designed.
- Redesign the recycle gas cooler, IP economizer, and IP drum to avoid heat transfer across the pinch point.
- Cool the gas at the outlet of the product gas cooler to a temperature lower than in the original Case 1 prior to adding steam to the gas.
- Further explore the option of using heat recovered in the sulfation area for steam generation rather than coal drying.
- Study the economic feasibility of developing and using a gas turbine system with a reheat stage.

All of the above recommendations, with the exception of the last one, were considered in the optimization studies reported here. It should be noted that each recommendation considered alone would improve the cost effectiveness of the original Case 1 power plant design. In general, however, when more than one recommendation is considered, some others may become less attractive or the recommended values might change.

2.5 TECHNICAL APPROACH

Several design options for the IGCC power plant shown in Figures 2-1 and 2-2 were studied. All of the design configurations developed and analyzed use

- an air-blown KRW gasifier with a constant carbon conversion ratio (96.5 percent),
- the same processes for gas conditioning, external desulfurization and sulfation, and

- the same combustion gas turbines receiving the fuel gas (constant energy flow rate based on heating value) always at 1000°F.

The major design changes studied refer to the

- total plant heat exchanger network (product gas cooler, heat recovery in sulfation and in booster air compression, recycle gas cooler, exit gas cooler, HRSG and feedwater preheaters),

- gasification temperature,

- coal drying process, and

- steam turbine.

The size of all plant components was kept variable.

To facilitate the comparison of results, the cases addressed in the present study employ the same basis as the cases developed in previous studies sponsored by DOE [1] and EPRI [2, 3]. The design approach, costing methodology, and economic evaluations are on a common basis.

Despite the fact that certain plant systems are not yet commercially available, technological maturity was assumed in the plant designs. In other words, each design is for an "n-th" plant, with several commercial plants assumed to have preceded the plant under consideration. The cost estimates do not, however, reflect widespread penetration of the technology in the utility market. All design configurations were based on the Plant Wansley site of Georgia Power Company.

For each design configuration, the detailed plant operation was simulated, the capital and O&M costs were estimated, a thermoeconomic analysis at the component level was conducted, and, finally, the cost of electricity was calculated. The latest results available from research projects and studies were used to estimate performance and costs.

The total plant performance was simulated using the THESIS (Thermodynamic and Economic Simulation System) software package [4, 5]. Simulation of the gasifier was based on data supplied by MWK and obtained using MWK's proprietary computer programs. The ASPEN-based material and energy balances provided by MWK for the gasification-island sections downstream of the gasifier were used to adjust the THESIS predictions. In addition, MWK furnished the data required to estimate the capital costs of the major components in the gasification island.

GE supplied the performance and cost data for the combustion gas turbines. The cost estimates for the HRSG and the steam cycle, as well as the economic evaluations and the calculation of the levelized COE, were done according to data and procedures provided by SCS.

The thermodynamic evaluation and comparison of the various design configurations were based on the exergy method. This method calculates the exergy destruction and exergetic efficiency of each plant component, as well as the exergy losses associated with material streams rejected to the environment. The exergy method provides an objective basis for detailed performance comparisons from the thermodynamic viewpoint and for estimating the potential to increase the overall plant thermal efficiency.

The thermoeconomic evaluation assesses the performance of a component or group of components from a combined thermodynamic and economic viewpoint and estimates the potential for improving the overall plant cost effectiveness through design changes. The procedures used for the thermoeconomic analysis and optimization are highly specialized and are still undergoing development. These procedures are discussed in greater detail in Section 3 of the report.

2.6 ORGANIZATION OF THE REPORT

The Contents of the report provide a detailed outline of the report material. Sections 1, 2 and 3 of the report contain background material. The optimal design configurations developed in this study are discussed and compared in Section 4. Section 5 presents the effect of important design and cost parameters on the overall thermal efficiency and the COE. Design configurations are described in Sections 2 and 4. Detailed results of the thermodynamic and thermoeconomic evaluations are presented in Sections 2 and 4 and in the Appendix.

3.0 THERMOECONOMIC EVALUATION AND OPTIMIZATION OF ENERGY SYSTEMS

This section contains a very brief introduction to the exergy analysis and the thermoeconomic evaluation and optimization of energy systems. These techniques are somewhat specialized and will not be described in detail in this section but will be summarized to the level necessary to discuss some of the final results and key conclusions. More details on the methodology are given in References [5] through [8].

3.1 EXERGY ANALYSIS

The second law of thermodynamics complements and enhances an energy balance by enabling calculation of both the real thermodynamic value of an energy carrier, and the real inefficiencies and losses from processes or systems. The concept of exergy (thermodynamic availability) is extremely useful for this purpose since an energy analysis generally fails to identify energy waste or the effective use of fuels and energy resources.

Exergy (E) is the maximum useful work attainable from an energy carrier under the conditions imposed by a given environment. Exergy is a thermodynamic property that depends on both the state of the carrier being considered and the state of the environment. It expresses the maximum capability of the energy carrier to cause changes. In most cases we can think of exergy as the useful part of energy, i.e., the part of energy that can theoretically be transformed into any other form of energy.

Unlike total energy, a part of the total exergy supplied to a system is irreversibly destroyed in all real processes. The **exergy destruction (E_D)** usually represents the largest part of what the layman calls "energy waste." The other part of "energy waste" is the **exergy loss (E_L)**, i.e., the exergy transfer out of a system associated with a stream rejected to the environment. The term "destruction" is used to identify the unrecoverable loss of exergy within the system, as distinct from the loss of exergy in an output stream. Both exergy destruction and exergy loss are identified through an exergy analysis.

In addition, an exergy analysis calculates the **exergetic efficiency** (second-law efficiency) of each plant component. The exergetic efficiency evaluates the true component performance from the thermodynamic viewpoint and is very useful in overall plant design evaluations. The definition of the exergetic efficiency must be consistent with the **purpose** of using the system or component being considered.

To understand the term exergetic efficiency, it is helpful to think of each component as having a "product," which represents the desired result from the component, and a "fuel," which represents the driving force for the process, or the resources used to obtain the "product." In the following, the terms "fuel" and "product" for a component are used without quotation marks. Using this terminology, the exergetic efficiency (ζ_k) of the component is defined as the ratio of the exergy in the product ($\dot{E}_{P,k}$) to the exergy in the fuel ($\dot{E}_{F,k}$)

$$\zeta_k = \frac{\dot{E}_{P,k}}{\dot{E}_{F,k}} \tag{3-1}$$

The **exergy balance** shows that the difference between exergy in the fuel and exergy in the product is the sum of exergy destruction and exergy loss in the component being considered:

$$\dot{E}_{F,k} - \dot{E}_{P,k} = \dot{E}_{D,k} + \dot{E}_{L,k} \tag{3-2}$$

The greater the percentage of the fuel exergy retained in the product, and, thus, the lower the extent of exergy destruction and exergy loss, the higher the exergetic efficiency of the component.

The objectives of an exergy analysis are:

- To identify the real thermodynamic losses (exergy destruction and exergy losses) in an energy system and to understand the effects causing the losses (chemical reactions, heat transfer, mixing, friction, etc.)

- To facilitate feasibility and optimization studies during the preliminary design phase of a project, as well as process improvement studies for an existing system

- To assist decision-making concerning plant operation and maintenance and allocation of research funds

In an exergy analysis we calculate, among others, the exergy flow rate of each stream ($\dot{E}_i$), the flow rate of exergy destruction ($\dot{E}_{D,k}$), and the exergetic efficiency (ζ_k) of each plant component. The thermodynamic evaluation of each plant component is also based on the ratios (θ) of exergy destruction in a plant component to (a) the total plant exergy destruction ($\sum_k \dot{E}_{D,k}$), Equation 3-3, and (b) the total exergy input to the plant ($\dot{E}_{tot,in}$), Equation 3-4:

$$\theta_k = \frac{\dot{E}_{D,k}}{\sum_k \dot{E}_{D,k}} \tag{3-3}$$

$$\theta_k^* = \frac{\dot{E}_{D,k}}{\dot{E}_{tot,in}} \tag{3-4}$$

These exergy destruction ratios can be used for comparisons among various components of the same plant and among similar components of different plants which use the same fuels as energy input to the total plant.

3.2 THERMOECONOMIC EVALUATION

Exergy is not only a measure of the true thermodynamic value of an energy carrier but is also closely related to the economic value of the carrier since users pay only for the useful part of energy. **A thermoeconomic analysis combines an exergy analysis with an economic analysis at the component level.** The objectives of a detailed thermoeconomic analysis include all the previously mentioned objectives of an exergy analysis in addition to the following:

- To shed light on the cost formation process, and, thus, facilitate studies to effectively reduce the product costs in an energy system.
- To estimate economically optimal operating conditions for a given design configuration.
- To understand the interactions between the thermodynamic performance of each plant component and the cost of the final plant product(s).
- To calculate the production costs of various products generated in the same process.
- To enable cost minimization studies in very complex energy systems.

In addition to mass, energy, and exergy balances, **cost balances** are formulated for each system component by assigning a cost value to the exergy (not the energy) of each stream entering or exiting the component. This procedure, **exergy costing,** is based on the finding that exergy is the only rational basis for assigning costs to streams as well as to "energy waste" (exergy destruction and exergy losses). With the aid of cost balances and some auxiliary assumptions, the cost per unit of exergy for each stream is calculated. Subsequently, we determine the average cost of (a) providing a unit of fuel exergy to the k-th plant component ($c_{F,k}$), (b) generating a unit of product exergy in the k-th plant component ($c_{P,k}$), and (c) the exergy destruction rate in the k-th plant component ($\dot{D}_k$). Using this terminology, the cost balance is written as follows:

$$c_{F,k}\,\dot{E}_{F,k} + \dot{Z}_k = c_{P,k}\dot{E}_{P,k} + c_{F,k}\dot{E}_{L,k} \tag{3-5}$$

Here, $\dot{Z}_k$ expresses the contribution of the investment costs and the operating (excluding fuel) and maintenance (O&M) costs associated with the k-th component to the product cost $c_{P,k}$. In the discussion below, these costs are called "capital costs" in order to distinguish them from the exergy costs ("fuel costs") for a plant component.

In the following discussion of the thermoeconomic evaluation we assume that the variables $c_{F,k}$ and $\dot{E}_{P,k}$ remain constant. In addition to the variables discussed in section 3.1, the following parameters are used for evaluating the performance of the k-th component or group of components from the thermoeconomic viewpoint.

1. The **cost of exergy destruction** in the k-th system is calculated in this study from the following relationship:

$$\dot{D}_k = c_{F,k}\dot{E}_{D,k} \tag{3-6}$$

This is the cost of the fuel used to cover the exergy destruction in the system.

2. The **relative cost difference** (d_k) between average cost per exergy unit of product and average cost per exergy unit of fuel:

$$d_k = \frac{c_{P,k} - c_{F,k}}{c_{F,k}} = \frac{\dot{D}_k + \dot{Z}_k}{c_{F,k}\dot{E}_{P,k}} = \frac{1 - \zeta_k}{\zeta_k} + \frac{\dot{Z}_k}{c_{F,k}\dot{E}_{P,k}} \tag{3-7}$$

This equation reveals the real cost sources in the k-th system, which are (a) the capital costs ($\dot{Z}_k$), and (b) the exergy destruction in the system, as expressed by the first term on the equation right side. In general, the higher the relative cost difference d_k, the more attention should be paid to the k-th system.

3. The **thermoeconomic factor** f_k:

$$f_k = \frac{\dot{Z}_k}{\dot{Z}_k + \dot{D}_k} \tag{3-8}$$

which expresses the contribution of the capital costs to the relative cost difference d_k.

4. For the thermoeconomic evaluation of heat exchangers we use, in addition to those discussed above, the variable r_k defined by:

$$r_k = \frac{\dot{Z}_k}{\dot{E}_{P,k}} \tag{3-9}$$

This variable states the capital costs required to transfer a unit of exergy to the cold stream of the heat exchanger.

All these variables are used in the thermoeconomic evaluation to determine what changes in the plant structure or in a variable (temperature, pressure, etc.) could lead to a decrease in the cost of electricity.

3.3 THERMOECONOMIC OPTIMIZATION

Cost optimization for a complex energy-conversion system such as the IGCC power plant presented in section 2.3 is usually expensive and requires knowledge of engineering, science, and business. The goal of optimization is to find the design configuration and the values of the system variables (the temperature, pressure, and chemical composition of flow streams, equipment size, materials, etc.) that minimize the cost of electricity. This usually involves a trade-off between capital and fuel costs for the entire system. Typical problems in the design and operation of energy systems have many workable solutions - sometimes an infinite number. Selecting the best solution requires engineering judgment, intuition, and critical analysis.

In many cases a rigorous cost optimization for a complex energy system is not possible because some of the cost functions that are needed to express the capital cost of a component as a function of thermodynamic variables (temperatures, pressures, etc.) are either unavailable or inaccurate. But even in cases in which all the information is available and acceptably accurate, it is expensive and time-consuming to formulate and solve an optimization problem with an extremely large number of equations, constraints, and highly interdependent variables.

Traditionally, design optimization includes the following steps. First, a detailed system configuration is developed; material and energy balances are conducted for this configuration. Then, product costs are estimated through an economic analysis. The third step includes development of a modified/new configuration that accounts for the corresponding material and energy balances. Subsequently the product costs for the new configuration are calculated. The last two steps are repeated several times.

Development of new process configurations is based, among other factors, on the experience and intuition of design engineers. Several decisions must be made with respect to thermodynamic variables. The final selection criterion, however, is economic. It is apparent that judiciously combining the thermodynamic and economic analyses, as in the thermoeconomic analysis, is advantageous to the optimization process.

Assuming well-designed total system configurations, the contribution of the capital costs to the final product costs decreases with decreasing thermodynamic efficiency (increasing exergy destruction), whereas the fuel cost increases with decreasing efficiency (see Figure 3-1).

Conventional optimization techniques seek the best trade-off between capital costs and fuel costs for the entire system. In thermoeconomics, a fuel is defined for each plant component. Thus the search for an optimum is simplified since these trade-offs can be made at the component level. If the relationship between investment costs and thermodynamic efficiency of the k-th component is known (e.g., Equation 3-10), then the optimal thermodynamic efficiency from the cost viewpoint can be calculated, Equation 3-11.

$$I_k = I_{o,k} + g_k \left[\frac{\zeta_k}{1 - \zeta_k} \right]^{n_k} E_{P,k} \qquad (3\text{-}10)$$

$$\zeta_k^{OPT} = \frac{1}{1 + \sqrt[n_k+1]{F_k}} \qquad (3\text{-}11)$$

with

$$F_k = \frac{(\epsilon_k + \gamma_k) n_k g_k}{\tau c_{F,k}} \qquad (3\text{-}12)$$

In these equations, $I_{o,k}$ represents the part of investment costs (I_k), which is independent of the component efficiency and capacity, ϵ_k is the capital recovery factor, γ_k represents a coefficient which indicates what part of the fixed O&M costs depends on the investment cost I_k, τ is the annual time of plant operation at the nominal capacity and g_k and n_k are constants which depend on the component being considered. The variable F_k is called the thermoeconomic similarity number of the k-th component.

The cost data provided for this project did not always support a relationship according to Equation 3-10. In those cases, the following relationship between investment costs and exergy destruction was used:

$$I_k = \frac{g_k^*}{E_{D,k}^{n_k}} E_{P,k} \qquad (3\text{-}13)$$

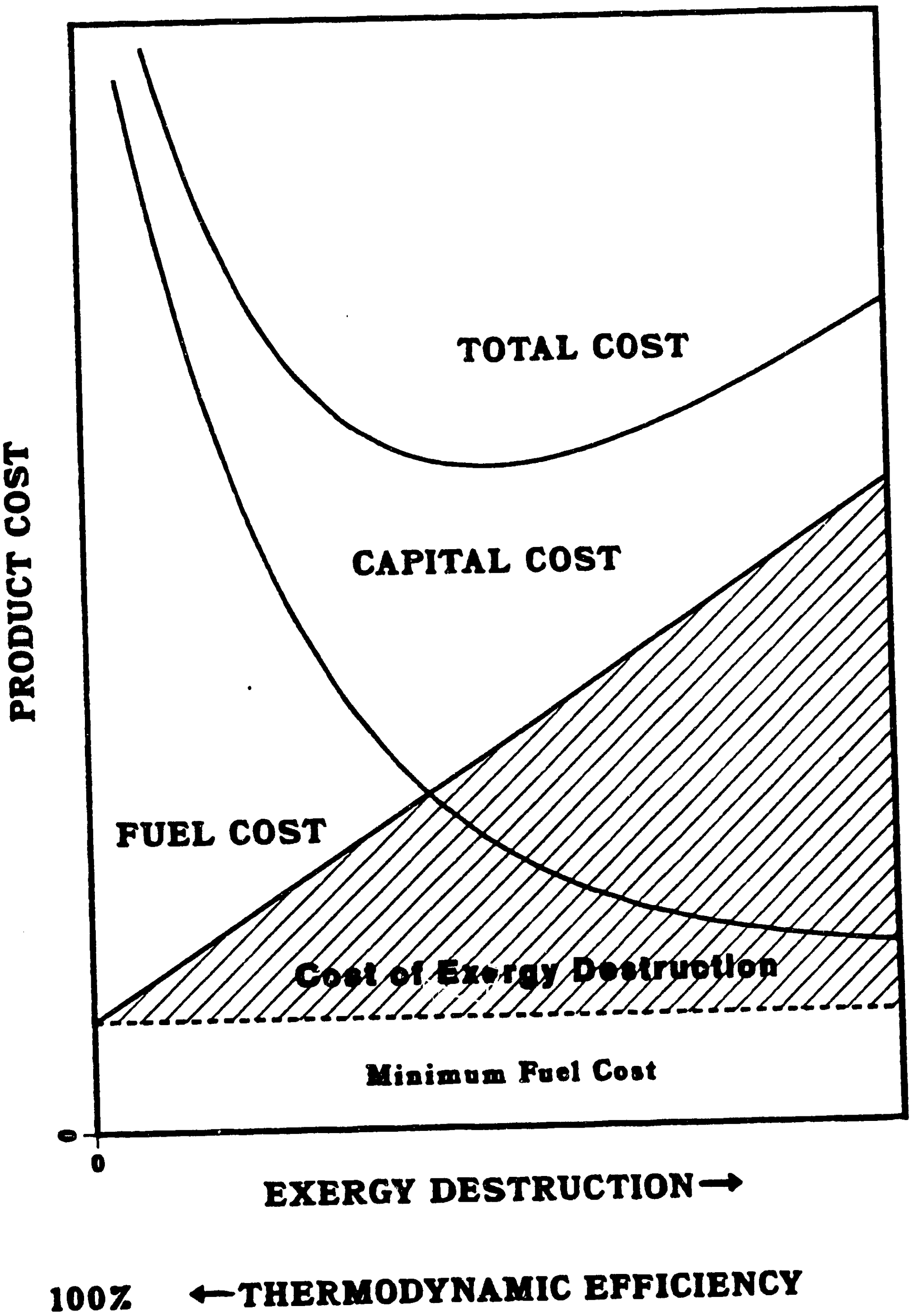

Figure 3-1. Contribution of Fuel and Capital Costs to the Product Cost as a Function of the Thermodynamic (Exergetic) Efficiency.

It is apparent that the numerical values of the constants g_k and g_k^* in Equations 3-10 and 3-13 are different. The same symbol is used in both equations to indicate the similarities between the equations. When Equation 3-13 is used, the thermoeconomically optimal exergetic efficiency, exergy destruction, and investment cost are given by

$$\varepsilon_k^{OPT} = \frac{1}{1 + \sqrt[n_k+1]{F_k^* / E_{P,k}^{n_k}}} \tag{3-14}$$

$$E_{D,k}^{OPT} = \sqrt[n_k+1]{F_k^* E_{P,k}} \tag{3-15}$$

$$I_k^{OPT} = g_k^* \sqrt[n_k+1]{E_{P,k} / (F_k^*)^{n_k}} \tag{3-16}$$

with

$$F_k^* = \frac{(\epsilon_k + \gamma_k) n_k g_k^*}{\tau c_{F,k}} \tag{3-17}$$

The thermoeconomic similarity number, Equation 3-12 or 3-17, plays a significant role in optimizing the component performance from the thermoeconomic viewpoint using either Equations 3-10 and 3-11 or Equations 3-13 through 3-16.

All of the above optimization results have been obtained by assuming $E_{P,k}$ = constant and $c_{F,k}$ = constant. These assumptions are fulfilled when a single plant component is optimized. It is apparent, however, that these assumptions are not valid when the total plant is optimized and several design changes (including changes in the design structure) are considered simultaneously. In this case, an iterative procedure is required to optimize the total plant. The thermoeconomic variables discussed in section 3.2 are used in the iteration to achieve a fast convergence.

3.4 BENEFITS OF THERMOECONOMICS

The discussion in this section is more general than the scope of the present project dictates. This is done because thermoeconomics is a significantly younger discipline than exergy analysis and the benefits are not obvious to many energy engineers.

Today the field of thermoeconomics has matured to the point where it is a valuable analytical tool for the design, operation, and maintenance of energy systems; it is not yet, however, a fully developed discipline. Thus, studies involving further development of the basic aspects of thermoeconomics are currently being carried out in parallel with applications of this field to practical problems.

The effectiveness of reducing costs in the design or operation of an energy system increases when we understand the **real** causes and sources of costs. A thermoeconomic analysis identifies these sources and indicates the changes required to reduce product costs. This information, complemented by the engineer's intuition and judgment, assists in the effective reduction of the product costs in energy systems on a relatively short time scale compared with traditional approaches. Decisions about the design, operation, and repair or replacement of equipment are facilitated.

In addition, thermoeconomics provides an objective cost allocation to more than one product of the same process. For instance, a thermoeconomic analysis of a cogeneration plant (which produces electricity and process steam) will provide the cost of steam and the cost of electricity separately. The cost ratio of steam to electricity calculated by the analysis does not necessarily have to be reflected in their selling prices, but the plant operators should know the real cost of generating each form of energy. In this project, thermoeconomics calculates the cost at which the electric power is generated in the combustion turbine generator and the steam turbine generator separately.

The thermoeconomic analysis also shows how much raw fuel is required to produce each stream in the system. Finally, thermoeconomics helps managers decide how to allocate research and development funds to improve plant components that contribute most significantly to the product costs.

It is true that many conclusions obtained by a thermoeconomic optimization could also be obtained through a large number of conventional energy and economic analyses. The advantage of thermoeconomics is that it replaces an expensive and subjective search for cost reduction with an objective, well-informed, systematic, and, therefore, shorter search in which all of the cost sources are properly identified and evaluated. The savings in both engineering and computer time are significant. Application of thermoeconomic analysis to new energy system concepts and complex installations (particularly those with several chemical reactions) results in significant savings.

4.0 RESULTS OF THE THERMOECONOMIC OPTIMIZATION

This section discusses the results obtained from the thermoeconomic optimization of the IGCC power plant described in section 2.3. Based on the recommendations from the previous thermoeconomic analysis [1] conducted for case 1 (see also section 2.4), the optimization efforts focused on the gasifier (mainly in determining the optimum coal moisture and gasification temperature), the steam turbines, and the total plant heat exchanger network. The latter includes the product gas cooler and heat recovery steam drum, heat recovery in sulfation and HP drum, recycle gas cooler, exit gas cooler, heat recovery in booster air compression, HRSG, and feedwater preheaters.

In addition to the factors considered in this study, a complete Case 1 design optimization should consider various options for hot gas cleanup and an optimization of the in-bed sulfur removal per pass in the gasifier. These studies exceed the scope of the present project.

All performance and cost calculations were conducted using exactly the same assumptions presented and discussed in sections 4, 7, and 8 of Reference [1]. The only change was made in the value of the plant capacity factor used to minimize the cost of electricity. In all base case evaluations reported in Reference [1] a capacity factor of 65 percent was assumed to facilitate comparisons with other DOE-sponsored studies. However, the plant availability analysis, conducted with the aid of the UNIRAM software, indicated that the plant would be available 85 percent of the time [1]. In the Southern Company, the average plant availabilty is currently about 90 percent. Since the Case 1 IGCC power plant would be dispatched as a baseload plant, its capacity factor could approach the equivalent availability factor. Therefore, it was decided to use a plant capacity factor of 85 percent in the optimization. Tables 4-1 through 4-4 summarize the most important assumptions made to calculate the total plant facilities investment, the operating and maintenance costs, and the cost of electricity.

The final criterion used to evaluate the optimization results was the 30-year levelized cost of electricity. This cost was calculated in both current and constant dollars. Calculations on a constant dollar basis assume a zero inflation rate during plant construction and operation. The calculation procedure used to calculate the levelized cost of electricity emulates the DOE cost of electricity computer program, as developed at the Morgantown Energy Technology Center. This program is generally consistent with the EPRI Technical Assessment Guide (TAG) [9] and has a number of built-in costing assumptions based on experience and utility costing approaches.

TABLE 4-1
Process Contingency Factors and Maintenance Material Cost as a Percentage of Process Plant Cost (PPC) for each IGCC Plant Area

Area	Process Contingency Factor [%]	Maintenance Material Costs [% of PPC*]
Booster Air Compression	0	2.4
Coal Receiving and Handling	0	1.2
Limestone Receiving and Handling	0	1.2
Coal Pressurization/Feeding	5	2.4
Ash Depressurization/LASH Transport	10	2.4
Ash and Fines Handling and Disposal	0	1.2
Gasification/Fines Recycle	15	2.4
Heat Recovery	5	2.4
Fines Removal/Depressurization	15	2.4
Gas Cooling	0	2.4
Recycle Gas System	10	2.4
Resaturation/Startup Heater	0	2.4
Sulfation	15	2.4
Chloride Removal	10	2.4
Zinc Ferrite Sulfur Removal	10	2.4
Balance of Gasification Island	0	2.4
Gas Turbine System	5	0.9
Heat Recovery Steam Generation	5	0.9
Steam Turbine System	0	0.9
General Facilities	0	0.9

*These percentages are applied to the sum of each area process plant cost and the prorated engineering fees (excluding contingencies) to yield a total weighted average percent. The resulting value is multiplied by TPC to produce maintenance material costs. Maintenance labor costs are estimated based on plant staffing plans.

TABLE 4-2

Expenditure Schedule During Construction Period (4.5 Years)

Year	Expenditure
1	5%
2	15%
3	30%
4	35%
5	15%

TABLE 4-3

Unit Costs for Calculating the Annual O&M Costs (Mid-1990 Dollars)

Cost Item	Cost
Fuel, Illinois No. 6 coal	$35.00/ton
Limestone	$11.20/ton
Nahcolite	$261.25/ton
Zinc Ferrite	$6,270.00/ton
Miscellaneous	$1.00/ton
Ash and sorbent disposal	$2.94/ton
Average O&M labor	$21.80/hr

TABLE 4-4

Economic Assumptions for Cost of Electricity Calculation

Project life	30 years
Book life	30 years
Tax life	30 years
Federal and state income tax rate (composite)	37.7%
Investment tax credit	0%
Tax depreciation method	ACRS
Annual inflation rate	4.5%
Real annual escalation rates (over inflation)	
Fuel	0.5%
Operating & maintenance	0%
Plant facilities	0%
Year dollar basis	mid-1990
Capacity factor	85%

Financial Structure

Type of Security	% of Total	Cost in Current Dollars [%]	Cost in Constant Dollars [%]
Debt	45	10.3	5.6
Preferred stock	10	9.8	5.1
Common stock	45	13.8	8.9
Discount rate, [%] (cost of capital)		11.8	7.0

In the following, three optimized cases are presented. Subsequently, the changes and options considered in the optimization are discussed separately for each IGCC plant area. Case 1CO1 is the cost optimal IGCC plant design when a relatively high value for the steam high pressure is used. Case 1CO2 represents a cost optimal IGCC plant design with a value for the steam high pressure comparable with the value used in the original Case 1. Finally, Case 1TO1 represents a thermodynamically optimized case which demonstrates the potential for improving the overall plant thermal efficiency.

4.1 DESCRIPTION OF CASE 1CO1

Figures 4-1 and 4-2 show simplified flow diagrams of the gasification island and the power island. Tables 4-5, 4-6 and 4-7 summarize the results of the simulation and thermodynamic analysis for this case. Detailed results of the thermoeconomic analysis are given in the Appendix. Compared with the original Case 1, the following design changes were identified for the thermoeconomically optimal IGCC power plant design of Case 1CO1. The corresponding design options and values used in the original Case 1 (Figures 2-1 and 2-2; Tables 2-1, 2-2 and 2-3) are given in parentheses.

1. The coal moisture at the gasifier inlet is 11.12 weight percent (4.98 weight percent).

2. The gasification temperature is 1920°F (1900°F).

3. The raw gas is cooled in the product gas cooler to T_{12} = 1168°F (T_{12} = 1252°F).

4. The pressure of the HP steam generated in the gasification island is P_{16} = 2055 psia (P_{16} = 1600 psia).

5. The temperature and mass flow rate of the quench steam, which is mixed with the gas at the exit of the product gas cooler, are T_{39} = 663°F (T_{39} = 500°F) and $\dot{m}_{39}$ = 401,759 lbm/hr ($\dot{m}_{39}$ = 389,707 lbm/hr), respectively. The ratio $\dot{m}_{39}/\dot{m}_1$ is smaller in Case 1CO1 than in the original Case 1. Note that the temperature and moisture content of the gas after quenching, stream 159, remained constant during the optimization studies.

6. The combustion gas exiting the sulfator at T_{48} = 1098°F is filtered, mixed with the gas turbine exhaust, and supplied to the HRSG. (In the original Case 1, the combustion gas exited the sulfator at T_{48} = 1400°F and was used for coal drying.) As a result of this change, more HP saturated steam is generated in the sulfation area in Case 1CO1 than in the Original Case 1.

7. The boiler feedwater (BFW) preheater in the air booster compression area preheats low-pressure (high-pressure) feedwater. The heat rejection in the subsequent trim cooler is 2.95 MMBtu/hr (36.23 MMBtu/hr).

8. The recycle gas, stream 20, is extracted from the main gas stream at the outlet of the exit gas cooler (at the outlet of the zinc ferrite system).

9. The recycle gas cooler is used to preheat low-temperature (high-temperature) feedwater.

10. The steam cycle of Case 1CO1 does not generate or use any intermediate-pressure (IP) steam.

11. The HRSG of Case 1CO1 does not contain any steam drums. The heat supplied to the HRSG is used for superheating and reheating steam and for feedwater preheating but not for steam generation. HP steam generation in Case 1CO1 occurs exclusively in the gasification island.

12. The main steam pressure at the HP turbine inlet is P_{101} = 1849psia (P_{101} = 1,450 psia) and at the HP turbine outlet P_{139} = 500 psia (P_{139} = 350 psia). Thus, no extraction from the HP turbine is required in Case 1CO1. The blast steam and quench steam are taken at the HP turbine exhaust.

13. The steam required for the deaerator operation is extracted from the LP turbine in Case 1CO1 whereas it is generated in the FWH1 of the HRSG in the original Case 1. The deaerator operating pressure is 20 psia (24.5 psia).

14. No desuperheating of the quench steam is required in Case 1CO1. The desuperheater is shown in Figure 4-2 but the design mass flow rate of stream 149 is zero.

The justification for the above changes and the effects of the changes on the plant performance and costs are discussed in section 4.4. Section 4.5 compares Case 1CO1 with the original Case 1 and the other optimized cases.

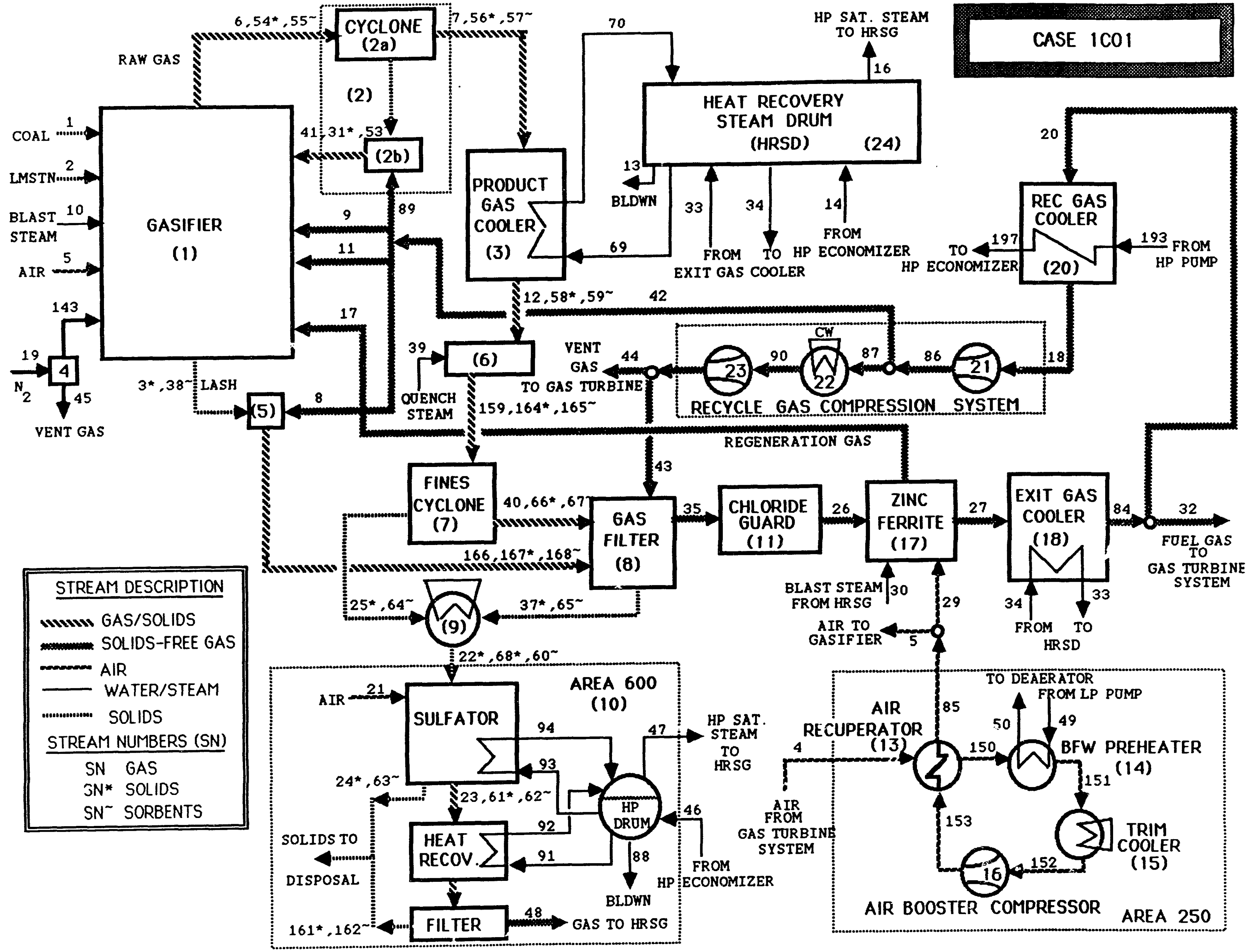

Figure 4-1. Flow Diagram of the Gasification Island in Case 1C01

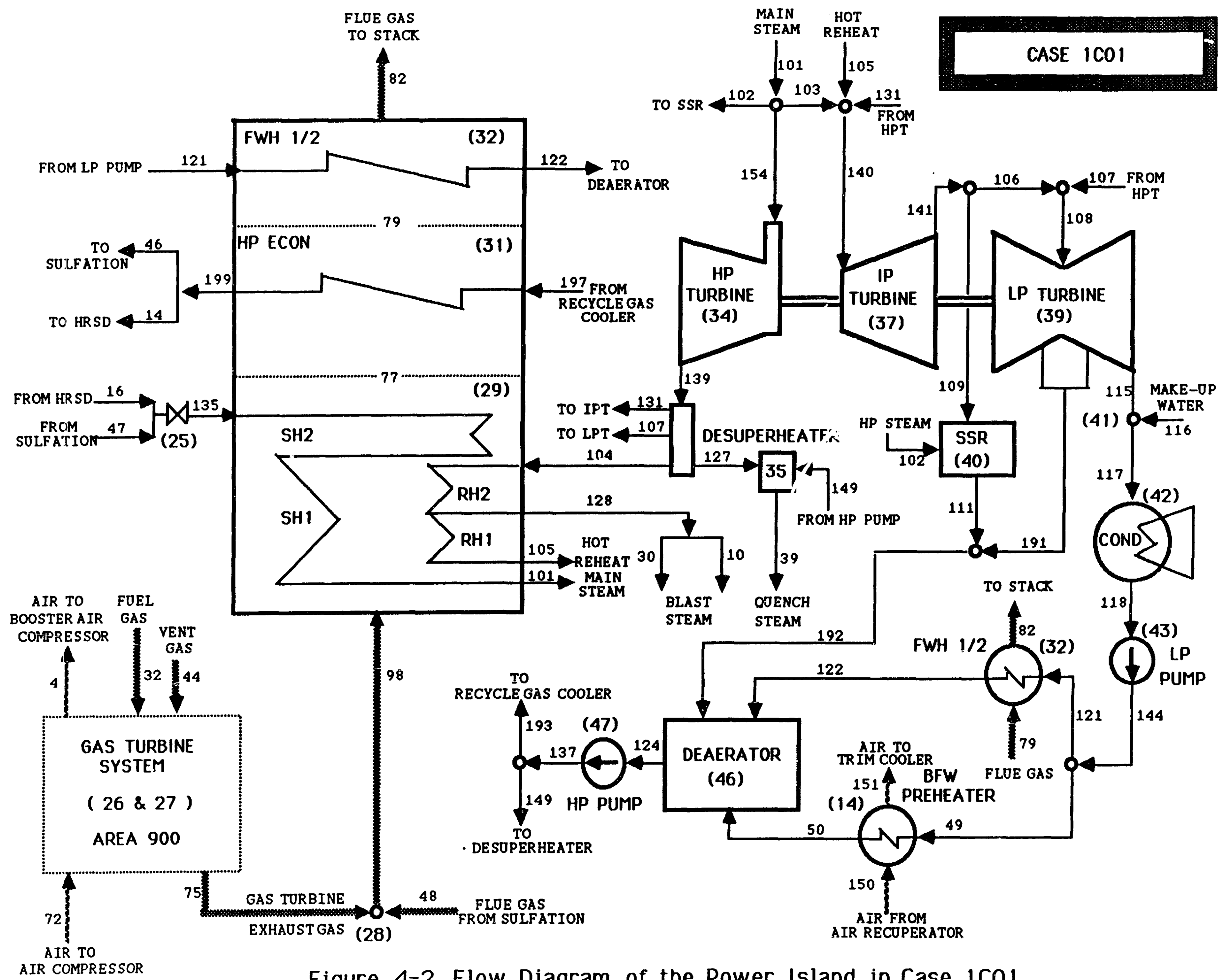

Figure 4-2. Flow Diagram of the Power Island in Case 1C01

TABLE 4-5

Mass Flow Rate, Temperature, Pressure and Flow Rates of Enthalpy, Entropy and Exergy for Each Stream of Case 1CO1

Stream No	$\dot{m}$ [lb/hr]	T [F]	P [psia]	$\dot{H}$ [MMBtu/hr]	$\dot{S}$ [MBtu/hrR]	$\dot{E}$ [MW]
1	345485	90.0	14.7	-503.74	134.72	1203.80
2	87703	90.0	14.7	-456.19	18.75	0.00
3	28031	500.0	425.0	-138.03	9.72	6.77
4	1140136	760.0	200.0	193.52	1896.78	52.48
5	1135222	650.0	450.0	160.88	1798.16	57.50
6	1611124	1920.0	400.0	-817.73	3361.49	1034.86
7	1611124	1920.0	395.0	-817.72	3363.10	1034.60
8	16115	411.0	430.0	-32.16	30.35	7.14
9	36896	411.0	430.0	-73.64	69.49	16.35
10	6897	950.0	450.0	-37.16	17.74	1.11
11	49374	411.0	430.0	-98.55	92.98	21.88
12	1611124	1168.1	385.0	-1220.41	3163.34	948.76
13	11857	639.7	2055.0	-73.55	20.27	0.72
14	1197697	639.7	2055.0	-7429.38	2047.10	72.64
15	16502	523.9	750.0	-32.30	30.98	7.51
16	1185840	639.7	2055.0	-6812.91	2518.30	151.86
17	34075	1200.0	430.0	-153.30	86.02	6.37
18	126280	330.0	285.3	-255.48	238.07	54.89
19	13094	500.0	450.0	1.40	20.20	0.64
20	126280	1000.0	295.0	-225.58	265.05	59.31
21	197485	160.0	20.0	3.96	326.47	0.76
22	29504	500.0	14.7	-141.46	10.34	10.52
23	190224	1600.0	19.8	-49.48	369.53	13.10
24	34428	150.0	14.7	-181.30	7.08	0.88
25	13260	1015.0	14.7	-29.36	8.02	32.86
26	2043491	1007.2	345.0	-3506.43	4320.67	998.50
27	2043144	1187.6	305.0	-3508.32	4373.67	987.31
29	4914	650.0	450.0	0.70	7.78	0.25
30	28935	950.0	450.0	-155.89	74.42	4.65
31	279919	1862.0	430.0	-482.45	242.08	722.28
32	1916864	1000.0	295.0	-3424.20	4023.26	900.29
33	304344	639.7	2055.0	-1748.52	646.32	38.98
34	304344	639.7	2055.0	-1887.86	520.18	18.46
35	2043353	1007.2	365.0	-3505.94	4310.51	1000.06
37	29504	494.0	14.7	-141.52	10.27	10.51
38	50587	500.0	425.0	-209.37	14.41	0.30
39	401759	663.1	450.0	-2226.89	983.83	54.23
40	2012883	1015.0	385.0	-3446.07	4241.17	988.21
41	7393	1862.0	430.0	-10.78	16.58	4.01

TABLE 4-5

Mass Flow Rate, Temperature, Pressure and Flow Rates of Enthalpy, Entropy and Exergy for Each Stream of Case 1CO1 (Cont'd)

Stream No	$\dot{m}$ [lb/hr]	T [F]	P [psia]	$\dot{H}$ [MMBtu/hr]	$\dot{S}$ [MBtu/hrR]	$\dot{E}$ [MW]
42	109778	411.0	450.0	-219.12	206.30	48.70
43	14355	524.0	750.0	-28.10	26.95	6.54
44	2147	524.0	750.0	-4.20	4.03	0.98
45	9215	90.0	14.7	0.03	15.15	0.02
46	419823	639.7	2055.0	-2604.18	717.56	25.46
47	415666	639.7	2055.0	-2388.09	882.73	53.23
48	190224	1098.0	19.6	-76.01	354.91	7.68
49	797647	120.4	23.0	-5419.51	802.74	0.45
50	797647	228.0	20.0	-5333.51	938.73	3.75
53	113266	1862.0	430.0	-435.63	53.56	6.95
54	294652	1920.0	400.0	-496.46	259.67	762.85
55	115575	1920.0	400.0	-442.97	55.31	7.44
56	14733	1920.0	395.0	-24.82	12.98	38.14
57	2310	1920.0	395.0	-8.85	1.11	0.15
58	14733	1168.1	385.0	-31.48	9.65	36.73
59	2310	1168.1	385.0	-9.24	0.91	0.07
60	52897	500.0	14.7	-218.93	15.07	0.31
61	68	1600.0	19.8	-0.32	0.04	0.01
62	137	1600.0	19.8	-0.59	0.07	0.01
63	68454	150.0	14.7	-320.11	13.80	0.01
64	2079	1015.0	14.7	-8.38	0.78	0.05
65	50818	494.0	14.7	-210.39	14.41	0.29
66	1473	1015.0	385.0	-3.26	0.89	3.65
67	231	1015.0	385.0	-0.93	0.09	0.01
68	13260	500.0	14.7	-32.36	5.54	32.38
69	881496	639.7	2055.0	-5467.97	1506.65	53.46
70	881496	639.7	2055.0	-5064.39	1871.98	112.89
72	5958123	90.0	14.7	18.91	9803.01	0.99
75	6736999	1098.0	15.0	-4801.56	13433.59	260.86
77	6927222	703.2	14.9	-5651.89	13222.50	132.81
79	6927222	305.1	14.9	-6400.78	12436.36	39.97
82	6927222	279.8	14.9	-6447.12	12375.72	36.16
84	2043144	1000.0	295.0	-3649.78	4288.31	959.60
85	1140136	650.0	450.0	161.57	1805.94	57.75
86	126280	411.0	450.0	-252.04	237.31	56.02
87	16502	411.0	450.0	-32.94	31.01	7.32
88	4156	639.7	2055.0	-25.78	7.10	0.25
89	7393	411.0	430.0	-14.75	13.92	3.28
90	16502	365.7	450.0	-33.20	30.70	7.29

TABLE 4-5

Mass Flow Rate, Temperature, Pressure and Flow Rates of Enthalpy, Entropy and Exergy for Each Stream of Case 1CO1 (Cont'd)

Stream No	$\dot{m}$ [lb/hr]	T [F]	P [psia]	$\dot{H}$ [MMBtu/hr]	$\dot{S}$ [MBtu/hrR]	$\dot{E}$ [MW]
91	57149	639.7	2055.0	-354.50	97.68	3.47
92	57149	639.7	2055.0	-328.33	121.36	7.32
93	358517	639.7	2055.0	-2223.90	612.78	21.74
94	358517	639.7	2055.0	-2059.76	761.36	45.91
98	6927222	1098.0	15.0	-4877.58	13793.28	267.79
101	1601506	1000.0	1848.6	-8654.23	3863.81	290.79
102	677	1000.0	1848.6	-3.66	1.63	0.12
103	2756	1000.0	1848.6	-14.89	6.65	0.50
104	1175588	663.1	500.0	-6519.94	2862.55	160.18
105	1139756	1000.0	470.0	-6110.95	2946.79	189.30
106	1160466	720.0	150.0	-6377.71	3025.37	143.07
107	2010	663.1	500.0	-11.15	4.89	0.27
108	1162476	719.9	150.0	-6388.85	3030.52	143.30
109	764	720.0	150.0	-4.20	1.99	0.09
111	1440	805.2	150.0	-7.85	3.81	0.19
115	1126619	120.6	1.7	-6534.34	3064.73	17.92
116	453603	80.0	14.5	-3100.26	423.80	0.16
117	1580222	120.6	1.7	-9634.60	3489.73	17.89
118	1580222	120.4	1.7	-10736.76	1590.24	0.87
121	782576	120.4	23.0	-5317.11	787.57	0.44
122	782576	178.7	20.0	-5271.47	862.55	1.74
124	1617520	228.0	20.0	-10815.64	1903.62	7.60
127	401759	663.1	450.0	-2226.89	983.83	54.23
128	35831	950.0	450.0	-193.05	92.15	5.75
131	18718	663.1	500.0	-103.81	45.58	2.55
135	1601506	638.6	2039.9	-9201.01	3402.00	204.94
137	1617520	232.7	2105.0	-10800.68	1910.12	10.94
139	1598074	663.1	500.0	-8863.10	3891.30	217.75
140	1161230	994.3	470.0	-6229.65	2999.86	192.22
141	1161230	720.0	150.0	-6381.90	3027.36	143.17
143	3879	500.0	450.0	0.41	5.98	0.19
144	1580222	120.4	23.0	-10736.62	1590.32	0.90
149	0	-	-	-	-	-
150	1140136	450.0	199.0	104.24	1812.78	39.85
151	1140136	138.4	198.0	16.93	1695.79	33.10
152	1140136	127.7	176.0	13.98	1700.01	31.56
153	1140136	341.9	453.0	73.64	1712.66	47.01
154	1598074	1000.0	1848.6	-8635.68	3855.53	290.17
159	2012883	1015.0	385.0	-3446.07	4241.17	988.21

TABLE 4-5

Mass Flow Rate, Temperature, Pressure and Flow Rates of Enthalpy, Entropy and Exergy for Each Stream of Case 1CO1 (Cont'd)

Stream No	$\dot{m}$ [lb/hr]	T [F]	P [psia]	$\dot{H}$ [MMBtu/hr]	$\dot{S}$ [MBtu/hrR]	$\dot{E}$ [MW]
161	68	1098.0	19.6	-0.34	0.04	0.00
162	137	1098.0	19.6	-0.61	0.06	0.00
164	14733	1015.0	385.0	-32.63	8.91	36.51
165	2310	1015.0	385.0	-9.31	0.86	0.05
166	16115	479.7	375.0	-31.78	30.97	7.15
167	28031	479.7	375.0	-138.22	9.52	6.74
168	50587	479.7	375.0	-209.58	14.19	0.27
170	345485	90.0	14.7	-503.74	134.72	1203.80
176	6927222	1054.1	15.0	-4965.28	13736.65	251.21
190	1601506	950.0	1938.0	-8709.86	3817.48	281.95
191	35857	374.8	22.0	-202.81	95.23	2.46
192	37298	391.3	22.0	-210.66	99.40	2.59
193	1617520	232.7	2105.0	-10800.68	1910.12	10.94
197	1617520	250.8	2083.9	-10771.22	1952.27	12.78
199	1617520	639.7	2055.0	-10033.56	2764.66	98.10

TABLE 4-6

Exergy Flow Rates of Fuel ($\dot{E}_F$), Product ($\dot{E}_P$), and Exergy Destruction ($\dot{E}_D$), Exergy Destruction Ratio (θ), Exergy Destruction to Total Exergy Input Ratio (θ^*) and Exergetic Efficiency (ζ) for Each Area and the Total Plant in Case 1CO1

Area	$\dot{E}_F$ [MW]	$\dot{E}_P$ [MW]	$\dot{E}_D$ [MW]	θ [%]	θ^* [%]	ζ [%]
Area 250: Booster Air Compressor	23.35	13.51	9.84	1.36	0.82	57.86
Area 300: KRW Gasification	1288.93	1028.71	260.23	35.95	21.57	79.81
Area 380: Recycle Gas Compression	61.44	58.06	3.38	0.47	0.28	94.50
Area 400: Gas Conditioning	201.59	182.77	18.82	2.60	1.56	90.66
Area 500: External Desulfurization	1003.40	986.49	16.91	2.34	1.40	98.31
Area 600: Sulfation	36.70	28.02	8.68	1.20	0.72	76.35
Area 900: Gas Turbine System	641.21	363.93	277.28	38.31	22.98	56.76
Area 1000: HRSG	232.38	207.34	25.04	3.47	2.07	89.22
Area 1100: Steam Cycle	264.40	209.95	54.45	7.52	4.51	79.41
Total Plant Exergy Losses			37.87	5.23	3.14	
Service Station Power			11.15	1.54	0.92	
Steam Transport Losses			0.16	0.02	0.01	
Total Plant	1206.35	482.55	723.81	100.00	60.00	40.00

TABLE 4-7
Heat Loss, Power Supplied (Generated), Exergy Destruction Flow Rate, Exergy Destruction Ratio (θ), Exergy Destruction to Total Exergy Input Ratio (θ^*) and Exergetic Efficiency (ζ) for Plant Components in Case 1CO1

	Component	$\dot{Q}_{Loss}$ [MW]	$\dot{W}$ [MW]	$\dot{E}_D$ [MW]	θ [%]	θ^* [%]	ζ [%]
1	Gasifier	5.51	1.10	229.50	31.71	19.02	81.85
2	Cyclones	2.44	0.00	2.29	0.32	0.19	99.02
3	Product Gas Cooler	1.80	0.00	27.91	3.86	2.31	68.05
4	Coal Hopper System	0.00	0.03	0.46	0.06	0.04	31.12
6	Quench Steam Mixing	0.00	0.00	15.01	2.07	1.24	98.56
7/8	Fines Cyclone & Gas Filter	0.00	0.00	1.71	0.24	0.14	99.08
9	Solids Conveyor & Cooler	0.91	0.00	0.50	0.07	0.04	98.86
10	Area 600: Sulfation	1.10	1.31	8.68	1.20	0.72	76.35
11	Chloride Guard	0.14	0.00	1.56	0.22	0.13	99.15
13	Air Recuperator	0.39	0.00	1.89	0.26	0.16	85.06
14	BFW Preheater	0.38	0.00	3.45	0.48	0.29	48.89
15	Trim Cooler	0.87	0.00	1.55	0.21	0.13	-
16	Booster Air Compressor	0.00	18.41	2.96	0.41	0.25	83.93
17	Zinc Ferrite System	0.00	0.00	9.72	1.34	0.81	99.03
18	Exit Gas Cooler	0.62	0.00	7.19	0.99	0.60	74.05
20	Recycle Gas Cooler	0.13	0.00	2.57	0.36	0.21	41.72
21	Recycle Gas Compressor I	0.65	1.84	0.71	0.10	0.06	61.45
23	Recycle Gas Compressor II	0.00	0.29	0.07	0.01	0.01	74.71
25	Steam Transport Losses	0.00	0.00	0.16	0.02	0.01	99.92
26	Gas Turbine/Air Compressor	39.62	(311.64)	84.94	11.73	7.04	78.58
27	Combustion Chamber	0.00	0.00	192.34	26.57	15.94	82.62
29	Superheater & Reheater	3.40	0.00	14.26	1.97	1.18	89.44
31	HP Economizer	3.29	0.00	7.52	1.04	0.62	91.90
32	Feedwater Heater 1/2	0.20	0.00	2.51	0.35	0.21	34.05
34	HP Turbine	0.00	(65.59)	6.82	0.94	0.57	90.58
37	IP Turbine	0.00	(43.91)	5.14	0.71	0.43	89.52
39	LP Turbine	0.00	(100.45)	22.48	3.11	1.86	81.71
40	Steam Seal Regulator	0.00	0.00	0.03	0.00	0.00	86.42
42	Condenser	323.01	0.00	17.02	2.35	1.41	-
43	LP Pump	0.00	0.04	0.01	0.00	0.00	68.19
46	Deaerator	0.00	0.00	0.47	0.07	0.04	94.14
47	HP Pump	0.00	4.87	1.53	0.21	0.13	68.51

4.2 DESCRIPTION OF CASE 1CO2

This case represents an economically optimal design of Case 1 at a steam high-pressure value (P_{HP} = 1515 psia) comparable with the value in the original case 1 (P_{HP} = 1600 psia). The simplified flow diagrams for this case are presented in Figures 4-3 and 4-4. The results of the simulation and thermodynamic analyses are shown in Tables 4-8, 4-9 and 4-10. The structure of the gasification island is identical in the optimal Cases 1CO1 and 1CO2. Both cases have the same gasification temperature (1920°F) and the same coal moisture at the gasifier inlet (11.12 weight percent). The major differences between these two cases are

- in the steam high-pressure value ($P_{HP} = P_{16} = P_{47}$ = 1515 psia in Case 1CO2 versus P_{HP} = 2055 psia in Case 1CO1).

- in the HRSG design as it is apparent from a comparison of Figures 4-2 and 4-4 and the values shown in Tables 4-5 through 4-10 (e.g., in addition to the components included in the HRSG of Case 1CO1, the HRSG of Case 1CO2 has an HP steam drum, a blast steam reheater and a second feedwater heater),

- in the steam turbine (pressure ratio and steam extractions), and

- in the design of the LP feedwater preheating process (Case 1CO2 has two more feedwater heaters than Case 1CO1; the steam extraction from the LP turbine is used in the FWH1 in Case 1CO2 versus the deaerator in Case 1CO1).

The design of Case 1CO2 is closer than the design of Case 1CO1 to the design of the original Case 1. The differences in performance and costs between the two optimized Cases 1CO1 and 1CO2 are discussed in section 4.5.

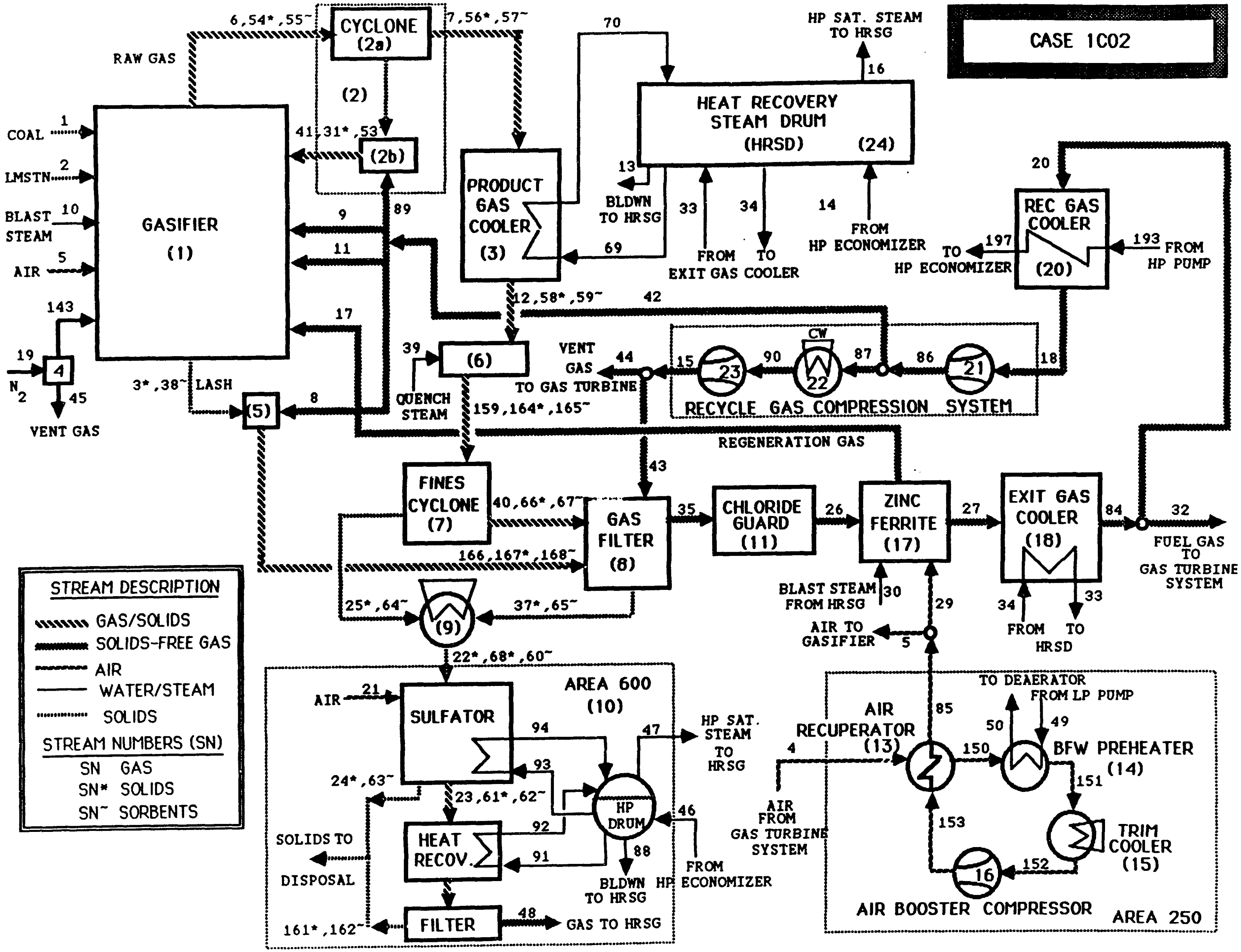

Figure 4-3. Flow Diagram of the Gasification Island in case 1C02

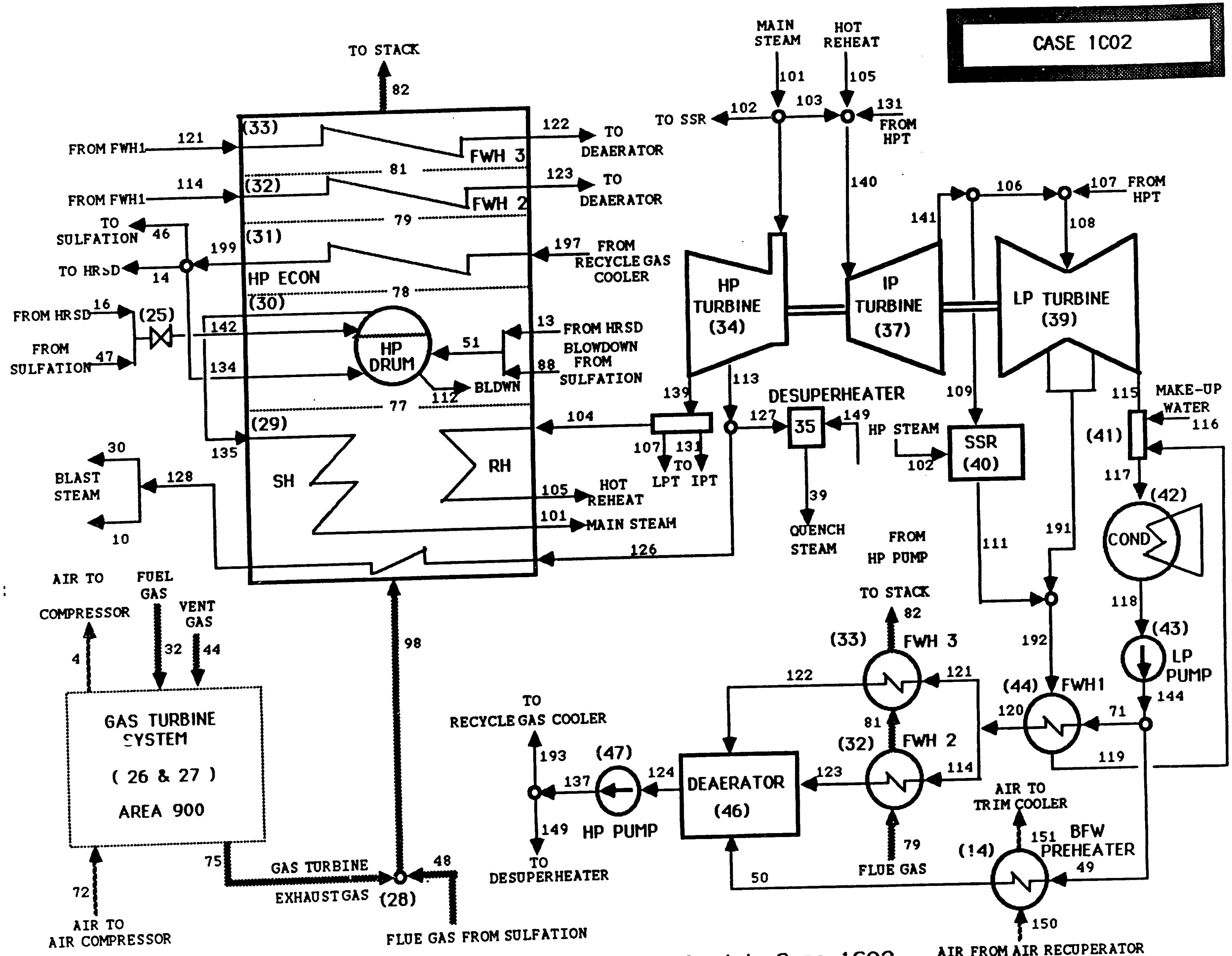

Figure 4-4. Flow Diagram of the Power Island in Case 1C02

TABLE 4-8

Mass Flow Rate, Temperature, Pressure and Flow Rates of Enthalpy, Entropy and Exergy for Each Stream of Case 1CO2

Stream No	$\dot{m}$ [lb/hr]	T [F]	P [psia]	$\dot{H}$ [MMBtu/hr]	$\dot{S}$ [MBtu/hrR]	$\dot{E}$ [MW]
1	345485	90.0	14.7	-503.74	134.72	1203.80
2	87703	90.0	14.7	-456.19	18.75	0.00
3	28031	500.0	425.0	-138.03	9.72	6.77
4	1140136	760.0	200.0	193.52	1896.78	52.48
5	1135222	650.0	450.0	160.88	1798.16	57.50
6	1611124	1920.0	400.0	-817.73	3361.49	1034.86
7	1611124	1920.0	395.0	-817.72	3363.10	1034.60
8	16115	411.0	430.0	-32.16	30.35	7.14
9	36896	411.0	430.0	-73.64	69.49	16.35
10	6897	950.0	450.0	-37.16	17.74	1.11
11	49374	411.0	430.0	-98.55	92.98	21.88
12	1611124	1133.7	385.0	-1238.23	3152.28	945.32
13	10087	597.5	1515.0	-63.23	16.66	0.51
14	1018847	597.5	1515.0	-6387.13	1682.38	51.61
15	16502	523.9	750.0	-32.30	30.98	7.51
16	1008760	597.5	1515.0	-5763.15	2195.94	130.03
17	34075	1200.0	430.0	-153.30	86.02	6.37
18	126280	330.0	285.3	-255.48	238.07	54.89
19	13094	500.0	450.0	1.40	20.20	0.64
20	126280	1000.0	295.0	-225.58	265.05	59.31
21	197485	160.0	20.0	3.96	326.47	0.76
22	29504	500.0	14.7	-141.46	10.34	10.52
23	190224	1600.0	19.8	-49.48	369.53	13.10
24	34428	150.0	14.7	-181.30	7.08	0.88
25	13260	1015.0	14.7	-29.36	8.02	32.86
26	2043491	1007.2	345.0	-3506.43	4320.67	998.50
27	2043144	1187.6	305.0	-3508.32	4373.67	987.31
29	4914	650.0	450.0	0.70	7.78	0.25
30	28935	950.0	450.0	-155.89	74.42	4.65
31	279919	1862.0	430.0	-482.45	242.08	722.28
32	1916864	1000.0	295.0	-3424.20	4023.26	900.29
33	250665	597.5	1515.0	-1432.07	545.67	32.31
34	250665	597.5	1515.0	-1571.41	413.91	12.70
35	2043353	1007.2	365.0	-3505.94	4310.51	1000.06
37	29504	494.0	14.7	-141.52	10.27	10.51
38	50587	500.0	425.0	-209.37	14.41	0.30
39	401759	744.7	450.0	-2208.79	999.40	57.03
40	2012883	1015.0	385.0	-3446.07	4241.17	988.21
41	7393	1862.0	430.0	-10.78	16.58	4.01

TABLE 4-8

Mass Flow Rate, Temperature, Pressure and Flow Rates of Enthalpy, Entropy and Exergy for Each Stream of Case 1CO2 (Cont'd)

Stream No	$\dot{m}$ [lb/hr]	T [F]	P [psia]	$\dot{H}$ [MMBtu/hr]	$\dot{S}$ [MBtu/hrR]	$\dot{E}$ [MW]
42	109778	411.0	450.0	-219.12	206.30	48.70
43	14355	524.0	750.0	-28.10	26.95	6.54
44	2147	524.0	750.0	-4.20	4.03	0.98
45	9215	90.0	14.7	0.03	15.15	0.02
46	345775	597.5	1515.0	-2167.66	570.97	17.52
47	342352	597.5	1515.0	-1955.89	745.26	44.13
48	190224	1098.0	19.6	-76.01	354.91	7.68
49	797647	120.4	23.0	-5419.51	802.74	0.45
50	797647	228.0	20.0	-5333.51	938.73	3.75
51	13510	596.2	1500.0	-84.69	22.31	0.68
53	113266	1862.0	430.0	-435.63	53.56	6.95
54	294652	1920.0	400.0	-496.46	259.67	762.85
55	115575	1920.0	400.0	-442.97	55.31	7.44
56	14733	1920.0	395.0	-24.82	12.98	38.14
57	2310	1920.0	395.0	-8.85	1.11	0.15
58	14733	1133.7	385.0	-31.74	9.49	36.68
59	2310	1133.7	385.0	-9.26	0.90	0.06
60	52897	500.0	14.7	-218.93	15.07	0.31
61	68	1600.0	19.8	-0.32	0.04	0.01
62	137	1600.0	19.8	-0.59	0.07	0.01
63	68454	150.0	14.7	-320.11	13.80	0.01
64	2079	1015.0	14.7	-8.38	0.78	0.05
65	50818	494.0	14.7	-210.39	14.41	0.29
66	1473	1015.0	385.0	-3.26	0.89	3.65
67	231	1015.0	385.0	-0.93	0.09	0.01
68	13260	500.0	14.7	-32.36	5.54	32.38
69	758095	597.5	1515.0	-4752.49	1251.81	38.41
70	758095	597.5	1515.0	-4331.08	1650.28	97.72
71	787560	120.4	23.0	-5350.97	792.59	0.45
72	5958123	90.0	14.7	18.91	9803.01	0.99
75	6736999	1098.0	15.0	-4801.56	13433.59	260.86
77	6927222	719.4	15.0	-5620.73	13248.63	137.73
78	6927222	655.1	14.9	-5744.15	13141.96	118.74
79	6927222	318.9	14.9	-6375.24	12469.95	42.05
81	6927222	291.3	14.9	-6426.01	12404.52	37.70
82	6927222	279 8	14.8	-6447.11	12377.17	35.93
84	2043144	1000.0	295.0	-3649.78	4288.31	959.60
85	1140136	650.0	450.0	161.57	1805.94	57.75
86	126280	411.0	450.0	-252.04	237.31	56.02

TABLE 4-8

Mass Flow Rate, Temperature, Pressure and Flow Rates of Enthalpy, Entropy and Exergy for Each Stream of Case 1CO2 (Cont'd)

Stream No	$\dot{m}$ [lb/hr]	T [F]	P [psia]	$\dot{H}$ [MMBtu/hr]	$\dot{S}$ [MBtu/hrR]	$\dot{E}$ [MW]
87	16502	411.0	450.0	-32.94	31.01	7.32
88	3423	597.5	1515.0	-21.46	5.65	0.17
89	7393	411.0	430.0	-14.75	13.92	3.28
90	16502	365.7	450.0	-33.20	30.70	7.29
91	47069	597.5	1515.0	-295.08	77.72	2.38
92	47069	597.5	1515.0	-268.91	102.46	6.07
93	295283	597.5	1515.0	-1851.12	487.59	14.96
94	295283	597.5	1515.0	-1686.98	642.79	38.06
98	6927222	1098.0	15.0	-4877.58	13793.28	267.79
101	1569513	1000.0	1359.3	-8457.11	3852.49	281.47
102	663	1000.0	1359.3	-3.57	1.63	0.12
103	2701	1000.0	1359.3	-14.55	6.63	0.48
104	1108245	626.6	300.0	-6154.20	2750.65	140.35
105	1108245	1000.0	282.0	-5935.91	2930.85	175.30
106	1128541	838.4	150.0	-6134.38	2996.96	150.20
107	1970	626.6	300.0	-10.94	4.89	0.25
108	1130511	838.0	150.0	-6145.31	3002.02	150.42
109	748	838.4	150.0	-4.07	1.99	0.10
111	1412	881.8	150.0	-7.64	3.77	0.19
112	15694	596.2	1500.0	-98.41	25.89	0.79
113	437590	744.7	500.0	-2406.88	1082.77	62.72
114	47556	138.9	22.5	-322.24	49.35	0.04
115	1118246	120.6	1.7	-6452.94	3098.55	18.29
116	453284	80.0	14.5	-3098.08	423.50	0.16
117	1585206	120.6	1.7	-9643.72	3537.38	18.27
118	1585206	120.4	1.7	-10770.63	1595.26	0.87
119	13676	136.4	3.3	-92.70	14.14	0.01
120	787560	138.9	22.5	-5336.42	817.28	0.73
121	740003	138.9	22.5	-5014.19	767.93	0.69
122	740003	167.0	20.0	-4993.40	801.88	1.31
123	47556	234.4	20.0	-272.22	121.91	3.01
124	1585206	228.0	20.0	-10599.57	1865.59	7.45
126	35831	744.7	500.0	-197.08	88.66	5.14
127	401759	744.7	450.0	-2208.79	999.40	57.03
128	35831	950.0	450.0	-193.05	92.15	5.75
131	18344	626.6	300.0	-101.87	45.53	2.32
134	220584	597.5	1515.0	-1382.84	364.24	11.17
135	1569513	596.2	1500.0	-8966.62	3417.97	202.14
137	1585206	231.5	1565.0	-10588.70	1870.32	9.88

TABLE 4-8

Mass Flow Rate, Temperature, Pressure and Flow Rates of Enthalpy, Entropy and Exergy for Each Stream of Case 1CO2 (Cont'd)

Stream No	$\dot{m}$ [lb/hr]	T [F]	P [psia]	$\dot{H}$ [MMBtu/hr]	$\dot{S}$ [MBtu/hrR]	$\dot{E}$ [MW]
139	1128559	626.6	300.0	-6267.00	2801.07	142.92
140	1129290	993.8	282.0	-6052.32	2983.96	177.95
141	1129290	838.4	150.0	-6138.44	2998.95	150.30
142	1351112	596.2	1500.0	-7719.05	2942.22	173.99
143	3879	500.0	450.0	0.41	5.98	0.19
144	1585206	120.4	23.0	-10770.48	1595.33	0.90
149	0	-	-	-	-	-
150	1140136	450.0	199.0	104.24	1812.78	39.85
151	1140136	138.4	198.0	16.93	1695.79	33.10
152	1140136	127.7	176.0	13.98	1700.01	31.56
153	1140136	341.9	453.0	73.64	1712.66	47.01
154	1566149	1000.0	1359.3	-8438.99	3844.24	280.87
159	2012883	1015.0	385.0	-3446.07	4241.17	988.21
161	68	1098.0	19.6	-0.34	0.04	0.00
162	137	1098.0	19.6	-0.61	0.06	0.00
164	14733	1015.0	385.0	-32.63	8.91	36.51
165	2310	1015.0	385.0	-9.31	0.86	0.05
166	16115	479.7	375.0	-31.78	30.97	7.15
167	28031	479.7	375.0	-138.22	9.52	6.74
168	50587	479.7	375.0	-209.58	14.19	0.27
170	345485	90.0	14.7	-503.74	134.72	1203.80
191	12265	205.8	3.7	-70.29	33.75	0.38
192	13676	277.1	3.7	-77.93	38.27	0.45
193	1585206	231.5	1565.0	-10588.70	1870.32	9.88
194	218401	596.2	1500.0	-1247.58	475.76	28.15
197	1585206	250.0	1533.7	-10559.25	1912.60	11.70
199	1585206	597.5	1515.0	-9937.63	2617.59	80.31

TABLE 4-9

Exergy Flow Rates of Fuel ($\dot{E}_F$), Product ($\dot{E}_P$), and Exergy Destruction ($\dot{E}_D$), Exergy Destruction Ratio (θ), Exergy Destruction to Total Exergy Input Ratio (θ^*) and Exergetic Efficiency (ζ) for Each Area and the Total Plant in Case 1CO2

Area	$\dot{E}_F$ [MW]	$\dot{E}_P$ [MW]	$\dot{E}_D$ [MW]	θ [%]	θ^* [%]	ζ [%]
Area 250: Booster Air Compressor	23.35	13.51	9.84	1.35	0.82	57.86
Area 300: KRW Gasification	1287.90	1024.25	263.65	36.08	21.86	79.53
Area 380: Recycle Gas Compression	61.44	56.22	5.22	0.71	0.43	91.50
Area 400: Gas Conditioning	200.90	182.77	18.12	2.48	1.50	90.98
Area 500: External Desulfurization	944.09	926.27	17.81	2.44	1.48	98.11
Area 600: Sulfation	36.70	26.79	9.92	1.36	0.82	72.98
Area 900: Gas Turbine System	641.21	363.93	277.28	37.94	22.99	56.76
Area 1000: HRSG	232.61	205.57	27.04	3.70	2.24	88.38
Area 1100: Steam Cycle	255.25	201.62	53.63	7.34	4.44	78.99
Total Plant Exergy Losses			36.96	5.06	3.06	
Service Station Power			11.20	1.53	0.93	
Steam Transport Losses			0.16	0.02	0.01	
Total Plant	1206.35	475.52	730.83	100.00	60.58	39.42

TABLE 4-10

Heat Loss, Power Supplied (Generated), Exergy Destruction Flow Rate, Exergy Destruction Ratio (θ), Exergy Destruction to Total Exergy Input Ratio (θ^*) and Exergetic Efficiency (ζ) for Plant Components in Case 1CO2

	Component	$\dot{Q}_{LOSS}$ [MW]	$\dot{W}$ [MW]	$\dot{E}_D$ [MW]	θ [%]	θ^* [%]	ζ [%]
1	Gasifier	5.51	1.10	229.50	31.40	19.02	81.85
2	Cyclones	2.44	0.00	2.29	0.31	0.19	99.02
3	Product Gas Cooler	1.88	0.00	31.51	4.31	2.61	65.31
4	Coal Hopper System	0.00	0.03	0.46	0.06	0.04	31.12
6	Quench Steam Mixing	0.00	0.00	14.32	1.96	1.19	98.62
7/8	Fines Cyclone &Gas Filter	0.00	0.00	1.71	0.23	0.14	99.08
9	Solids Conveyor & Cooler	0.91	0.00	0.50	0.07	0.04	98.86
10	Area 600: Sulfation	1.10	1.31	9.92	1.36	0.82	72.98
11	Chloride Guard	0.14	0.00	1.56	0.21	0.13	99.15
12	Air Extraction Cooler	0.00	0.00	0.00	0.00	0.00	0.00
13	Air Recuperator	0.39	0.00	1.89	0.26	0.16	85.06
14	BFW Preheater	0.38	0.00	3.45	0.47	0.29	48.90
15	Trim Cooler	0.87	0.00	1.55	0.21	0.13	-
16	Booster Air Compressor	0.00	18.41	2.96	0.40	0.25	83.93
17	Zinc Ferrite System	0.00	0.00	9.72	1.33	0.81	99.03
18	Exit Gas Cooler	0.62	0.00	8.10	1.11	0.67	70.78
19	HP Drum in HRSG	0.55	0.00	2.70	0.37	0.22	85.79
20	Recycle Gas Cooler	0.13	0.00	2.60	0.36	0.22	41.23
21	Recycle Gas Compressor I	0.65	1.84	0.71	0.10	0.06	61.45
23	Recycle Gas Compressor II	0.00	0.29	0.07	0.01	0.01	74.71
25	Steam Transport Losses	0.00	0.00	0.16	0.02	0.01	99.92
26	Gas Turbine/Air Compressor	39.62	(311.64)	84.94	11.62	7.04	78.58
27	Combustion Chamber	0.00	0.00	192.34	26.32	15.94	82.62
29	Superheater & Reheater	3.30	0.00	14.93	2.04	1.24	88.44
30	Blast Reheater	0.02	0.00	0.24	0.03	0.02	72.42
31	HP Economizer	2.77	0.00	8.09	1.11	0.67	89.46
32	Feedwater Heater 1	0.22	0.00	1.37	0.19	0.11	68.39
33	Feedwater Heater 2	0.09	0.00	1.16	0.16	0.10	35.06
34	HP Turbine	0.00	(67.74)	7.48	1.02	0.62	90.06
37	IP Turbine	0.00	(24.84)	2.82	0.39	0.23	89.81
39	LP Turbine	0.00	(108.99)	22.75	3.11	1.89	82.73
40	Steam Seal Regulator	0.00	0.00	0.03	0.00	0.00	88.40
42	Condenser	330.26	0.00	17.40	2.38	1.44	-
43	LP Pump	0.00	0.04	0.01	0.00	0.00	67.21
44	Feedwater Heater 3	0.07	0.00	0.15	0.02	0.01	64.92
46	Deaerator	0.13	0.00	0.62	0.09	0.05	92.27
47	HP Pump	0.00	3.54	1.12	0.15	0.09	68.50

4.3 DESCRIPTION OF CASE 1TO1

Development of Case 1TO1 focused on the overall plant thermal efficiency. This case represents the thermodynamically optimal case and demonstrates the potential for increasing the plant efficiency. By assuming (a) a constant carbon conversion ratio in the gasifier, (b) a given minimum temperature difference for steam generation and feedwater preheating, and (c) a given minimum temperature at the HRSG exit, the optimal thermodynamic efficiency of an IGCC power plant design similar to Case 1 is obtained when the following conditions are fulfilled **simultaneously:**

1. The gasification temperature is equal to the minimum value (here 1800°F) for which a constant carbon conversion ratio is assumed.

2. The coal is supplied to the gasifier with "as received" moisture.

3. The sulfator combustion gas is used to generate HP steam before exiting the sulfation at about the same temperature at which the gas turbine combustion gases exit the gas turbine. The two combustion gas streams are mixed and enter the HRSG together.

4. The steam high pressure (P_{HP}) is equal to its "thermodynamically optimum" value, which is obtained when

   (a) all HP steam generation occurs exclusively in the gasification island,

   (b) the potential for generating HP steam in the gasification island is **fully** used in all areas (including recycle gas compression and booster air compression) by not allowing any heat transfer across the high-temperature pinch point (PP-1 in Figures 4-7 through 4-10),

   (c) the HRSG is used only for steam superheating, steam reheating and feedwater preheating, **and**

   (d) any additional heat required for feedwater preheating is provided through steam extractions from the steam turbine.

   This optimal P_{HP} value is unique for a given design configuration.

(5) The minimum temperature differences during HP steam preheating and steam generation in the HRSG, recycle gas cooler and extraction air co (which was added to the booster air compression area) are about equal to

assumed minimal values. It is apparent that the overall plant efficiency increases with decreasing minimum temperature differences in these heat exchangers.

All these conditions were taken into account in developing the design for the thermodynamically optimal Case 1TO1, which is shown in Figures 4-5 and 4-6. Tables 4-9 and 4-10 contain the results of the simulation and thermodynamic analyses. The design changes are discussed in the following section.

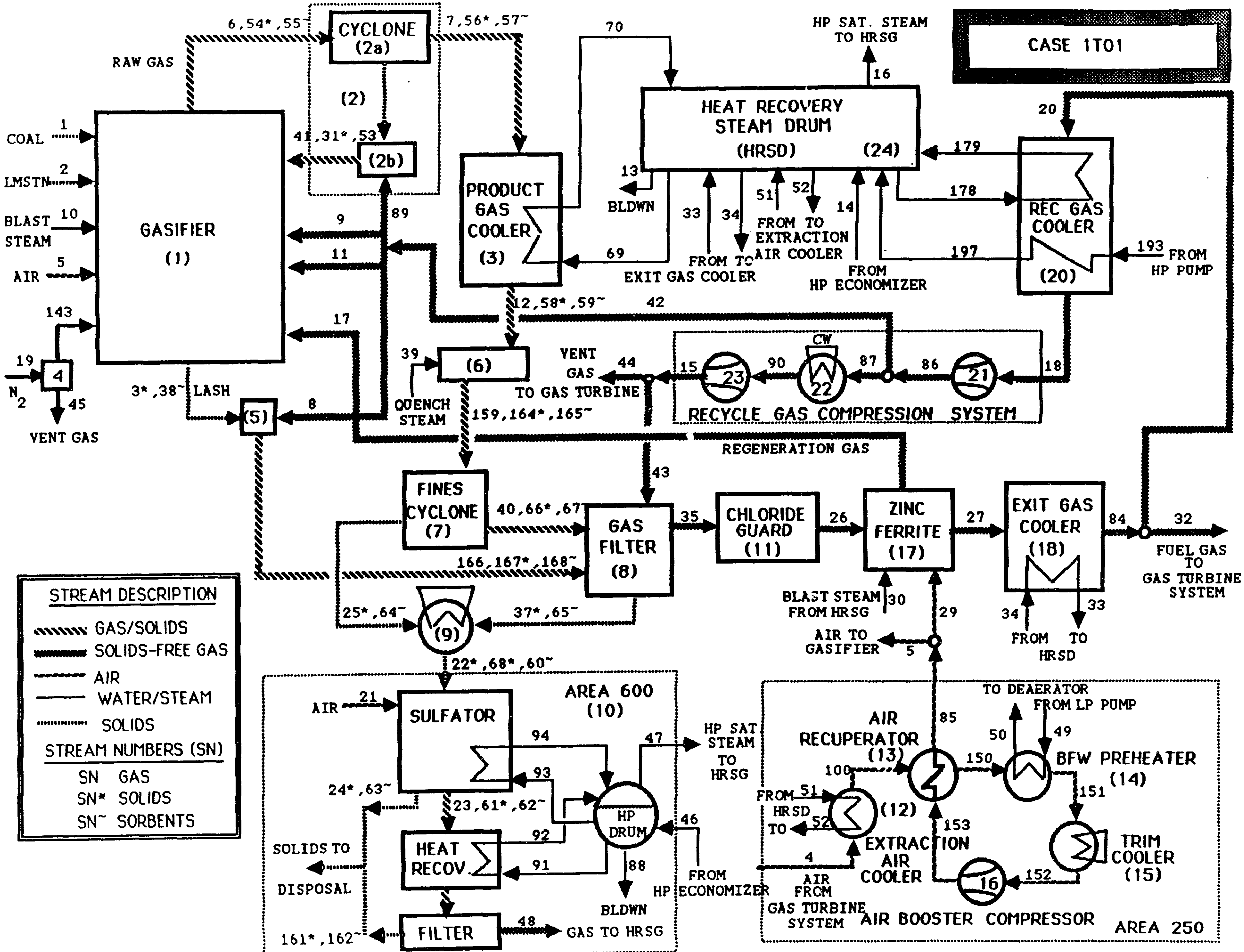

Figure 4-5. Flow Diagram of the Gasification Island in Case 1T01

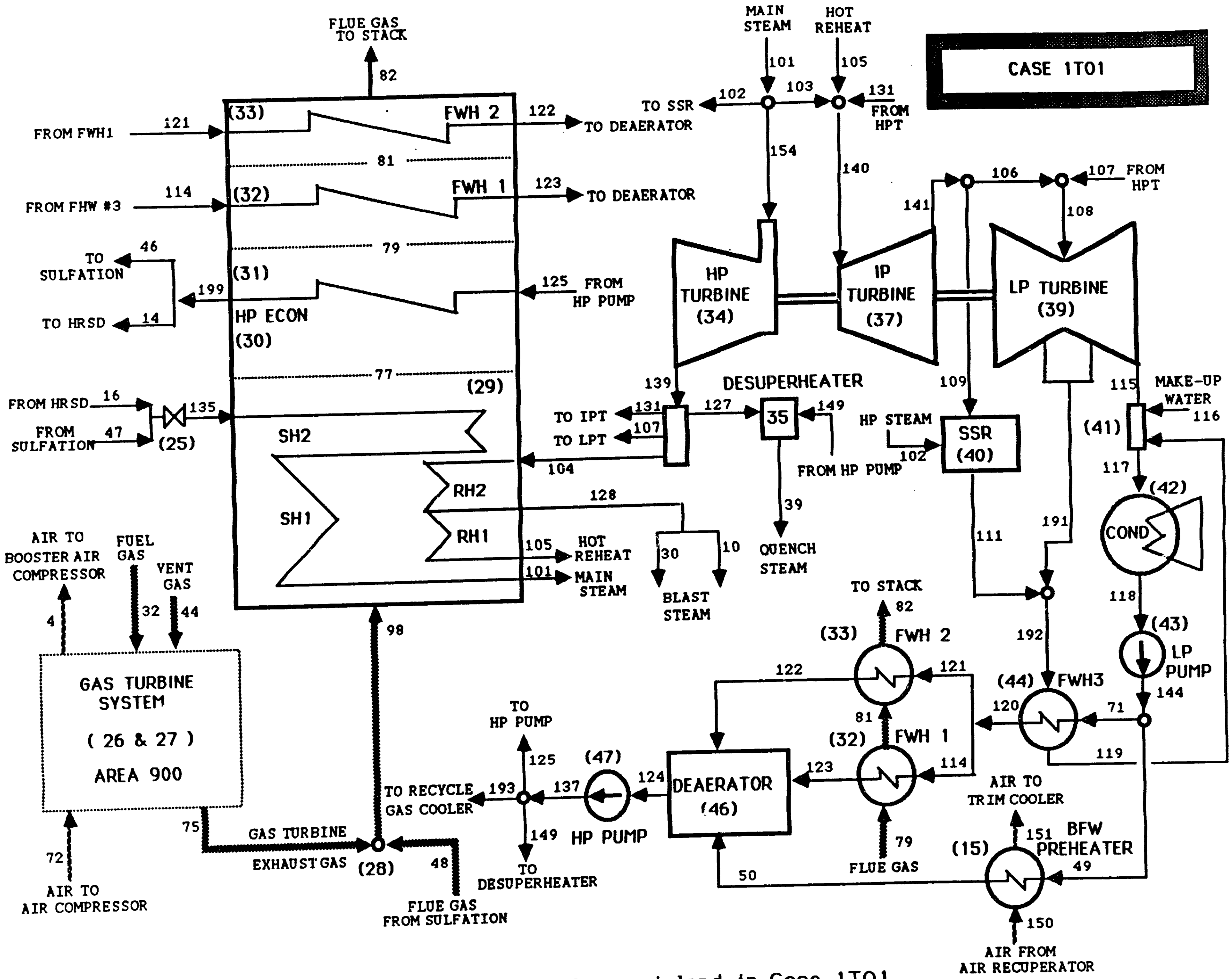

Figure 4-6. Flow Diagram of the Power Island in Case 1T01

TABLE 4-11

Mass Flow Rate, Temperature, Pressure and Flow Rates of Enthalpy, Entropy and Exergy for Each Stream of Case 1TO1

Stream No	$\dot{m}$ [lb/hr]	T [F]	P [psia]	$\dot{H}$ [MMBtu/hr]	$\dot{S}$ [MBtu/hrR]	$\dot{E}$ [MW]
1	336224	90.0	14.7	-490.24	131.11	1171.53
2	85352	90.0	14.7	-443.96	18.25	0.00
3	27279	500.0	425.0	-134.33	9.46	6.59
4	1063331	760.0	200.0	180.48	1769.01	48.94
5	1058549	650.0	450.0	150.01	1676.71	53.62
6	1534463	1800.0	400.0	-826.73	3198.90	1017.43
7	1534463	1800.0	395.0	-826.72	3200.46	1017.18
8	15683	411.0	430.0	-31.66	29.75	7.26
9	47910	411.0	430.0	-96.72	90.88	22.19
10	6695	950.0	450.0	-36.07	17.22	1.08
11	48542	411.0	430.0	-98.00	92.08	22.48
12	1534463	1179.8	385.0	-1144.47	3039.42	950.00
13	11283	649.4	2199.8	-69.79	19.45	0.72
14	1102110	649.4	2199.8	-6817.27	1899.90	69.86
15	16054	524.0	750.0	-31.79	30.36	7.63
16	1128432	649.4	2199.8	-6494.61	2379.46	143.86
17	33179	1200.0	430.0	-149.27	83.76	6.20
18	135658	330.0	285.3	-277.62	257.61	61.69
19	12743	500.0	450.0	1.36	19.66	0.62
20	135658	1000.0	295.0	-245.18	286.88	66.48
21	192192	160.0	20.0	3.86	317.72	0.74
22	28713	500.0	14.7	-137.67	10.06	10.23
23	185125	1600.0	19.8	-48.16	359.63	12.75
24	33505	150.0	14.7	-176.44	6.89	0.86
25	12904	1015.0	14.7	-28.58	7.80	31.98
26	1960011	1007.1	345.0	-3400.24	4177.39	998.58
27	1959673	1191.2	305.0	-3402.09	4228.43	987.76
29	4782	650.0	450.0	0.68	7.57	0.24
30	28174	950.0	450.0	-151.80	72.46	4.52
31	272416	1750.0	430.0	-489.38	226.83	698.51
32	1824015	1000.0	295.0	-3296.56	3857.28	893.92
33	319726	649.4	2199.8	-1840.16	674.19	40.76
34	319726	649.4	2199.8	-1977.71	551.17	20.27
35	1959878	1007.1	365.0	-3399.78	4167.54	1000.09
37	28713	494.0	14.7	-137.72	10.00	10.23
38	49231	500.0	425.0	-203.76	14.02	0.29
39	395766	645.4	450.0	-2197.67	965.57	52.83
40	1930230	1015.0	385.0	-3340.85	4099.67	987.96
41	7468	1750.0	430.0	-11.35	16.73	4.14

TABLE 4-11

Mass Flow Rate, Temperature, Pressure and Flow Rates of Enthalpy, Entropy and Exergy for Each Stream of Case 1TO1 (Cont'd)

Stream No	$\dot{m}$ [lb/hr]	T [F]	P [psia]	$\dot{H}$ [MMBtu/hr]	$\dot{S}$ [MBtu/hrR]	$\dot{E}$ [MW]
42	119603	411.0	450.0	-241.47	226.40	55.47
43	13965	524.0	750.0	-27.65	26.41	6.64
44	2089	524.0	750.0	-4.14	3.95	0.99
45	8968	90.0	14.7	0.03	14.74	0.02
46	434796	649.4	2199.8	-2689.49	749.53	27.56
47	430491	649.4	2199.8	-2477.66	907.75	54.88
48	185125	1098.0	19.6	-73.97	345.40	7.47
49	498973	120.5	34.5	-3390.19	502.17	0.29
50	498973	250.3	30.0	-3325.11	603.39	3.06
51	35145	649.4	2199.8	-217.40	60.59	2.23
52	35145	649.4	2199.8	-202.28	74.11	4.48
53	110230	1750.0	430.0	-426.76	50.89	6.14
54	286754	1800.0	400.0	-505.89	242.91	737.32
55	112477	1800.0	400.0	-434.18	52.50	6.55
56	14338	1800.0	395.0	-25.30	12.15	36.87
57	2248	1800.0	395.0	-8.68	1.05	0.13
58	14338	1179.8	385.0	-30.54	9.45	35.76
59	2248	1179.8	385.0	-8.99	0.89	0.07
60	51479	500.0	14.7	-213.06	14.66	0.30
61	66	1600.0	19.8	-0.31	0.04	0.01
62	133	1600.0	19.8	-0.58	0.06	0.01
63	66619	150.0	14.7	-311.53	13.43	0.01
64	2023	1015.0	14.7	-8.16	0.75	0.05
65	49456	494.0	14.7	-204.75	14.02	0.28
66	1434	1015.0	385.0	-3.18	0.87	3.55
67	225	1015.0	385.0	-0.91	0.08	0.01
68	12904	500.0	14.7	-31.49	5.40	31.52
69	740216	649.4	2199.8	-4578.72	1276.04	46.92
70	740216	649.4	2199.8	-4260.26	1560.85	94.37
71	1075538	120.5	34.5	-7307.55	1082.44	0.62
72	5974227	90.0	14.7	18.96	9829.51	0.99
75	6736999	1098.0	15.0	-4652.88	13427.06	259.55
77	6922125	702.3	14.9	-5501.10	13206.17	131.35
79	6922125	313.1	14.9	-6231.79	12442.50	40.23
81	6922125	286.9	14.9	-6279.89	12380.14	36.18
82	6922125	279.8	14.9	-6292.81	12363.73	35.04
84	1959673	1000.0	295.0	-3541.73	4144.16	960.41
85	1063331	650.0	450.0	150.69	1684.29	53.86
86	135658	411.0	450.0	-273.88	256.79	62.92

TABLE 4-11

Mass Flow Rate, Temperature, Pressure and Flow Rates of Enthalpy, Entropy and Exergy for Each Stream of Case 1TO1 (Cont'd)

Stream No	$\dot{m}$ [lb/hr]	T [F]	P [psia]	$\dot{H}$ [MMBtu/hr]	$\dot{S}$ [MBtu/hrR]	$\dot{E}$ [MW]
87	16054	411.0	450.0	-32.41	30.39	7.45
88	4304	649.4	2199.8	-26.63	7.42	0.27
89	7468	411.0	430.0	-15.08	14.17	3.46
90	16054	365.7	450.0	-32.67	30.09	7.42
91	59187	649.4	2199.8	-366.11	102.03	3.75
92	59187	649.4	2199.8	-340.65	124.80	7.55
93	371304	649.4	2199.8	-2296.76	640.08	23.53
94	371304	649.4	2199.8	-2137.01	782.95	47.34
95	185125	4421.9	20.0	114.53	408.66	52.53
96	66	4421.9	20.0	-0.12	0.10	0.05
97	133	4421.9	20.0	-0.45	0.10	0.04
98	6922125	1098.0	15.0	-4726.86	13777.11	266.29
100	1063331	703.4	195.0	165.13	1757.97	46.22
101	1558923	1000.0	1980.0	-8430.74	3745.79	283.58
102	659	1000.0	1980.0	-3.56	1.58	0.12
103	2682	1000.0	1980.0	-14.51	6.45	0.49
104	1139639	645.4	500.0	-6332.32	2764.46	153.54
105	1104770	1000.0	470.0	-5923.37	2856.34	183.49
106	1124929	719.7	150.0	-6182.58	2932.57	138.66
107	1956	645.4	500.0	-10.87	4.75	0.26
108	1126886	719.5	150.0	-6193.45	2937.57	138.89
109	743	719.7	150.0	-4.09	1.94	0.09
111	1402	801.1	150.0	-7.65	3.70	0.18
114	47235	195.2	33.8	-317.40	53.27	0.14
115	1049405	120.6	1.7	-6086.65	2854.44	16.69
116	446223	80.0	14.5	-3049.82	416.91	0.16
117	1574511	120.6	1.7	-9666.74	3361.72	16.79
118	1574511	120.4	1.7	-10697.96	1584.50	0.86
119	78883	192.6	11.9	-530.27	88.64	0.22
120	1075538	195.2	33.8	-7227.05	1212.96	3.19
121	1028302	195.2	33.8	-6909.65	1159.69	3.05
122	1028302	207.6	30.0	-6896.93	1178.95	3.67
123	47235	257.3	33.8	-270.01	119.55	3.35
124	1574511	250.3	30.0	-10492.38	1904.00	9.64
125	1536905	255.6	2250.0	-10226.49	1864.95	12.86
126	1139639	950.0	475.0	-6141.07	2923.75	183.93
127	395766	645.4	450.0	-2197.67	965.57	52.83
128	34869	950.0	450.0	-187.87	89.68	5.60
131	18220	645.4	500.0	-101.24	44.20	2.45

TABLE 4-11

Mass Flow Rate, Temperature, Pressure and Flow Rates of Enthalpy, Entropy and Exergy for Each Stream of Case 1TO1 (Cont'd)

Stream No	$\dot{m}$ [lb/hr]	T [F]	P [psia]	$\dot{H}$ [MMBtu/hr]	$\dot{S}$ [MBtu/hrR]	$\dot{E}$ [MW]
135	1558923	648.4	2184.8	-8972.28	3288.16	198.59
137	1574511	255.6	2250.0	-10476.72	1910.58	13.17
139	1555582	645.4	500.0	-8643.48	3773.43	209.58
140	1125673	993.9	470.0	-6039.11	2907.86	186.29
141	1125673	719.7	150.0	-6186.66	2934.51	138.75
143	3775	500.0	450.0	0.40	5.82	0.18
144	1574511	120.5	34.5	-10697.74	1584.61	0.91
149	0	0.0	0.0	0.00	0.00	0.00
150	1063331	392.0	194.0	81.87	1675.07	35.17
151	1063331	138.5	193.1	15.80	1583.41	30.58
152	1063331	127.7	176.0	13.04	1585.49	29.43
153	1063331	341.9	453.0	68.68	1597.29	43.84
154	1555582	1000.0	1980.0	-8412.67	3737.76	282.97
159	1930230	1015.0	385.0	-3340.85	4099.67	987.96
161	66	1098.0	19.6	-0.33	0.03	0.00
162	133	1098.0	19.6	-0.60	0.05	0.00
164	14338	1015.0	385.0	-31.75	8.67	35.53
165	2248	1015.0	385.0	-9.06	0.84	0.05
166	15683	479.6	375.0	-31.28	30.36	7.28
167	27279	479.6	375.0	-134.52	9.26	6.56
168	49231	479.6	375.0	-203.96	13.81	0.26
170	336224	90.0	14.7	-490.24	131.11	1171.53
176	6922125	1054.5	15.0	-4813.43	13721.22	249.92
177	135658	704.4	289.1	-259.74	275.97	63.97
178	33346	649.4	2199.8	-206.27	57.48	2.11
179	33346	649.4	2199.8	-191.92	70.31	4.25
191	77481	299.0	13.2	-440.89	206.78	4.36
192	78883	307.9	13.2	-448.54	210.96	4.47
193	37605	255.6	2250.0	-250.22	45.63	0.31
197	37605	649.4	2199.8	-232.61	64.83	2.38
199	1536905	649.4	2199.8	-9506.76	2649.44	97.42

TABLE 4-12

Exergy Flow Rates of Fuel, Product, and Exergy Destruction, Exergy Destruction Ratio (θ), Exergy Destruction to Total Exergy Input Ratio (θ^*) and Exergetic Efficiency (ζ) for Each Area and the Total Plant in Case 1TO1

Area	$\dot{E}_F$ [MW]	$\dot{E}_P$ [MW]	$\dot{E}_D$ [MW]	θ [%]	θ^* [%]	ζ [%]
Area 250: Booster Air Compressor	19.52	12.29	7.23	1.05	.62	62.30
Area 300: KRW Gasification	1262.38	1022.34	240.05	34.55	20.45	80.99
Area 380: Recycle Gas Compression	69.08	63.11	5.98	0.86	0.51	91.35
Area 400: Gas Conditioning	195.70	176.89	18.82	2.71	1.60	90.39
Area 500: External Desulfurization	936.86	920.62	16.24	2.34	1.38	98.27
Area 600: Sulfation	35.72	27.60	8.12	1.17	0.69	77.26
Area 900: Gas Turbine System	636.19	360.41	275.78	39.70	23.49	56.65
Area 1000: HRSG	231.98	208.95	23.03	3.32	1.96	90.07
Area 1100: Steam Cycle	257.08	205.47	51.61	7.43	4.40	79.92
Total Plant Exergy Losses			36.92	5.31	3.14	
Service Station Power			10.77	1.55	0.92	
Steam Transport Losses			0.15	0.02	0.01	
Total Plant	1174.05	479.35	694.70	100.00	59.17	40.83

TABLE 4-13
Heat Loss, Power Supplied (Generated), Exergy Destruction Flow Rate, Exergy Destruction Ratio (θ), Exergy Destruction to Total Exergy Input Ratio (θ^*) and Exergetic Efficiency (ζ) for Plant Components in Case 1TO2

	Component	$\dot{Q}_{LOSS}$ [MW]	$\dot{W}$ [MW]	$\dot{E}_D$ [MW]	θ [%]	θ^* [%]	ζ [%]
1	Gasifier	5.22	1.07	219.15	31.55	18.67	82.28
2	Cyclones	1.85	0.00	1.79	0.26	0.15	99.15
3	Product Gas Cooler	1.42	0.00	20.90	3.01	1.78	69.42
4	Coal Hopper System	0.00	0.03	0.45	0.07	0.04	31.12
5	Lash Transport & Mixing	0.00	0.00	0.03	0.00	0.00	99.77
6	Quench Steam Mixing	0.00	0.00	15.12	2.18	1.29	98.54
7/8	Fines Cyclone & Gas Filter	0.00	0.00	1.67	0.24	0.14	99.07
9	Solids Conveyor & Cooler	0.88	0.00	0.49	0.07	0.04	98.86
10	Area 600: Sulfation	1.07	1.27	8.12	1.17	0.69	77.26
11	Chloride Guard	0.14	0.00	1.51	0.22	0.13	99.15
12	Air Extraction Cooler	0.07	0.00	0.47	0.07	0.04	82.82
13	Air Recuperator	0.37	0.00	1.03	0.15	0.09	90.70
14	BFW Preheater	0.29	0.00	1.83	0.26	0.16	60.19
15	Trim Cooler	0.81	0.00	1.15	0.16	0.10	-
16	Booster Air Compressor	0.00	17.17	2.76	0.40	0.23	83.93
17	Zinc Ferrite System	0.00	0.00	9.38	1.35	0.80	99.06
18	Exit Gas Cooler	0.61	0.00	6.86	0.99	0.58	74.93
20	Recycle Gas Cooler	0.14	0.00	0.59	0.08	0.05	87.76
21	Recycle Gas Compressor I	0.70	2.00	0.77	0.11	0.07	61.46
23	Recycle Gas Compressor II	0.00	0.29	0.07	0.01	0.01	74.71
25	Steam Transport Losses	0.00	0.00	0.15	0.02	0.01	99.92
26	Gas Turbine/Air Compressor	37.31	(311.64)	83.02	11.95	7.07	78.96
27	Combustion Chamber	0.00	0.00	192.76	27.75	16.42	82.55
29	Superheater & Reheater	3.41	0.00	14.37	2.07	1.22	89.33
31	HP Economizer	3.21	0.00	6.57	0.95	0.56	92.79
32	Feedwater Heater 1	0.21	0.00	0.84	0.12	0.07	79.18
33	Feedwater Heater 2	0.06	0.00	0.52	0.07	0.04	54.80
34	HP Turbine	0.00	(66.57)	6.82	0.98	0.58	90.70
37	IP Turbine	0.00	(42.55)	4.98	0.72	0.42	89.52
39	LP Turbine	0.00	(96.35)	21.48	3.09	1.83	81.77
40	Steam Seal Regulator	0.00	0.00	0.03	0.00	0.00	86.11
42	Condenser	302.22	0.00	15.92	2.29	1.36	-
43	LP Pump	0.00	0.07	0.02	0.00	0.00	67.31
44	Feedwater Heater 3	0.36	0.00	1.68	0.24	0.14	60.43
46	Deaerator	0.10	0.00	0.44	0.06	0.04	95.68
47	HP Pump	0.00	5.10	1.57	0.23	0.13	69.21

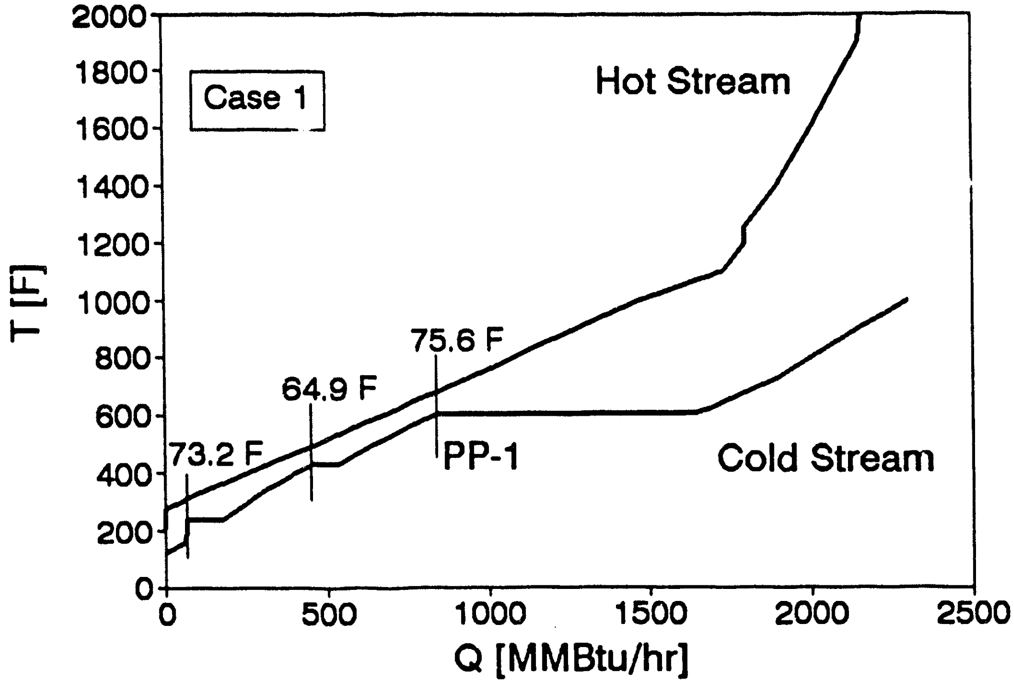

Figure 4-7. Composite Curves for the Heat Exchanger Network of the IGCC Plant in Case 1.

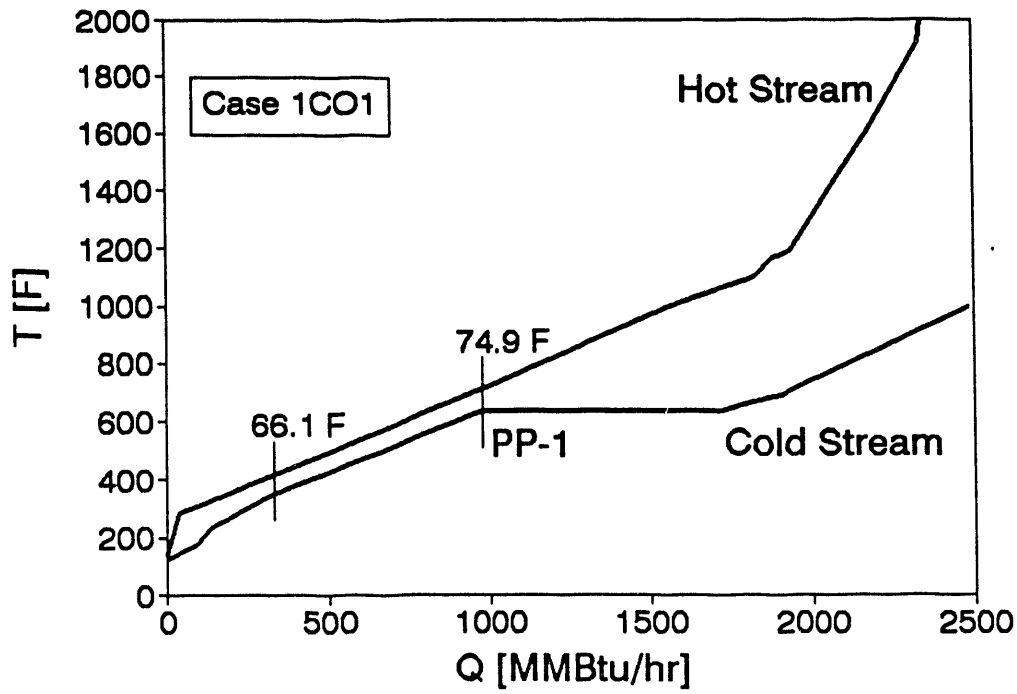

Figure 4-8. Composite Curves for the Heat Exchanger Network of the IGCC Plant in Case 1CO1.

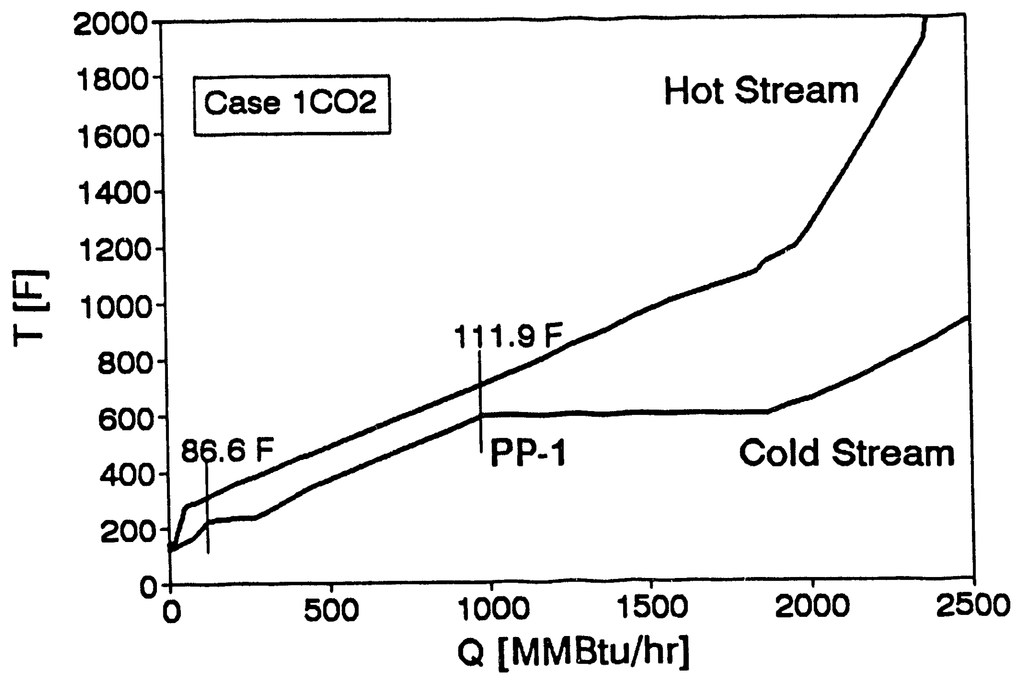

Figure 4-9. Composite Curves for the Heat Exchanger Network of the IGCC Plant in Case 1CO2.

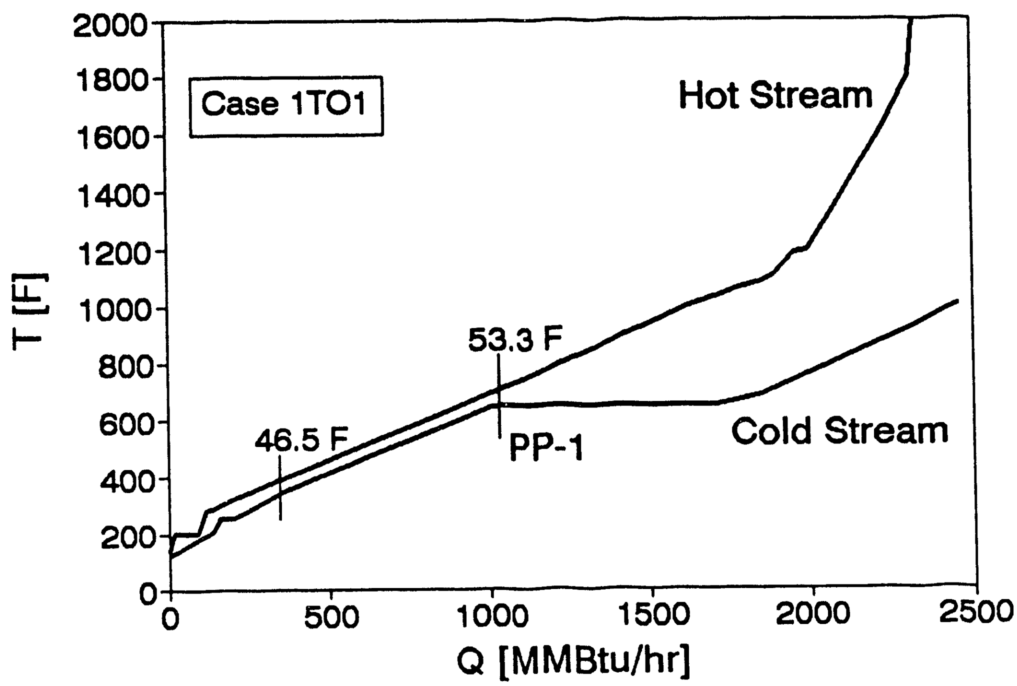

Figure 4-10. Composite Curves for the Heat Exchanger Network of the IGCC Plant in Case 1TO1.

4.4 DISCUSSION OF THE DESIGN CHANGES

This section discusses the major design changes considered in each plant area for the optimized Cases 1CO1, 1CO2 and 1TO1 as compared with the original Case 1. The pinch analysis [10] was used to identify design changes in the heat exchanger network that would improve the overall plant efficiency by avoiding heat transfer across the pinch points in the composite curves (Figures 4-7 through 4-10). Most of the identified design changes were incorporated in the thermodynamically optimal design of Case 1TO1. Some possible process modifications, however, were not incorporated in the designs of Cases 1CO1 and 1CO2 because they could not be justified economically. Our studies confirmed that the usefulness of pinch analysis techniques in IGCC power plant design studies is very limited when the objective is minimization of the electricity cost. The exergy analysis and the thermoeconomic evaluation techniques are much more powerful tools for the design optimization of power generating plants than the pinch analysis techniques. The recommendations given in the following refer to design changes to be conducted in the original Case 1 design.

4.4.1 Area 250 – Air Booster Compression

Area 250 consists of the air recuperator, BFW preheater, trim cooler and booster air compressor. As indicated in Reference [1], the initial design of this area was close to the thermoeconomic optimum. The only change considered in the cost optimal Cases 1CO1 and 1CO2 compared with the original Case 1 refers to the pressure level of the feedwater, which is preheated in the BFW preheater. By preheating LP (low-temperature) feedwater (Cases 1CO1, 1CO2, and 1TO1) instead of HP feedwater (original Case 1) both the performance and the cost effectiveness of the plant increase because the amount of heat rejected to the environment in the trim cooler decreases when the temperature difference T_{49}-T_{151} (Figures 2-1, 4-1, 4-3, and 4-5) is assumed constant.

For the thermodynamic optimum, however, an additional change is required in the original Case 1 due to some heat transfer across the pinch point that is formed by the high-pressure vaporization process. This pinch point is indicated as PP-1 in the heat transfer composite curves for Cases 1CO1, 1CO2 and 1TO1, respectively (Figures 4-7 through 4-10). Therefore, in Case 1TO1 and in accordance with criterion 4(b) presented in section 4-3, a heat exchanger called "extraction air cooler" was added to this area. This heat exchanger cools the air coming from the gas turbine system down to a temperature (T_{100}) which corresponds to the temperature ($T_{H,PP1}$) on the hot composite curve at PP-1 in Figure 4-10.

However, the MWK capital cost estimates indicated that using this cooler is not cost effective because of the relatively large total surface area required to cool the air in this cooler and the subsequent air recuperator. Therefore, the extraction air cooler was used only in the design of the thermodynamically optimal Case 1TO1.

Recommendation for Area 250: Increase the size of the BFW preheater and preheat LP (low-temperature) instead of HP (high-temperature) feedwater.

4.4.2 Area 300 - KRW Gasification

Area 300 contains the following components: pressurization air compressor, pressurization air receiver, coal feed vent filter, feed surge drum, feed pressurization hopper, feed hopper, coal feeder, gasifier, ash feeder and cyclone. In the Case 1 optimization studies significant changes were considered only in the gasifier. The performance of the remaining components was assumed to be constant; only their size was varied with changing mass flow rates. For the optimization we assumed that an increase in the moisture of the coal feed affects the investment costs of the coal feeding system only through the increased coal flow rate. The two major independent parameters of the gasifier considered in the design optimization are the coal moisture content at the gasifier inlet and the gasification temperature. These parameters are discussed below.

Coal Moisture at the Gasifier Inlet

The high-temperature gas cleanup process used in all cases discussed here requires that the moisture content of the gas supplied to the zinc ferrite system is about 30 percent by volume. This is achieved by extracting steam (stream 39) from the HP turbine and mixing it with the gas exiting the product gas cooler (stream 12). This mixing process has the additional advantage of cooling the gas to a temperature of 1,015°F.

The undesired effect of the mixing process refers to the extremely high exergy destruction, which is greater than 17 MW in the Original Case 1.* A reduction in the amount of steam mixed with the gas results in (a) a larger mass flow rate to be expanded in the steam turbine, and, consequently, more electric power generated in the steam turbine generator, and (b) a lower temperature of stream 12 (assuming a constant temperature (T_{159}) after the mixing) with the consequence that more steam can be generated in the product gas cooler.

In addition, the coal, before being fed into the gasifier in the original Case 1, is dried using the high-temperature sensible heat of the combustion gas exiting the sulfation area. This heat, however, could be used elsewhere to generate more HP steam and preheat

*The exergy destruction in the mixing process represents an "energy waste" which is easily detected in an exergy analysis but cannot be identified through an energy balance.

feedwater. Finally, the quench steam (stream 39) is cooled in the desuperheater before being mixed with the gas (stream 12). This desuperheating process adds unnecessarily to the total exergy destruction.

Thus, the combination of (a) the requirement of the high-temperature desulfurization process for a high moisture content of the gas to be cleaned, and (b) the coal drying process, which removes moisture from the gas exiting the gasifier, results in the following performance penalties in the original Case 1:

- The valuable heat content of the combustion gas exiting the sulfation area (stream 48) is used counterproductively. Instead of being used to generate steam and preheat feedwater, it is used to reduce the moisture content of the gasification gas (stream 6), when, in a subsequent step, moisture is added to this gas at a very high cost.

- A relatively high steam flow rate is extracted from the HP steam turbine and is finally expanded in the gas turbine. Since the pressure ratio in the latter is significantly lower than the combined pressure ratio in the IP and LP steam turbines, this steam extraction decreases the electric power generation in the total plant.

- The potential for generating steam in the product gas cooler is reduced when the temperature of the quench steam decreases since the condition of stream 159 remains constant.

- The exergy destruction of the quench steam desuperheating and, particularly, the mixing process of streams 12 and 39 is significant.

From the above discussion it becomes apparent that eliminating the coal drying process (a) allows the heat of the sulfation combustion gas to be used in the HRSG, (b) increases the power production in the steam turbines, (c) increases the steam generation in the product gas cooler, and (d) decreases the exergy destruction in the desuperheater and quench steam mixing process. These benefits outweigh the increase in the exergy destruction in the gasifier.

In addition to the positive effects discussed above, increasing the coal moisture at the gasifier inlet results in the two additional advantages: (1) A decreased mass flow rate of blast steam (stream 10) needs to be supplied to the gasifier per coal mass unit. Thus, more steam is left to be expanded in the IP and LP turbines. (2) The reduction in the mass flow rate of the blast steam leads to a higher per pass in-bed sulfur removal in the gasifier. This results in a reduction of the size and the O&M costs in the zinc ferrite system.

In the optimization studies, various values of the coal moisture at the gasifier inlet were considered. These studies concluded that both the economic and the thermodynamic optimal points are achieved when the coal is supplied to the gasifier with the as received moisture of 11.12 weight percent. Therefore, the optimized Cases 1CO1, 1CO2 and 1TO1 do not contain a coal drying step. For optimization purposes it was assumed that the reliability of coal feeding is not affected by this decision.

Recommendation for Coal Moisture: Supply the coal to the gasifier with the highest allowable moisture content which is dictated by the reliability of the coal feeding process. Should the actual coal moisture exceed this value, use a low-temperature flue gas extraction from the HRSG to dry the coal to the specified maximum moisture value. Some changes in the plant layout will be required to accommodate these recommendations.

Gasification Temperature

With increasing gasification temperature, the exergy destruction in the gasification system increases (thus, the exergetic efficiencies of the gasifier and the product gas cooler decrease) while the amount of steam generated in the product gas cooler also increases. The economically optimal gasification temperature depends on the overall plant design and is affected by the interaction between gasification island and power island. The thermoeconomic optimization has shown that the best gasification temperature is about 1920°F (Cases 1CO1 and 1CO2).

The thermoeconomic evaluation reported in Reference [1] concluded, based on the average cost per product exergy unit for the gasifier (c_P), that the cost optimal gasification temperature for Case 1 is between 1850°F and 1900°F. If the relative cost difference d, Equation 3-7, for the gasifier had been used instead of c_P to predict the cost optimal gasification temperature, the recommendation would have been in the range of 1900°F-1950°F, i.e., much closer to the optimal value of 1920°F. This effect is shown in the Appendix, Figure A-2. The current work has indicated that in general the relative cost difference (d) calculated for the gasifier accurately predicts the cost optimal gasification temperature for various design configurations.

The thermodynamically optimal gasification temperature is apparently equal to the lowest temperature for which a constant carbon conversion ratio may be assumed. This temperature was set to 1800°F in the present study. Case 1TO1, therefore, assumes this gasification temperature.

Recommendation for the Gasification Temperature: The economically optimal value depends on the overall plant design. The relative cost difference d, Equation 3-7, for the gasifier should be used to obtain a good and fast estimate of the cost optimal value of the gasification temperature.

4.4.3 Areas 360/380 - Heat Recovery and Recycle Gas Compression

These areas consist of the product gas cooler, heat recovery steam drum, recycle gas cooler, recycle gas compressor and recycle gas receiver. In the optimization, changes were considered in the product gas cooler and the recycle gas cooler. These changes are discussed below.

Product Gas Cooler

The product gas cooler cannot be optimized independently because its main operating parameters depend on decisions made with respect to other plant parameters. The inlet gas temperature depends on the gasification temperature (T_6); the gas temperature at the outlet is a function of the mass flow rate ($\dot{m}_{39}$) and temperature (T_{39}) of the quench steam used to provide a gas mixture (stream 159) with 30 percent by volume moisture content ($x_{H_2O,159}$) at T_{159} = 1015°F; the temperature on the steam side depends on the value of the steam high pressure (P_{HP}).

The thermoeconomic optimization indicated that it is always thermodynamically efficient and cost effective to adjust the performance and size of the product gas cooler in this way. In all cases considered here, the minimum temperature difference in this cooler was above 520°F.

Recommendation for the Product Gas Cooler: Adjust its size and effectiveness based on the following variables: T_6, T_{39}, $\dot{m}_{39}$, T_{159}, $x_{H_2O,159}$, and P_{HP}.

Recycle Gas Cooler

The following options were considered for the recycle gas cooler (RGC):

1. RGC preheats the high-temperature feedwater that is supplied to the heat recovery steam drum (HRSD).

2. RGC preheats the low-temperature feedwater that subsequently enters the HP economizer.

3. Part of RGC generates HP steam, and the remaining part preheats the high-temperature feedwater that is supplied to the HRSD.

4. The cooling of the recycle gas is obtained through spraying liquid water into the gas.

5. Cooling water is used exclusively to cool the recycle gas from T_{20} to T_{18}.

Among all heat exchangers, the recycle gas cooler has the highest value of investment cost to product exergy ratio, r_k, Equation 3-9. The relatively high investment cost associated with the RGC explains why the economically best options were found to be 5 and 4 above.* Both options, however, were rejected because of operating reasons: option 5 would cause undesirable water condensation in the recycle gas; option 4 was rejected because the additional water supplied to the recycle gas would disturb the gasifier operation.

Option 3 represents the thermodynamic optimum and was therefore used in Case 1TO1. This option avoids heat transfer across the pinch point PP-1 in Figures 4-7 through 4-10 and is in accordance with the criterion 4(b) in section 4-3.

The cost optimal Cases 1CO1 and 1CO2 employ option 2, whereas the original Case 1 uses option 1. A design configuration based on option 3, which would allow little heat transfer across the pinch point 1, could also be acceptable from the thermoeconomic viewpoint.

Recommendation for the Recycle Gas Cooler: Decrease the cooler surface area by preheating the relatively low-temperature feedwater.

The gas recycling process increases the total plant investment cost and decreases the overall plant efficiency. The recycle gas is mainly used to assure proper fluidization in the gasifier bed. The size of the recycle system was selected by MWK to maintain proper fluidization at reduced plant capacity. Future optimization studies for Case 1 should consider options which would allow for either elimination or reduction of the recycle gas flow rate. These options might be more cost effective than the design considered here. Consideration of these options in the present study would exceed the scope of work.

General Recommendation for the Gas Recycling Process: In future optimization studies, investigate the economic feasibility of design options allowing elimination or equipment size reduction for this process.

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*This is an interesting example in the analysis of the total heat exchanger network where the economically optimal solution is obtained when in a heat exchanger all heat is transferred across the pinch point. Solutions such as this cannot be predicted by the pinch analysis techniques discussed in Reference [10].

4.4.4 Area 400 - Gas Conditioning

This area contains the following plant components: Fines cyclone, cyclone fines lockhopper, cyclone fines depressurization lockhopper, cyclone lockhopper filter, gas filter, filter depressurization lockhopper, solids conveyor and chloride guard drums. Only changes in the size of these components were considered in the optimization process. Thus, no specific recommendations are made.

4.4.5 Area 500 - External Desulfurization

This area consists of the zinc ferrite reactors, guard cyclone, guard drum and exit gas cooler. Performance changes were considered only for the exit gas cooler. Since the gas temperatures at the inlet and outlet of this cooler are almost constant, the performance and investment cost of the cooler are mainly affected by the steam high-pressure value (P_{HP}).

The value of the investment cost to product exergy ratio, r_k (Equation 3-9) is lower for the exit gas cooler than for the recycle gas cooler. Therefore, in all optimal cases, the recycle gas (stream 20) was extracted at the exit gas cooler outlet instead of the zinc ferrite outlet. If the recycle gas cooler is used only for feedwater preheating, this change in the design structure allows more steam to be generated in the gasification island.

Recommendation for the Exit Gas Cooler: Let the entire clean gas which exits the zinc ferrite unit, flow through the exit gas cooler, and then extract the recycle gas stream.

The decrease in the flow rate of the blast steam supplied to the gasifier leads to an increase in the sulfur removal in the gasifier from 86.5 percent in the original Case 1 to 91.4 percent in all optimized Cases 1CO1, 1CO2 and 1TO1. Since the investment and O&M costs and the performance penalties associated with the operation of the zinc ferrite unit are significant, the economic feasibility of design options permitting elimination of the external desulfurization step should be investigated. Consideration of these options in the present study would exceed the scope of work. The general design of Case 1 without an external desulfurization step would be similar to the design of Case 6 in Reference [1].

The use of zinc titanate instead of zinc ferrite in the external desulfurization area could improve the efficiency and cost effectiveness of the total plant. The zinc titanate does not require addition of steam. Thus, the heat content of the fuel gas to the gas turbine would be increased and the steam flow rate to be expanded in the steam turbines would increase. At present, however, more test data are required to study the economic feasibility of zinc titanate.

General Recommendation for the External Desulfurization: In future optimization studies, investigate the economic feasibility of design changes aimed at either making redundant the external desulfurization step in Case 1 or using an external bed sorbent which does not require addition of steam.

4.4.6 Area 600 - Sulfation

The sulfation area contains the LASH receiver, screw feeder, sulfation air compressor, sulfator, sulfator solids screw cooler, solids disengager, sulfator heat recovery and sulfator steam drum. In the optimization, the performance of all components in this area except the sulfator heat recovery was assumed to remain constant. The surface area of the heat recovery was varied with varying combustion gas temperature (T_{48}) at the sulfation exit. The following options were considered for this gas after leaving the sulfation area:

1. The gas is used to dry the coal to a residual moisture content in the range of 4.98-9.50 weight percent.

2. The gas is filtered and mixed with the gas turbine exhaust gas. The mixture is sent to the HRSG.

3. The gas is supplied to the HRSG and mixed with the gas turbine exhaust gas at an intermediate HRSG position (e.g., between the superheater/reheater and the vaporization section, or between the vaporization and the HP economizer section).

Both the thermodynamic and the economic optima are achieved when option 2 is used. When the gas streams from the gas turbine and the sulfation are mixed (regardless of the position at which this occurs) both streams should have about the same temperature to avoid unnecessary exergy destruction in the mixing. This can be achieved by adjusting the combustion gas temperature (T_{48}) at the sulfation exit.

Recommendation for the Sulfation Area: Increase the heat recovery surface area to cool the combustion gas to about the temperature of the gas turbine exhaust gas. Add a filter to remove the solids from the combustion gas exiting the sulfation and mix this gas with the gas exiting the gas turbine system.

4.4.7 Area 900 - Gas Turbine System

In all optimization runs, the same gas turbine system with constant performance was assumed. Therefore, no recommendation, in addition to the one included in Reference

[1], which is repeated below, can be made here. The performance of the GE MS7001F gas turbine assumed in Reference [1] and in the optimization studies here corresponds to the MS7001F capabilities in October 1989. GE also supplied cost and performance data referring the MS7001F capabilities in October 1991. The effect of this change in the gas turbine performance on COE and plant efficiency is significant and is discussed in section 5.4. An increase in the gas turbine firing temperature leads in general to an increase in the efficiency of the gas turbine system. An associated benefit is the increase in the gas turbine exhaust temperature which results in higher temperature pinches in the HRSG. This represents a significant advantage for the HRSG operation when the gas turbines operate at partial load.

Recommendation for the Gas Turbine System: Study the economic feasibility of developing and using a gas turbine system with a reheat stage. Improve the gas turbine design to permit higher firing temperatures and higher system efficiencies.

4.4.8 Area 1000 - Heat Recovery Steam Generator System

In the original Case 1, the HRSG uses the heat of the exhaust flow from the combustion turbine to produce steam for both power production in the steam turbine and for process steam use in the gasification island. In all optimal cases, the HRSG also uses the heat of the exhaust flow from the sulfation system.

In the optimization studies, the HRSG design was adjusted according to the steam generation in the gasification island to provide steam superheating and reheating as well as feedwater preheating. HRSG designs with and without a steam drum were studied. At relatively low values of steam high pressure (P_{HP}), a steam drum is required in the HRSG. At P_{HP} values above 1800-1900 psia, steam generation is not necessary or possible in the HRSG and, thus, the steam drum becomes redundant. Reducing the number of steam drums in the total IGCC plant has a positive effect on the total investment costs without any associated efficiency penalties. These considerations are reflected in the designs of Cases 1CO1 (P_{HP} = 2055 psia; no drum in the HRSG), 1 CO2 (P_{HP} = 1515 psia; only an HP steam drum is considered in the HRSG) and 1TO1 (P_{HP} = 2200 psia; no drum in the HRSG).

The steam high pressure value (P_{HP}) is a very important parameter for the optimization of an IGCC power plant. This variable determines the flow rate of steam generated in the gasification island and the pinch temperatures in several heat exchangers (including the HRSG). As Cases 1CO1 and 1CO2 indicate, economically attractive solutions can be found at both relatively high and low P_{HP} values. The thermodynamic optimum will always be at high P_{HP} values. Generation of IP steam in the HRSG is not cost effective for the IGCC plants considered here.

Recommendation for the HRSG: Adjust the HRSG design to optimize the interaction between the gasification and power islands by keeping the steam high pressure (P_{HP}) value variable. If the P_{HP} value is relatively high, do not consider steam generation in the HRSG. Avoid any IP or LP steam generation in the HRSG.

4.4.9 Area 1100 – Steam Turbine System

In the optimization procedure, the design of this system was continuously adjusted to reflect changes in both the gasification island and the HRSG. When the steam high pressure (P_{HP}) value is relatively high, the design of the HP turbine section should be modified so that the HP exhaust steam conditions allow this steam to be used as quench steam (stream 39) and blast steam (streams 10 and 30). Then, no steam extraction in the HP turbine is necessary. This change was made in Cases 1CO1 and 1TO1.

In general, a decrease in the pressure at the outlet of the LP pump (i.e., a decrease in the deaerator operating pressure) leads to an increase in the overall plant efficiency and to a decrease in the COE. Therefore, this pressure should be kept at its lowest permissible value. Cases 1CO1 and 1CO2 use a deaerator operating pressure of 20 psia.

The design of the feedwater heaters 1, 2 and 3 is adjusted to accomplish, together with the BFW preheater in the air booster compression area, feedwater preheating up to the deaerator operating pressure. Case 1CO1 uses only one feedwater heater (located in the HRSG) compared with two feedwater heaters in the original Case 1 and Cases 1CO2 and 1TO1. All optimized cases use one LP steam extraction to preheat feedwater. This extraction is used directly in the deaerator in Case 1CO1 whereas a third feedwater heater is used in Cases 1CO2 and 1TO1.

Recommendations for the Steam Turbine System:

1. If the P_{HP} value permits, adjust the pressure at the HP turbine exhaust to supply quench steam to the gasification island directly.

2. Keep the deaerator operating pressure as low as possible.

3. In an optimal IGCC design, at least one LP steam extraction might be required to preheat feedwater.

4.5 COMPARISON OF CASES 1, 1CO1, 1CO2 AND 1TO1

Tables 4-14 through 4-21 summarize the major results of the material, performance and cost comparisons of Cases 1, 1CO1, 1CO2 and 1TO1. Among these cases, Case

1TO1 has the smallest coal flow rate (as received) and the highest cold gas efficiency whereas Cases 1CO1 and 1CO2 have the largest as received coal flow rate and the lowest cold gas efficiency. The differences in the cold gas efficiency can be explained by the differences in the gasification temperature. The sulfur removal in the gasifier is 91.4 percent for the three optimized cases and 86.5 percent for the original Case 1. The sulfur removal in the gasification island is the same, 99.4 percent, for all four cases.

Case 1CO1 has the largest net power output (482.5 MW) while the original Case 1 has the smallest output (458.4 MW). As expected, Case 1TO1 possesses the lowest net plant heat rate (8,181 Btu/kWh). Case 1CO1 is the next lowest (8,351 Btu/kWh) whereas the original Case 1 has the highest net plant heat rate (8,595 Btu/kWh). The difference in the overall plant thermal efficiency between Cases 1CO1 and 1CO2 is mainly caused by the difference in the steam high pressure value.

Figure 4-11 compares the temperature differences in the composite curves for Cases 1, 1CO1 and 1TO1. These temperature differences determine the costs associated with the total heat exchanger network. As expected Case 1TO1 has the smallest temperature differences in the heat exchanger network while the original Case 1 has the largest differences. The temperature differences in Cases 1CO1 and 1CO2 are very close to each other.

Because of the net power output differences among the cases, the capital cost network comparison (Tables 4-19 and 4-20) is made in terms of $/kW. The differences in the total capital requirement (TCR) among Cases 1, 1CO1 and 1CO2 (1335-1346 $/kW) are small (less than one percent). The highest TCR is for Case 1TO1 at 1421 $/kW, primarily the result of the optimization according to overall plant efficiency criteria.

The cost of electricity results are summarized in Table 4-21. The values are shown as 10-year levelized costs based on a 65 percent capacity factor (set I values) and as 30-year levelized costs based on a 85 percent capacity factor (set II values). The latter values were used for optimization purposes. The COE values are given in both current and constant dollars. The comparisons discussed below are made on a constant dollar basis for the set II values.

Case 1CO1 possesses the lowest COE with 38.3 mills/kWh, 1.83 percent lower than the COE of the original Case 1 (39.0 mills/kWh). Case 1CO2 is the next lowest with a COE of 38.4 mills/kWh, which is 1.61 percent lower than the COE of the original Case 1. The highest COE is for Case 1TO1 at 39.2 mills/kWh, 2.30 percent greater than for Case 1CO1, but only 0.44 percent greater than the COE for the original Case 1.

The difference in COE between Case 1CO1 (1CO2) and the original Case 1 results in savings of over 2.4 (2.2) million constant (mid-1990) dollars per year of plant operation. Compared with the original Case 1, the 30-year pre-tax present value of the cost savings in Cases 1CO1 and 1CO2 are 30.0 and 27.7 million constant mid-1990 dollars, respectively.

As we go from set II values to set I values in Table 4-21, the relative differences in COE among the original Case 1 and Cases 1CO1 and 1CO2 are reduced, whereas the differences in COE between Case 1TO1 and the remaining cases increase.

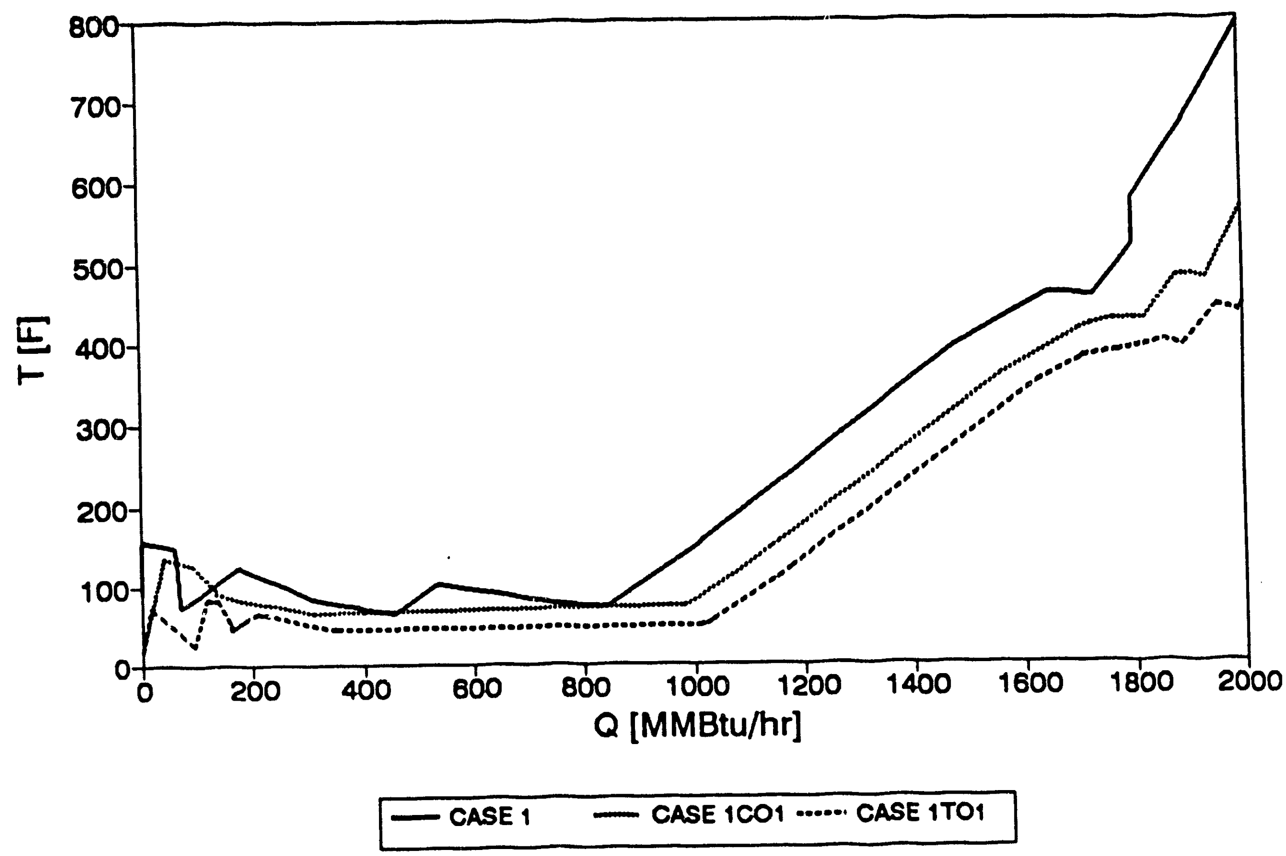

Figure 4-11 **Comparison of the Temperature Differences in the Composite Curves (Figures 4-7, 4-8 and 4-10) for Cases 1, 1CO1, 1TO1. The Case 1CO2 values are very close to the Case 1CO1 curve.**

TABLE 4-14

Comparison of Overall Material Balances

Case	1	1CO1	1CO2	1TO1
Material In, lbm/hr				
Limestone (Dried)	87,733	89,767	89,767	87,361
Coal (As Received)	337,788	345,485	345,485	336,224
BFW Makeup	460,924	453,603	453,284	446,223
Combustion Air to Gas Turbine	4,904,835	4,817,987	4,817,987	4,910,896
Cooling Tower Makeup	3,485,164	3,687,101	3,794,936	3,449,778
Air to Sulfator	195,650	197,485	197,485	192,192
Air Compressor Feed to Gasifier	1,073,957	1,140,136	1,140,136	1,063,331
Nitrogen to Coal Preparation	12,682	13,094	13,094	12,743
Coal Drying Dilution Air	218,665	—	—	—
Total Material In	**10,777,398**	**10,744,658**	**10,852,174**	**10,498,748**
Material Out, lbm/hr				
HRSG Stack Gas to Main Stack	6,740,748	6,927,222	6,927,222	6,922,125
Coal Drying Gas to Local Stack	424,339	—	—	—
Solids to Disposal	102,576	105,193	104,917	102,167
Vent Gas to Atmosphere	10,798	9,215	9,215	8,968
Boiler Blowdown	14,448	16,013	15,694	15,588
Cooling Tower Blowdown, Evaporation,Drift	3,485,164	3,687,101	3,794,936	3,449,778
Total Material Out	**10,778,073**	**10,744,744**	**10,851,984**	**10,498,626**

TABLE 4-15

Gasification Island Design Basis Performance Comparison

Case	1	1CO1	1CO2	1TO1
Feeds:				
Coal Feed (t/day, after drying)	3,792	4,146	4,146	4,035
Limestone Feed (t/day)	1,053	1,077	1,077	1,048
Air Supply (t/day)	12,888	13,682	13,682	12,760
Steam (MMlbm/day)	10,714	10,500	10,500	10,336
Products:				
Fuel Gas (LHV, MMBtu/day)	63,336	63,300	63,300	63,300
Steam (MMlbm/day)	24,894	38,436	32,426	37,414
Solid Waste (t/day)	1,231	1,259	1,259	1,226
Performance:				
Cold Gas Efficiency-LHV (%)	69.7	68.0	68.0	69.9
Carbon Conversion (%)				
Gasifier Only	96.5	96.5	96.5	96.5
Gasification Island	99.9	99.9	99.9	99.9
Sulfur Removal (%)				
Gasifier Only	86.5	91.4	91.4	91.4
Gasification Island	99.4	99.4	99.4	99.4

TABLE 4-16

Comparison of Steam Turbine Performance Data

Case	1	1CO1	1CO2	1TO1
Throttle Flow (lbm/hr)	1,344,245	1,617,520	1,569,513	1,555,582
Reheat Flow (lbm/hr)	1,025,827	1,139,750	1,108,245	1,104,770
Makeup Flow (lbm/hr)	460,924	453,603	453,284	446,223
HRSG Exhaust Temp (°F)	279.8	279.8	279.8	280.1
Steam Turbine Gross Power Output (kW)	182,190	209,948	201,574	205,472
Total GT Power Output (kW)	311,640	311,640	311,640	311,640
Total Gross Power Output (kW)	493,830	521,588	513,214	517,112

TABLE 4-17

Comparison of Overall Plant Performance

Case	1	1CO1	1CO2	1TO1
Gross Power Output (kW)	493,830	521,588	513,214	517,112
Gasifier Island Station Service (kW)	21,660	23,262	23,262	22,109
Power Island Station Service (kW)	11,389	13,352	12,048	13,293
Coal/Limestone Handling Station Service (kW)	2,372	2,425	2,425	2,360
Net Plant Power Output (kW)	458,409	482,549	475,479	479,350
Fuel Consumption (MMBtu/hr - HHV)	3,940	4,030	4,030	3,922
Net Plant Heat Rate (Btu/kWh - HHV)	8,595	8,351	8,475	8,181
Thermal Efficiency - HHV (%)	39.7	40.9	40.2	41.7

TABLE 4-18

Detailed Plant Station Service Values
(All Values in Kilowatts)

Case	1	1CO1	1CO2	1TO1
Gasification Island:				
Booster Air Compressor	17,341	18,405	18,405	17,165
KRW Gasification	1,004	1,098	1,098	1,068
Recycle Gas Compression	1,714	2,131	2,131	2,283
Sulfation	1,281	1,308	1,308	1,273
Miscellaneous	320	320	320	320
Total Gasification Island	21,660	23,262	23,262	22,109
Power Island:				
HRSG Feedpumps	3,420	4,872	3,540	5,100
Condensate Pumps	144	194	194	219
Circ. Water Pumps	2,027	2,144	2,193	2,006
Cooling Tower Fans	1,600	1,693	1,731	1,583
Vacuum Pumps	400	423	433	396
Service Water Pumps	227	227	227	227
Make-Up Pumps	19	19	19	19
Service Air System	149	149	149	149
Gas Turbine Aux.	1,550	1,550	1,550	1,550
Steam Turbine Aux.	1,500	1,728	1,659	1,691
Water Treatment	320	320	320	320
Powerhouse	15	15	15	15
Service Building/Control Room	18	18	18	18
Total Power Island	11,389	13,352	12,048	13,293
Coal & Limestone Handling	2,372	2,425	2,425	2,360
Total	35,421	39,039	37,735	37,762

TABLE 4-19
Breakdown of Process Plant Costs by Plant Section
Mid-1990 Dollars

	Case	1		1CO1		1CO2		1TO1	
	Plant size, MW (net at 90°F)	458.4		482.5		475.5		479.3	
Area No.	Plant Section Description	$M	$/kW	$M	$/kW	$M	$/kW	$M	$/kW
100	Coal Receiving/Handling	17,062	37.2	17,297	35.8	17,297	36.4	17,013	35.5
150	Limestone Receiving/Handling	9,755	21.3	9,892	20.5	9,892	20.8	9,727	20.3
250	Booster Compression	9,081	19.8	7,999	16.6	7,999	16.8	14,154	29.5
300	Gasification	56,571	123.4	64,364	133.4	60,773	127.8	65,969	137.6
380	Recycle Gas Compression	14,298	31.2	14,998	31.1	14,571	30.6	24,187	50.5
400	Gas Conditioning	35,911	78.3	37,011	76.7	37,011	77.8	36,140	75.4
500	External Desulfurization	20,633	45.0	22,613	46.9	21,305	44.8	22,530	47.0
600	Sulfation	13,618	29.7	20,296	42.1	18,071	38.0	20,631	43.0
900	Gas Turbine System	76,211	166.2	76,211	157.9	76,211	160.3	76,211	159.0
1000	HRSG System	25,829	56.3	25,609	53.1	25,014	52.6	27,953	58.3
1100	Steam Turbine System	50,668	110.5	54,477	112.9	53,933	113.4	54,718	114.2
1200	Ash & Fines Handling/Disposal	6,191	13.5	6,220	12.9	6,220	13.1	6,118	12.8
	Total Process Plant Cost	335,827	732.6	356,988	739.8	348,297	732.5	375,350	783.0
	General Plant Facilities	33,484	73.0	35,594	73.8	34,727	73.0	37,425	78.1
	Engineering Fees*	8,780	19.1	9,333	19.3	9,106	19.2	9,813	20.5
	Plant Construction Cost	378,090	824.8	401,915	832.9	392,130	824.7	422,588	881.6

*The Home Office Services are included in the Process Plant Cost of each area, thus they are not included in the Engineering Fees.

TABLE 4-20

Comparison of Capital Costs
Mid-1990 Dollars

Case	1		1CO1		1CO2		1TO1	
Capital Cost	$MM	$/kW	$MM	$/kW	$MM	$/kW	$MM	$/kW
Net Power Output, MW	458.4		482.5		475.5		479.3	
Plant Construction Cost*	378.1	825	401.9	833	392.1	825	422.6	882
Process Contingency	19.8	43	19.8	41	19.4	41	21.2	44
Project Contingency	79.6	174	84.3	175	82.3	173	88.8	185
Total Plant Cost	477.5	1042	506.0	1049	493.8	1030	532.5	1111
AFDC	93.7	204	99.3	206	96.9	204	104.5	218
Total Plant Investment	571.2	1246	605.3	1254	590.8	1242	637.0	1329
Owner's Cost	44.0	96	44.0	91	44.0	93	44.0	92
Total Capital Requirement	615.2	1342	649.4	1346	634.8	1335	681.0	1421

*Note: Plant construction cost = Process plant cost plus general facilities and engineering fees (contingencies excluded)

TABLE 4-21

COST OF ELECTRICITY COMPARISONS

Set I

10-Year Levelized Costs (mills/kWh) at 65% Capacity Factor
Constant mid-1990 Dollars (Current Dollars)

Case	1		CO1		CO2		TO1	
Capital Charges	26.7	(39.8)	26.8	(39.9)	26.6	(39.6)	28.3	(42.1)
Fuel Costs	13.2	(16.3)	12.8	(15.8)	13.0	(16.1)	12.6	(15.5)
O&M Costs	9.1	(11.3)	8.8	(10.8)	8.8	(10.8)	9.0	(11.0)
Cost of Electricity	49.1	(67.4)	48.4	(66.6)	48.4	(66.5)	49.9	(68.8)

Set II

30-Year Levelized Costs (mills/kWh) at 85% Capacity Factor
Constant mid-1990 Dollars (Current Dollars)

Case	1		CO1		CO2		TO1	
Capital Charges	16.7	(26.3)	16.7	(26.4)	16.6	(26.2)	17.7	(27.8)
Fuel Costs	13.6	(20.7)	13.2	(20.1)	13.4	(20.4)	13.0	(19.7)
O&M Costs	8.7	(13.3)	8.4	(12.7)	8.4	(12.7)	8.6	(13.0)
Cost of Electricity	39.0	(60.3)	38.3	(59.2)	38.4	(59.3)	39.2	(60.6)

5.0 PARAMETRIC STUDIES

This section discusses the effect of some important design parameters on the efficiency and cost of electricity (30 year levelized costs in constant mid-1990 dollars at 85 percent capacity factor). This effect is shown for the design configurations of the original Case 1 and Cases 1CO1, 1CO2 and 1TO1. When one parameter is varied, the plant configuration and the values of the remaining important parameters are kept constant. In the following, the effect of a design parameter is shown sometimes only for a portion of the total parameter variation range. This means that this parameter cannot be further varied without significant changes in the configuration being considered.

5.1 THE EFFECT OF GASIFICATION TEMPERATURE

Figures 5-1a and 5-2a illustrate the effect of the gasification temperature on the overall plant thermal efficiency and COE, respectively. In general, the plant efficiency decreases with increasing gasification temperature (e.g., for Cases 1 and 1CO2 as well as for the thermodynamically optimal values at a given gasification temperature). However, the decrease in efficiency in these cases is accompanied by a decrease in the total plant investment costs. The configuration of Case 1CO1 shows the opposite behavior: In the gasification temperature range of 1900°F-1940°F, both the overall plant efficiency and the total plant investment costs increase with increasing gasification temperature.

The cost of electricity is lowest in the gasification temperature range of 1880°F to 1940°F in the configurations of the original Case 1 and Cases 1CO1 and 1CO2. These minimal COE values express the best trade-off between investment costs and fuel costs when the gasification temperature is varied.

5.2 THE EFFECT OF COAL MOISTURE

The effect of coal moisture on plant efficiency and COE is demonstrated in Figures 5-1b and 5-2b. With increasing coal moisture, the total plant efficiency increases in Case 1TO1, whereas it slightly decreases in Cases 1CO1 and 1CO2. The COE, however, continually decreases in all cases with increasing coal moisture at the gasifier inlet. This effect can be explained by (a) the more effective use of the heat of the sulfation area combustion gas, and (b) the decrease in the investment and O&M costs of the zinc ferrite unit with increasing coal moisture.

5.3 THE EFFECT OF HP STEAM PRESSURE

As indicated previously, the steam high pressure value (P_{HP}) has a significant impact on the interaction between the gasification and power islands and, thus, the efficiency and cost of electricity of the IGCC power plant. Figures 5-1c and 5-2c present the effect of P_{HP} on these parameters. In general, the overall plant efficiency increases with increasing P_{HP} value. This value, however, cannot be meaningfully varied in a wider range than the one shown in Figures 5-1 and 5-2 without important changes in the design configuration.

The COE values as a function of P_{HP} are lowest at the P_{HP} design values for each case (2055 psia and 1515 psia for Cases 1CO1 and 1CO2, respectively). In Case 1TO1 the COE decreases with decreasing P_{HP}, but then this case no longer represents the thermodynamic optimum. The relatively high slope of the curves in Figures 5-1c and 5-2c indicates the importance of this pressure value on the final results.

The increased plant efficiency achieved with higher pressure steam needs to be balanced against the effect of capacity turndown on the HRSG performance. Operation of the gas turbine at a capacity factor below 80% might require a disproportional reduction in the fuel supplied to the gas turbine. In this case, both the inlet and outlet temperatures of the gas turbine are reduced. This reduction can affect significantly the temperature pinches in the HRSG, and, consequently, the HRSG operation at partial load.

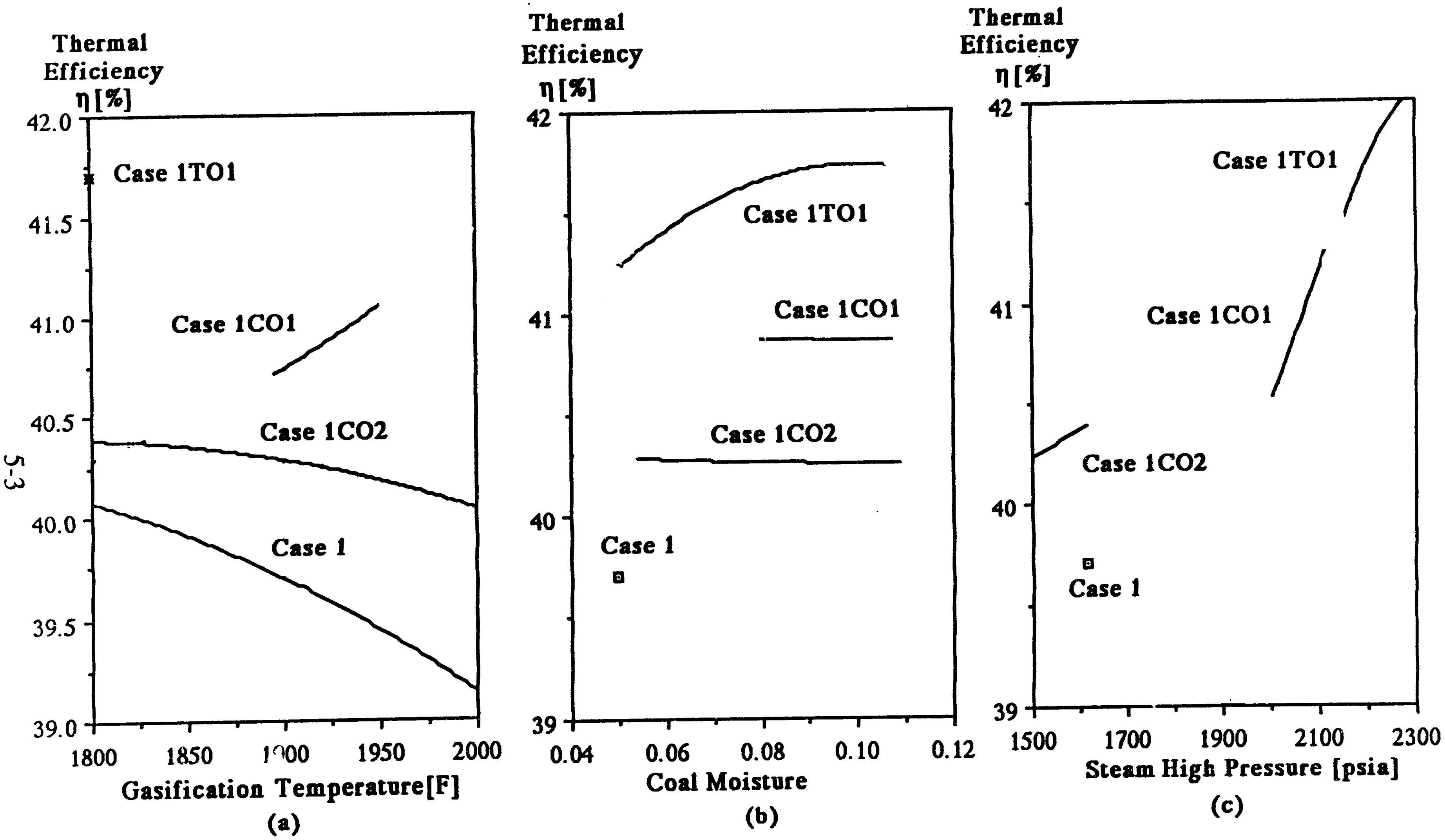

Figure 5-1 Overall Plant Thermal Efficiency as a Function of Gasification Temperature, Coal Moisture and Steam High Pressure for Various Cases.

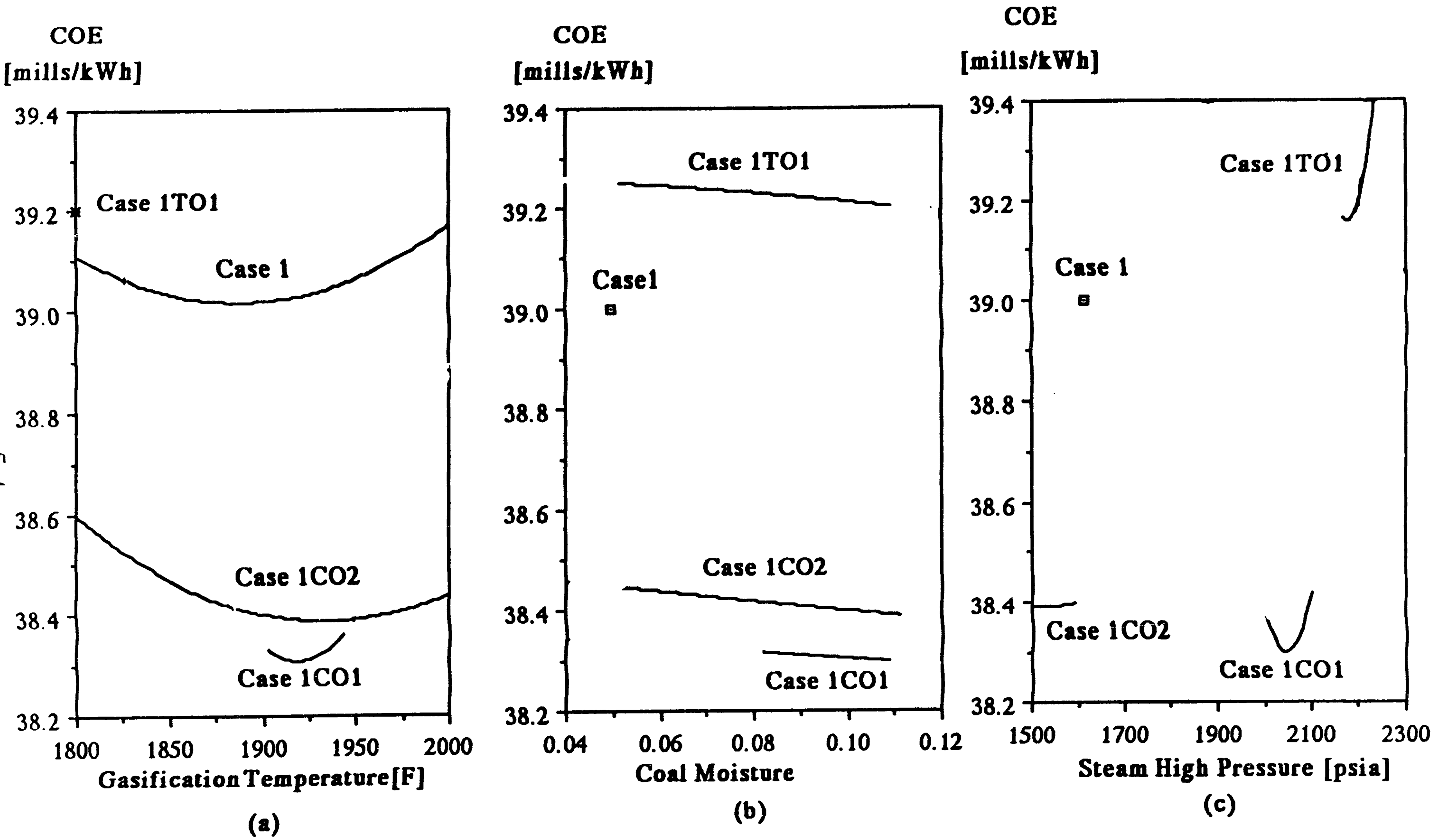

Figure 5-2 Cost of Electricity as a Function of Gasification Temperature, Coal Moisture and Steam High Pressure for Various Cases.

5.4 THE EFFECT OF GAS TURBINE PERFORMANCE

Factory prototype evaluations and engineering tests performed on the Virginia Power (Chesterfield) MS7001F gas turbine have enabled GE to refine the computer model used for gas turbine performance predictions and to relax some of the application restrictions. The torque limit in the gas turbine was raised to the equivalent of 192 MW electric power between October 1989 and October 1991. This, combined with the increase in air flow and effective firing temperature, increased the generator output by 9.2 MW for Case 1C, as shown in Table 5-1. When the ambient temperature is 59°F, instead of the 90°F assumed in this study, the increase in the generator output for Case 1C is 21.6 MW (from 163.3 MW to 184.9 MW). Case 1C has a coal moisture of 11.12 weight percent and the same gasifier island configuration as Cases 1CO1 and 1CO2. The only difference is in the gasification temperature, which is 1800°F in Case 1C and 1920°F in Cases 1CO1 and 1CO2.

After GE supplied the new data, an attempt was made to estimate the effect of the new performance data on the overall plant efficiency and COE for the Cases 1, 1CO1, 1CO2 and 1TO1. Since the new gas turbine exhaust temperature is higher than the old, some design changes in the HRSG, steam high pressure (P_{HP}), and steam cycle configuration were required. The results are summarized in Table 5-2.

The new gas turbine performance data have significant impact on the overall efficiency and the cost of electricity. Use of the new data results in (a) an increase in thermal efficiency by 0.7-0.8 percentage points in the optimized cases, and (b) a decrease in the cost of electricity by more than 2.5 percent. Among the cases compared in Table 5-2, Cases 1CO1 and 1CO2 are close to the corresponding optimal cases with the new gas turbine performance data. Case 1TO1, however, no longer represents the thermodynamically optimal case. This case would have a different design configuration, a higher P_{HP} value than Case 1TO1, and an efficiency above 42.5 percent.

The results presented in this section confirm that the efficiency of the gas turbine system significantly affects the performance and COE in IGCC power plants. Gas turbine design improvements permitting higher firing temperatures are very desirable in IGCC applications provided that the increase in cooling air requirements does not offset most of the increase in power generation.

TABLE 5-1

GE MS7001F Gas Turbine Performance Comparisons

	October 1989 Capability		October 1991 Capability
Case	1	1C	1C
Ambient Temperature (°F)	90	90	90
Relative Humidity (%)	60	60	60
Gas Turbine Compressor Air to Gasifier Island			
Ratio to Fuel Gas	0.5869	0.583	0.583
Mass Flow (lbm/sec)	149.1	147.3	155.0
Temperature (°F)	760	760	760
Pressure (psia)	200	190	195
Fuel Gas Flow to Gas Turbine			
Heating Value (LHV, Btu/scf)	87.4	87.6	87.6
Mass Flow (lbm/sec)	254.1	252.6	265.8
Temperature (°F)	1000	1000	1000
Pressure (psia)	295	290	300
Generator Output (MW)	155.8	156.4	165.6
Turbine Exhaust Flow			
Mass Flow (lbm/sec)	935.5	935.8	943.7
Temperature (°F)	1098	1095	1137

Relevant Site Conditions

Site Elevation	725 feet
Barometric Pressure (Elev. 725 ft.)	14.3 psia
Inlet Air Pressure Loss	3.5 in. H_2O
Exhaust Pressure Loss	15 in. H_2O
Steam Injection for NO_x Control	none
Supplementary Fuel	none
Generator Power Factor	0.9
Gas Turbine Auxiliary Load (not included above)	775 kW

TABLE 5-2

Comparison of the Overall Thermal Efficiency and the COE when the Old (October 1989) and the New (October 1991) Performance Capabilities of the GE MS7001F Gas Turbine are Used

Case	Pressure P_{HP} [psia]	Thermal Efficiency [%] Old	Thermal Efficiency [%] New	Cost of Electricity* [mills/kWh] Old	Cost of Electricity* [mills/kWh] New
1	1600	39.7	40.1	39.0	37.9
1CO1	2115	40.9	41.6	38.3	37.3
1CO2	1515	40.2	41.0	38.4	37.3
1TO1	2250	41.7	42.4	39.2	37.4

*Constant mid-1990 dollars; thirty year levelized cost at 85% capacity factor

5.5 COST OF ELECTRICITY SENSITIVITY STUDIES

Simple sensitivity analyses for the cost of electricity were conducted to evaluate the effect of some uncertainties associated with the COE estimation process. Figures 5-3 through 5-5 show the sensitivity of COE (expressed in current dollars) to total plant cost, real escalation rate for coal cost, and plant capacity factor, respectively. The sensitivity is presented for two sets of values. Set I refers to ten-year levelized costs at 65 percent capacity factor. Set II represents thirty-year levelized costs at 85 percent capacity factor. In these figures the cost differences between Cases 1CO1 and 1CO2 are small.

Most of the uncertainty in capital cost estimation is associated with the gasification island because the major processes therein have little or no commercial history. An overrun in estimated total plant cost affects Case 1TO1 more than the remaining cases, particularly in set I. The effect of total plant cost on COE is larger in set I than in set II while the effect of real escalation rate for coal cost is stronger in set II. The plant capacity factor has a dominant influence on COE, as shown in Figure 5-3. When the capacity factor changes from 65 percent to 85 percent, the savings in the COE are more than 15 percent in set I and more than 20 percent in set II. The results shown in Figure 5-5 assume that the plant will operate between 65 and 85 percent of the time at 100 percent capacity. If the given capacity factor value is obtained through plant operation at reduced capacity, the effect of capacity factor on the COE is more pronounced.

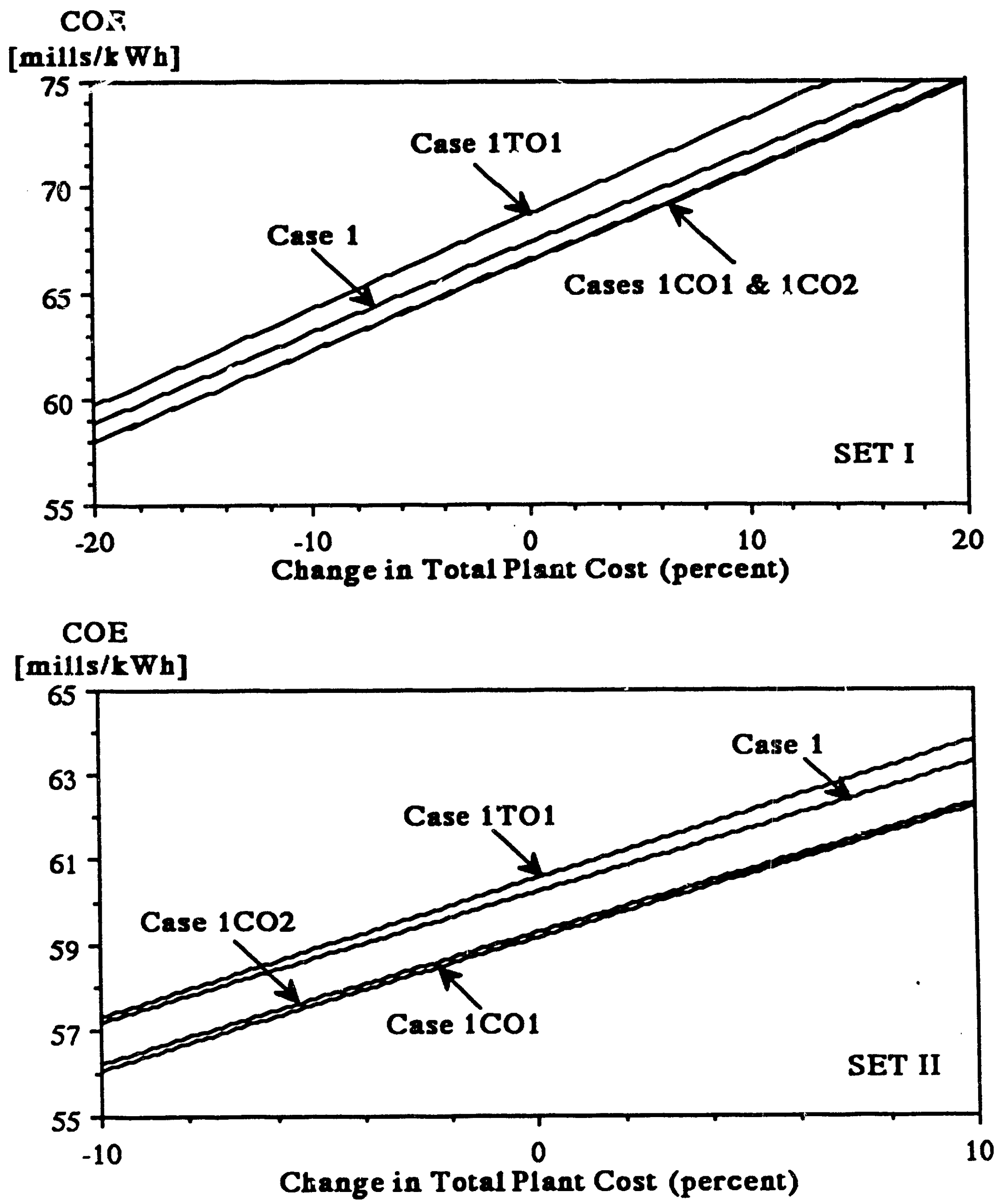

Figure 5-3 Sensitivity of COE (Current Dollars) to Changes in Total Plant Cost
Set I: Ten-Year Levelized Cost; 65% Capacity Factor
Set II: Thirty-Year Levelized Cost; 85% Capacity Factor

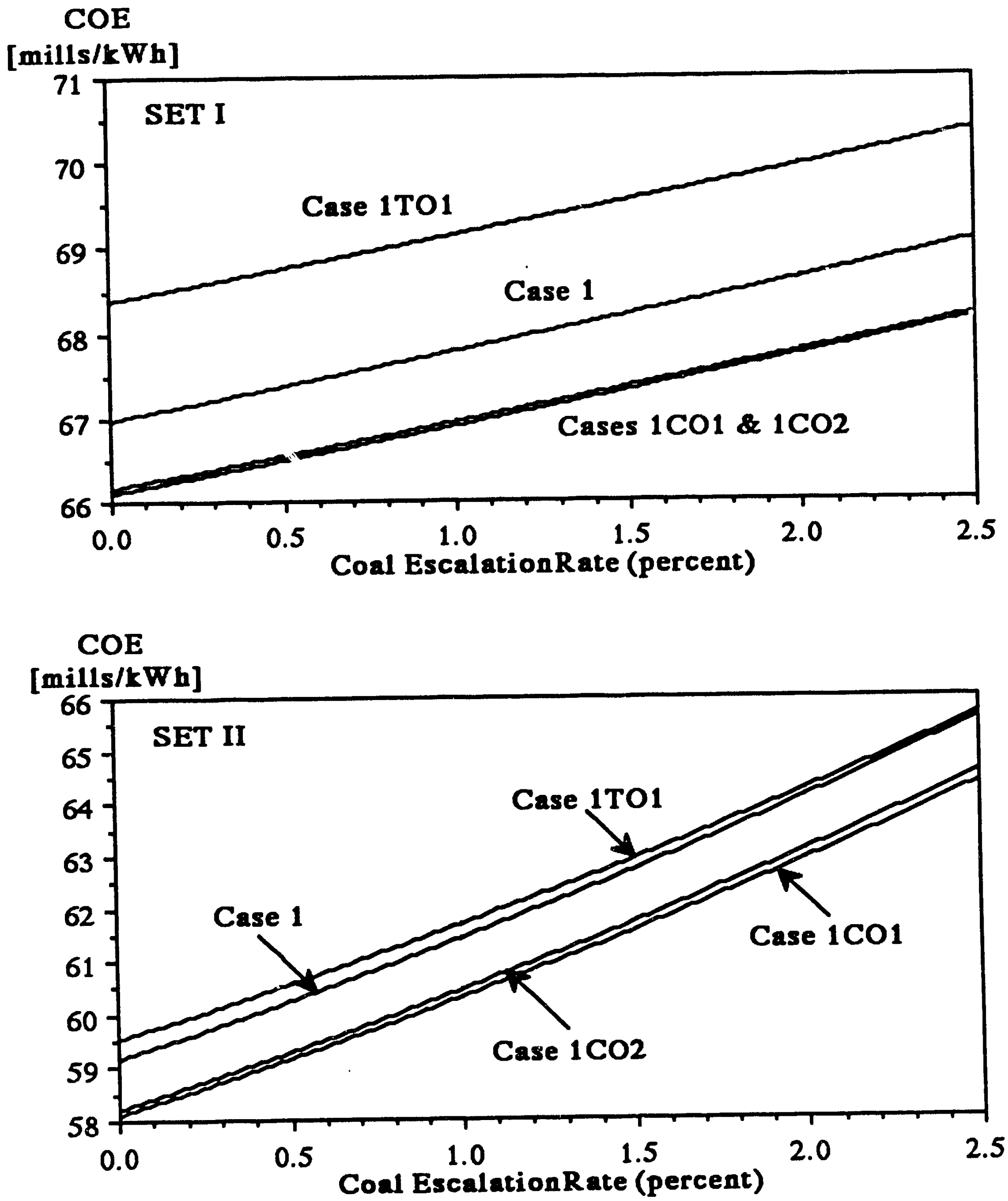

Figure 5-4 Sensitivity of COE (Current Dollars) to Changes in Real Escalation Rate of Coal Cost
Set I: Ten-Year Levelized Cost; 65% Capacity Factor
Set II: Thirty-Year Levelized Cost; 85% Capacity Factor

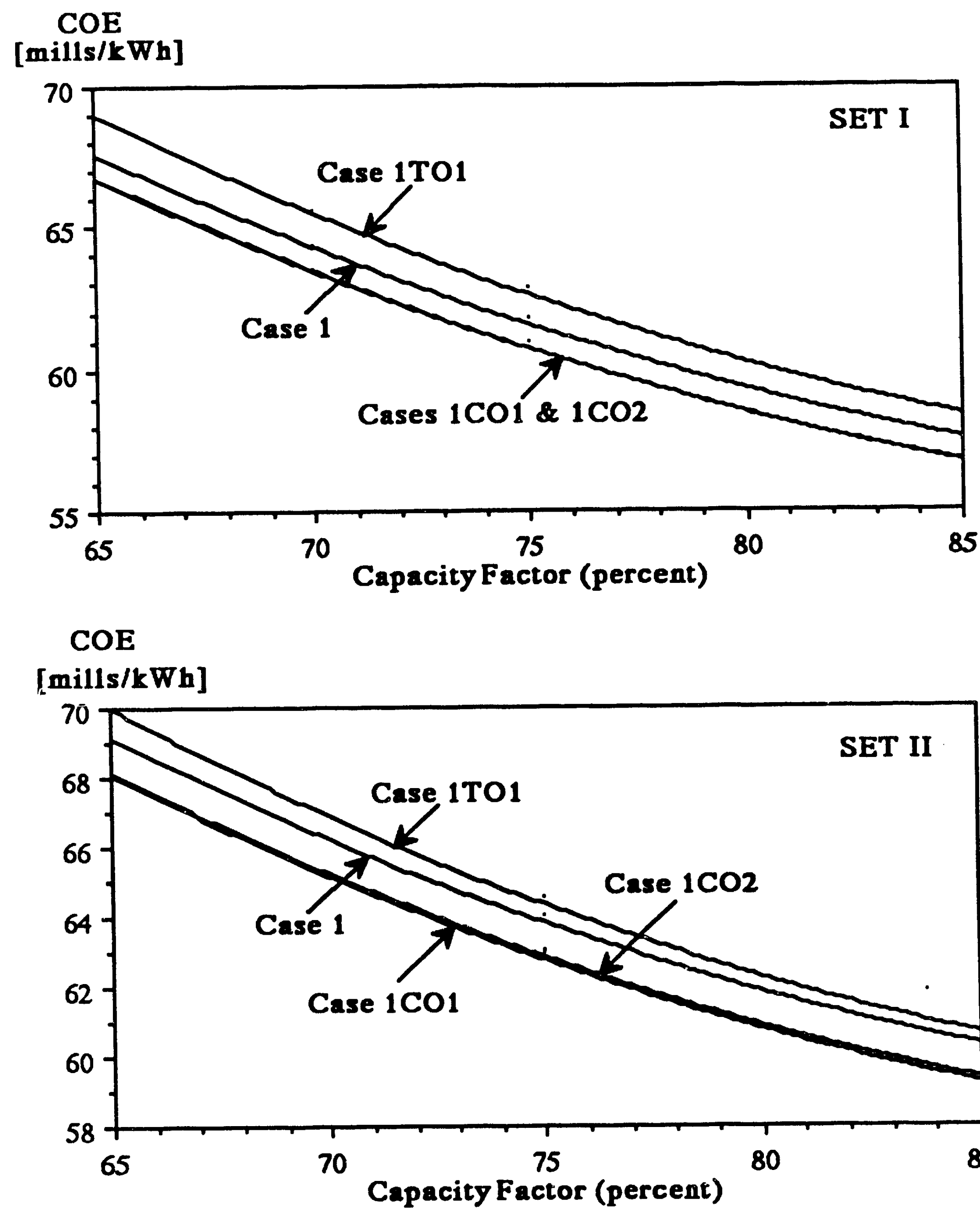

Figure 5-5 Sensitivity of COE (Current Dollars) to Changes in Plant Capacity Factor
Set I: Ten-Year Levelized Cost; 65% Capacity Factor
Set II: Thirty-Year Levelized Cost; 85% Capacity Factor

6.0 CONCLUSIONS AND RECOMMENDATIONS

Several IGCC power plant concepts based on air-blown gasification with hot gas cleanup were investigated in this study. The results were compared on the basis of efficiency and cost of electricity. The primary motivations for the study were to develop cost optimal design configurations and to understand the effects of important design parameters on efficiency and cost of electricity.

In addition to the original Case 1, which was developed in a previous study [1], three cases are discussed in this report. Two of them were optimized from the cost viewpoint and the third from the thermodynamic viewpoint. The comparisons among these cases are considered to be accurate when compared to each other since they were based on the same underlying assumptions. In evaluating the results from similar studies we should keep in mind that, in general, the results of performance predictions are more accurate than the results of cost estimates.

Table 6-1 compares the performance of the four cases discussed here. Table 6-2 provides a summary of estimated capital costs, O&M costs and cost of electricity. The major difference between the cost optimal Cases 1CO1 and 1CO2 is in the steam high pressure values which are 2055 psia and 1515 psia, respectively. Case 1CO1 has a lower heat rate (8351 Btu/kWh) but a slightly higher total capital requirement (1346 $/kW) than Case 1CO2. Assuming thirty-year levelized costs at 85 percent capacity factor, Case 1CO1 possesses the lowest cost of electricity (38.3 mills/kWh in constant mid-1990 dollars) among all four cases. When ten-year levelized costs at 65 percent capacity factor are assumed, both Cases 1CO1 and 1CO2 have practically the same cost of electricity. The performance and cost advantages of these cases over the original Case 1 are attributable to the optimization of the steam high pressure, gasification temperature, coal moisture at the gasifier inlet, steam turbine, and overall heat exchanger network.

Case 1TO1 represents the thermodynamically optimal case and was included here to demonstrate the potential for improving the overall plant efficiency. The thermal efficiency of this case is 41.7 percent, an improvement of about 5.0 percent compared with the original Case 1 (39.7 percent). This improvement is achieved with an increase of only about 0.5 percent in the cost of electricity. It should be noted that other cases not presented in this report with slightly lower efficiencies than Case 1TO1 have lower cost of electricity than the original Case 1. The three optimized cases present a lower environmental impact than the original Case 1, primarily due to their higher overall thermodynamic efficiency.

The difference in the cost of electricity between Case 1CO1 (1CO2) and the original Case 1 results in savings of over 2.4 (2.2) million constant (mid-1990) dollars

TABLE 6-1

Summary of Gasification Power Plant Performance

Case	1	1CO1	1CO2	1TO1
Gasification Temperature [°F]	1900	1920	1920	1800
Coal Moisture [wt%]	4.98	11.12	11.12	11.12
Steam High Pressure [psia]	1600	2055	1515	2200
Gasifier Feeds:				
Coal (t/day) (as prepared)	3,792	4,146	4,146	4,035
Limestone (t/day)	1,053	1,077	1,077	1,048
Air (t/day)	12,888	13,682	13,682	12,760
Steam (MMlbm/day)	10,714	10,500	10,500	10,336
Gasification Island Products:				
Fuel Gas (LHV, MMBtu/day)	63,336	63,300	63,300	63,300
Steam to Power Island (Mlbm/day)	24,894	38,436	32,426	37,414
Solid Waste (t/day)	1,231	1,259	1,259	1,226
Gasification Island Performance:				
Cold Gas Efficiency - LHV (%)	69.7	68.0	68.0	69.9
Carbon Conversion (%)				
Gasifier Only	96.5	96.5	96.5	96.5
Gasification Island	99.9	99.9	99.9	99.9
Sulfur Removal (%)				
Gasifier Only	86.5	91.4	91.4	91.4
Gasification Island	99.4	99.4	99.4	99.4
Overall Plant Performance:				
Coal Energy Input (MMBtu/hr)	3,940	4,030	4,030	3,922
Gross Power Output (MW)				
Combustion Turbine(s)	311.6	311.6	311.6	311.6
Steam Turbine(s)	182.2	210.0	201.6	205.5
Total Power	493.8	521.6	513.1	517.1
Station Service (MW)	35.4	39.0	37.7	37.8
Net Power Output (MW)	458.4	482.5	475.4	479.3
Net Heat Rate - HHV (Btu/kWh)	8595	8351	8475	8181
Thermal Efficiency - HHV (%)	39.7	40.9	40.2	41.7

TABLE 6-2

Summary of Gasification Power Plant Costs

Case	1		1CO1		1CO2		1TO1	
	$MM	$/kW	$MM	$/kW	$MM	$/kW	$MM	$/kW
Net Power Output, MW	458.4		482.5		475.5		479.3	
CAPITAL COST								
Plant Construction Cost	378	825	402	833	392	825	423	882
Process Contingency	20	43	20	41	19	41	21	44
Project Contingency	80	174	84	175	82	173	89	185
Total Plant Cost	477	1042	506	1049	494	1039	533	1111
AFDC	94	204	99	206	97	204	104	218
Total Plant Investment	571	1246	605	1254	591	1242	637	1329
Owner's Costs	44	96	44	91	44	93	44	92
Total Capital Requirement	615	1342	649	1346	635	1335	681	1421
O&M COST*								
Fixed O&M Costs, $/kW-Yr	47.08		46.48		46.41		48.43	
Variable O&M, mills/kWh	2.41		2.11		2.14		2.08	
Fuel Costs, mills/kWh	12.90		12.53		12.72		12.27	
COST OF ELECTRICITY								
Current, $, mills/kWh**	67.4		66.6		66.5		68.8	
Constant $, mills/kWh**	49.1		48.4		48.4		49.9	
Current $, mills/kWh***	60.3		59.2		59.3		60.6	
Constant $, mills/kWh***	39.0		38.3		38.4		39.2	

*First-Year O&M Cost, at 85% Capacity Factor
**Ten-Year Levelized Cost at 65% Capacity Factor
***Thirty-Year Levelized Cost at 85% Capacity Factor

per year of plant operation. Compared with the original Case 1, the thirty-year pre-tax present values of the cost savings in Cases 1CO1 and 1CO2 are 30.0 and 27.7 million constant mid-1990 dollars, respectively.

This study identified several design changes which improve the cost effectiveness of the Case 1 IGCC concept. These changes include the following:

- The coal should be supplied to the gasifier with the as received moisture of 11.12 weight percent, or with the highest moisture content allowed by the reliability of the coal feeding process.
- The gasification temperature and the steam high pressure should be optimized using the thermoeconomic variables discussed in section 3. This optimization refers to the interaction between the gasification island and the power island.
- The heat of the flue gas from the sulfation area should be used in the HRSG.
- The mass flow rate of the quench steam should be decreased and its temperature increased through adjustments in the steam turbine and elimination of the desuperheating process.
- The BFW preheater in area 250 (air booster compression) and the recycle gas cooler should preheat LP (or low-temperature) instead of HP (or high-temperature) feedwater.
- The recycle gas should be extracted from the clean gas after the exit gas cooler instead of after the zinc-ferrite unit.
- The size of the product gas cooler and the exit gas cooler should be increased to accommodate some of the above changes.
- The design of the HP steam turbine should be adjusted to the new steam high pressure values.
- The deaerator should be operated at the lowest possible pressure.
- At least one LP steam extraction should be used to preheat feedwater.

These recommendations refer only to the design of Case 1 and should not be used automatically in conjunction with other IGCC concepts. The cost optimal Cases 1CO1 and 1CO2 are certainly not unique. Other design configurations can be found with comparable cost of electricity values.

Future studies should include investigation of the economic feasibility of design options aimed at eliminating or modifying the external desulfurization step and the gas recycling process. Significant performance and cost benefits should be expected from the elimination or modification of these processes.

The thermoeconomic analysis and optimization techniques were very useful tools in conducting this study. Some optimization techniques were refined during the investigations and will be applied to future design optimizations of IGCC plants and other energy systems.

The parametric study conducted to investigate the impact of major design parameters on the efficiency and cost of electricity established the importance of the gas turbine performance, the steam high pressure, and the gasification temperature for the cost optimization process. The cost sensitivity studies confirmed that the plant capacity factor is the most important variable for cost-effective plant operation.

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APPENDIX

APPENDIX

This Appendix presents some detailed results of the thermoeconomic analysis of Cases 1 and 1CO1. The variables shown here were used in the thermoeconomic evaluation and optimization of the cases discussed in this report.

TABLE A-1

Cost of Fuel (c_F), Cost of Product (c_P), Relative Cost Difference (d), Investment Cost Rate (Z), Cost Rate of Exergy Destruction (D), and Thermoeconomic Factor (f) for each Area in the Original Case 1

Area		c_F [$/MWh]	c_P [$/MWh]	d [%]	Z [$/hr]	D [$/hr]	f [%]
Area 250	Booster Compression	17.94	38.11	112.36	135.4	145.0	48.28
Area 300	KRW Gasification	7.32	9.97	36.21	889.7	1784.8	33.27
Area 380	Recycle Gas Compression	10.53	15.49	47.03	234.5	23.2	91.00
Area 400	Gas Conditioning	12.49	14.60	16.92	607.8	267.2	69.46
Area 500	External Desulfurization	10.21	10.42	2.07	330.8	176.8	65.16
Area 600	Sulfation	7.31	22.17	203.20	233.5	89.0	72.41
Area 900	Gas Turbine System	9.62	20.26	110.54	1190.6	2649.1	31.01
Area 1000	Heat Recovery Steam Generator	14.83	19.61	32.21	403.6	508.3	44.26
Area 1100	Steam Cycle	20.78	27.71	33.38	648.8	1059.5	37.98
Gasification Island		7.40	11.31	52.89	2431.7	2257.1	51.86
Power Island		11.69	22.76	94.70	2242.9	4214.6	34.73
Total Plant		5.72	39.02	582.20	4674.6	4125.5	53.12

TABLE A-2

Cost of Fuel (c_F), Cost of Product (c_P), Relative Cost Difference (d), Investment Cost Rate ($\dot{Z}$), Cost Rate of Exergy Destruction ($\dot{D}$), and Thermoeconomic Factor (f) for each Area in Case 1CO1

Area		c_F [$/MWh]	c_P [$/MWh]	d [%]	$\dot{Z}$ [$/hr]	$\dot{D}$ [$/hr]	f [%]
Area 250	Booster Compression	18.47	40.75	120.63	119.3	181.7	39.63
Area 300	KRW Gasification	7.32	10.14	38.63	1005.3	1902.8	34.57
Area 380	Recycle Gas Compression	10.44	15.28	46.43	246.0	35.3	87.46
Area 400	Gas Conditioning	12.39	14.24	14.91	626.4	233.2	72.87
Area 500	External Desulfurization	10.16	10.38	2.12	229.4	171.8	57.18
Area 600	Sulfation	5.82	19.57	235.97	331.1	50.6	86.75
Area 900	Gas Turbine System	9.72	20.40	109.84	1190.6	2695.9	30.63
Area 1000	Heat Recovery Steam Generator	14.49	18.19	25.49	400.1	365.2	52.28
Area 1100	Steam Cycle	19.96	27.08	35.69	705.8	1086.9	39.37
Gasification Island		7.35	11.47	56.15	2687.3	2334.0	53.52
Power Island		11.94	22.85	91.38	2296.5	4260.9	35.02
Total Plant		5.72	38.31	569.42	4983.8	4142.2	54.61

TABLE A-3

Cost of Fuel (c_F), Cost of Product (c_P), Relative Cost Difference (d), Investment Cost Rate ($\dot{Z}$), Cost Rate of Exergy Destruction ($\dot{D}$), and Thermoeconomic Factor (f) for the Major Components in the Original Case 1

	Area	c_F [$/MWh]	c_P [$/MWh]	d [%]	$\dot{Z}$ [$/hr]	$\dot{D}$ [$/hr]	f [%]
1	Gasifier	7.03	9.08	29.03	545.1	1436.7	27.50
2	Cyclones	9.09	9.13	0.52	31.6	15.8	66.72
3	Product Gas Cooler	9.14	18.72	104.76	222.3	225.0	49.70
6	Quench Steam Mixing	9.55	9.72	1.73	0.0	167.8	0.00
7-9	Cyclones & Gas Filter	9.97	9.97	0.00	479.6	23.4	95.35
10	Area 600: Sulfation	7.31	22.17	203.20	233.5	89.0	72.42
11	Chloride Guard	9.97	10.11	1.44	128.2	14.9	89.62
12	Air Recuperator	22.07	27.47	24.47	15.5	39.2	28.27
13	BFW Preheater	22.19	31.43	41.67	30.3	9.3	76.46
15	Booster Air Compressor	22.86	38.13	66.76	83.8	63.7	56.81
16	Zinc Ferrite Unit	10.20	10.20	0.00	209.8	100.3	67.67
18	Exit Gas Cooler	15.87	28.72	81.01	120.9	119.0	50.40
20	Recycle Gas Cooler	15.93	46.75	193.43	83.2	25.5	76.58
21	Recycle Gas Compressor I	22.86	153.75	572.45	105.1	12.1	89.71
23	Recycle Gas Compressor II	22.86	250.07	993.73	46.2	1.7	96.47
26	Gas Turbine & Air Compressor	12.91	19.96	54.63	1131.1	1066.9	51.46
27	Combustion Chamber	13.02	15.83	21.61	59.5	2364.1	2.46
29	Superheater-Reheater-Blast RH	14.83	17.69	19.28	99.7	191.7	34.21
30	HP Drum	14.80	19.45	31.42	98.7	51.2	65.87
31	IP Superheater	14.10	21.04	49.16	3.2	6.8	31.91
32	IP Drum	14.72	21.05	42.93	18.3	41.5	30.63
33	HP2 Economizer	14.78	20.96	41.79	116.4	20.6	84.96
34	IP Economizer	14.18	23.67	66.99	1.9	11.1	14.80
35	HP1 Economizer	14.77	21.17	43.34	38.9	59.6	39.49
36	Feedwater Heater 1	14.71	24.27	64.96	6.4	61.5	9.48
37	Feedwater Heater 2	14.53	63.39	336.27	19.9	46.7	29.90
39	HP Turbine	21.76	26.53	21.92	133.4	133.2	50.05
43	Desuperheater	21.82	22.60	3.55	0.0	36.5	0.00
45	IP Turbine	21.36	26.19	22.60	67.2	68.7	49.47
48	LP Turbine	21.82	28.82	32.10	234.3	452.9	34.09
49	Seal Steam Regulator	21.48	35.81	66.69	1.9	0.5	79.85
52	LP Pump	22.86	207.39	807.09	5.6	0.3	94.58
54	Gland Seal Condenser	35.98	269.53	649.07	1.9	4.7	28.47
55	Deaerator	29.19	36.03	23.46	9.3	44.4	17.34
57	HP Pump	22.86	40.11	75.42	14.4	24.9	36.69
58	IP Pump	22.87	204.68	795.07	5.6	0.4	93.16

TABLE A-4

Cost of Fuel (c_F), Cost of Product (c_P), Relative Cost Difference (d), Investment Cost Rate ($\dot{Z}$), Cost Rate of Exergy Destruction ($\dot{D}$), and Thermoeconomic Factor (f) for the Major Components in Case 1CO1

Area	c_F [$/MWh]	c_P [$/MWh]	d [%]	$\dot{Z}$ [$/hr]	$\dot{D}$ [$/hr]	f [%]
1 Gasifier	7.04	9.12	29.54	539.3	1520.3	26.19
2 Cyclones	9.13	9.18	0.55	31.6	20.9	60.21
3 Product Gas Cooler	9.18	18.64	102.94	305.5	256.3	54.37
6 Quench Steam Mixing	9.58	9.72	1.47	0.0	143.8	0.00
7-9 Cyclones & Gas Filter	9.96	9.96	0.00	493.9	22.4	95.66
10 Area 600: Sulfation	5.82	19.57	235.97	331.1	50.6	86.75
11 Chloride Guard	9.96	10.11	1.48	132.5	15.6	89.50
13 Air Recuperator	22.20	27.60	24.31	16.1	41.9	27.72
14 BFW Preheater	22.31	49.92	123.73	14.1	76.9	15.50
16 Booster Air Compressor	22.93	35.32	54.06	87.1	67.8	56.23
17 Zinc Ferrite Unit	10.16	10.16	0.00	216.9	98.8	68.71
18 Exit Gas Cooler	14.75	26.85	82.06	142.3	106.0	57.30
20 Recycle Gas Cooler	14.77	76.30	416.45	75.4	38.0	66.45
21 Recycle Gas Compressor I	22.93	148.39	547.20	125.5	16.3	88.53
23 Recycle Gas Compressor II	22.93	244.88	968.02	45.2	1.7	96.37
26 Gas Turbine & Air Compressor	12.93	20.08	55.33	1131.1	1098.1	50.74
27 Combustion Chamber	13.03	15.84	21.53	59.5	2360.4	2.46
29 Superheater I - Reheater I	14.53	18.15	24.96	16.7	34.7	32.50
Superheater I - Reheater II	14.55	17.12	17.66	96.7	118.4	44.96
Superheater II	14.52	18.60	28.17	39.8	53.8	42.52
31 HP1 Economizer	14.55	18.65	28.21	240.7	109.4	68.76
32 Feedwater 1/2	14.26	46.62	226.83	6.2	35.8	14.65
34 HP Turbine	20.51	24.97	21.73	152.2	140.0	52.09
37 IP Turbine	20.62	25.36	22.95	101.9	106.0	49.01
39 LP Turbine	21.98	29.22	32.93	233.1	494.1	32.06
40 Seal Steam Regulator	20.31	34.48	69.79	2.1	0.6	77.86
43 LP Pump	22.97	211.34	820.01	5.3	0.3	94.34
46 Deaerator	35.18	38.64	9.82	9.7	16.6	36.72
47 HP Pump	22.93	39.34	71.60	19.6	35.2	35.81

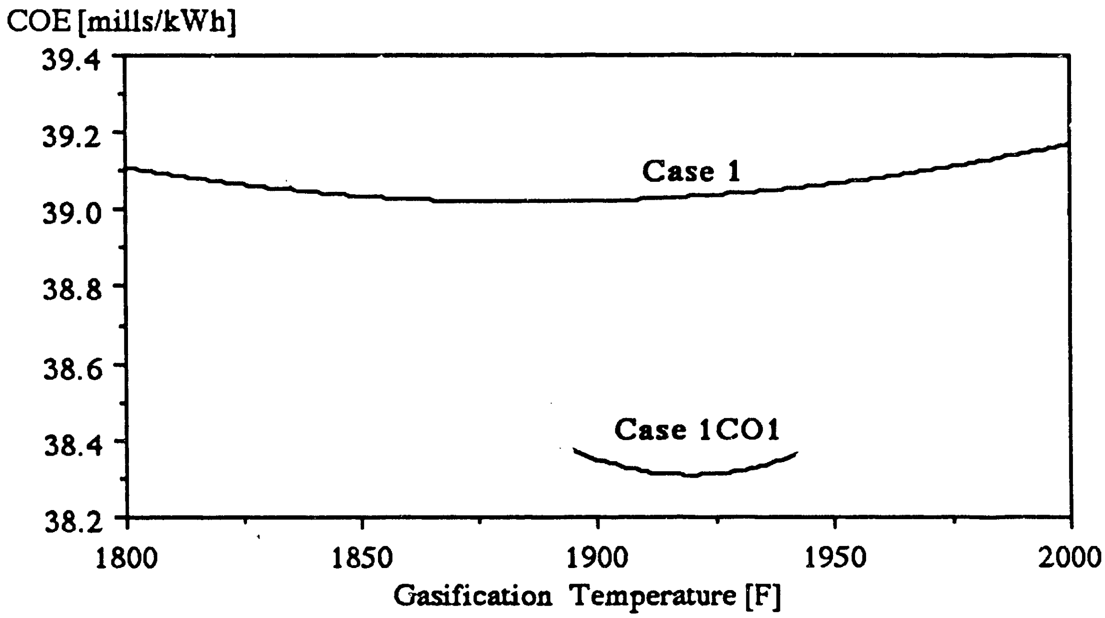

Figure A-1 Cost of Electricity as a Function of the Gasification Temperature for the Original Case 1 and Case 1CO1

Note: If the gasification temperature is the only variable in an otherwise fixed design configuration, then the minimum cost of electricity is obtained in the temperature range of 1850°F-1900°F in the configuration of the original Case 1 (as predicted in Reference [1]), and in the range of 1900°F-1940°F in the configuration of Case 1CO1. The cost of electricity in the configuration of Case 1CO1 is more sensitive to changes in the gasification temperature than in the configuration of the original Case 1. This figure demonstrates the effect of different configurations and other design variables on the optimum value of an important design parameter.

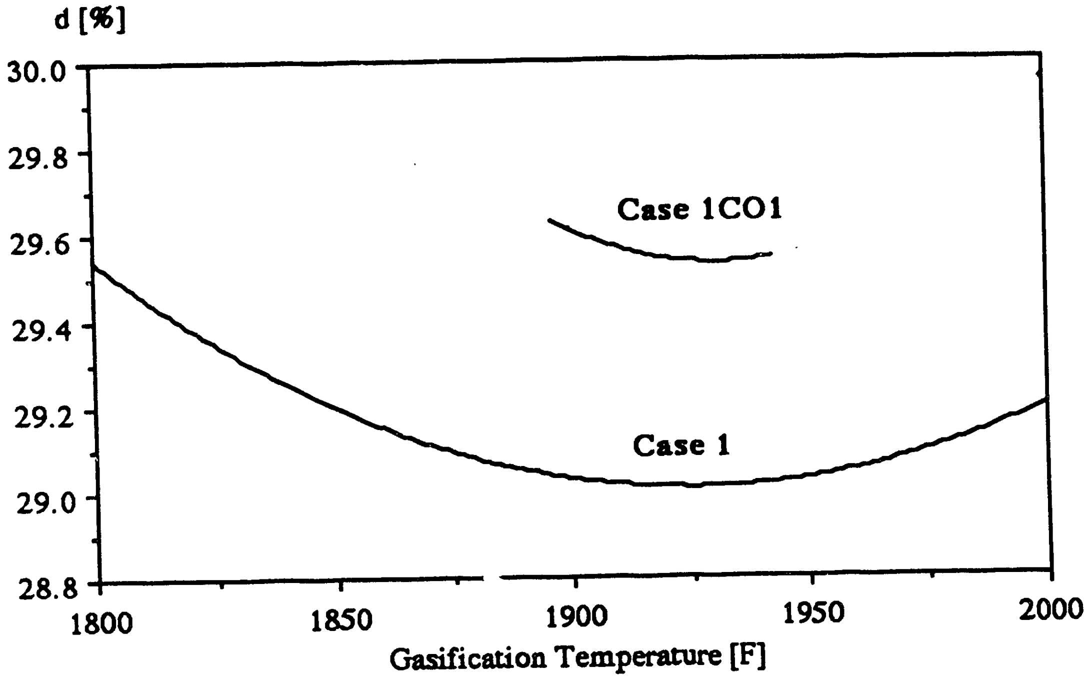

Figure A-2 Relative Cost Difference d for the Gasifier as a Function of the Gasification Temperature for the Original Case 1 and Case 1CO1

Note: The relative cost difference d (Equation 3-6) predicts a cost optimal gasification temperature in the range of 1900°F-1950°F for both the design configuration of Case 1 and Case 1CO1 in spite of the design differences in these cases. Thus, the relative cost difference d could be used in the beginning of the design process to estimate the optimal gasification temperature even before the design of the remaining areas in the plant is completed. This finding could result in significant savings in engineering time during the development of a "good" IGCC power plant design.

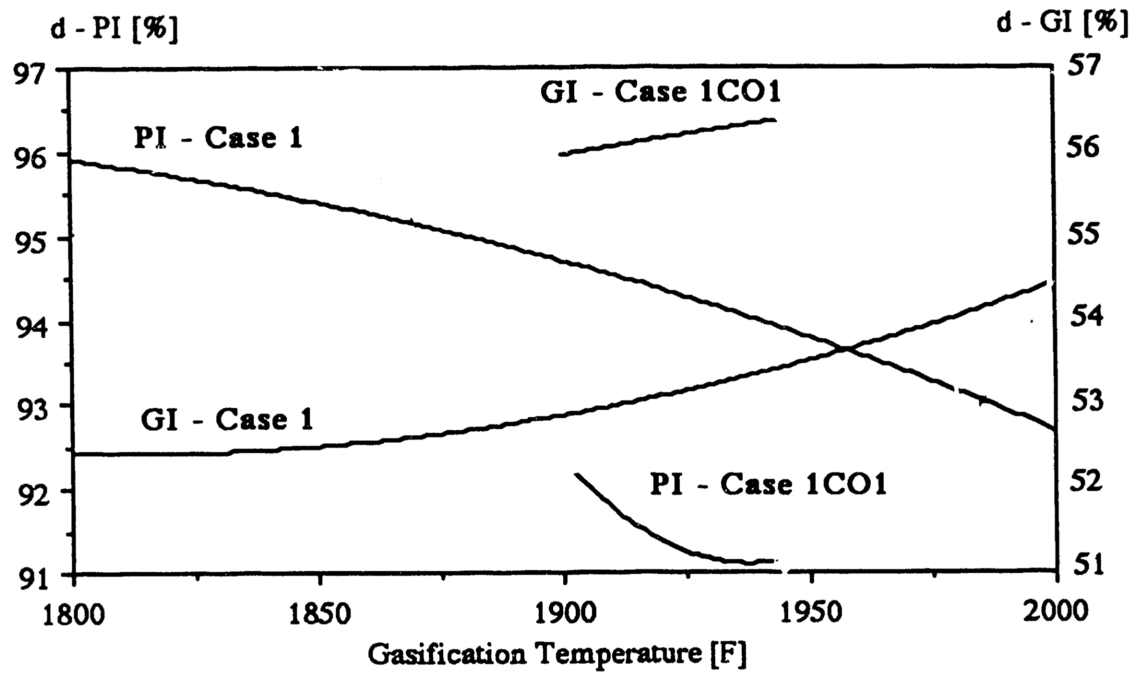

Figure A-3 Comparison of the relative cost difference d calculated for the Gasification Island (GI) and the Power Island (PI) for Cases 1 and 1CO1. The d values are given on the left axis for PI and on the right axis for GI.

Note: With increasing gasification temperature, the d value for the gasification island increases while it decreases for the power island. Compared with Case 1CO1, the original Case 1 has lower d values for the gasification island but higher values for the power island. The trends expressed in the above curves help to understand the development of the curves shown in Figure A-1.

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