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**SYSTEM STUDY OF AN MHD/GAS TURBINE
COMBINED-CYCLE BASELOAD POWER PLANT**

by

Kurt D. Annen

August 1981

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ABSTRACT

The MHD/gas turbine combined-cycle system has been designed specifically for applications where the availability of cooling water is very limited. The base case systems which were studied consisted of an MHD plant with a gas turbine bottoming plant, and required no cooling water. The gas turbine plant uses only air as its working fluid and receives its energy input from the MHD exhaust gases by means of metal tube heat exchangers. In addition to the base case systems, vapor cycle variation systems were considered which included the addition of a vapor cycle bottoming plant to improve the thermal efficiency. These systems required a small amount of cooling water. The MHD/gas turbine systems were modeled with sufficient detail, using realistic component specifications and costs, so that the thermal and economic performance of the system could be accurately determined.

Three cases of MHD/gas turbine systems were studied, with Case I being similar to an MHD/steam system so that a direct comparison of the performances could be made, with Case II being representative of a second generation MHD system, and with Case III considering oxygen enrichment for early commercial applications. The systems are nominally 800 MW_e to 1000 MW_e in size. The results show that the MHD/gas turbine system has very good thermal and economic performances while requiring either little or no cooling water. Compared to the MHD/steam system which has a cooling tower heat load of 720 MW, the Base Case I MHD/gas turbine system has a heat rate which is 13% higher and a cost of electricity which is only 7% higher while requiring no cooling water. The

Case I vapor cycle variation systems have thermal and economic performances which approach and even exceed those of the MHD/steam system while having a substantially lower cooling water requirement. Case II results show that an improved performance can be expected from second generation MHD/gas turbine systems. Case III results show that an oxygen enriched MHD/gas turbine system may be attractive for early commercial applications in dry regions of the country. Other aspects of the system study included an investigation of the sensitivity of the system to key variables, an irreversibility analysis, and an analysis of the nitric oxide emissions from the system.

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NOMENCLATURE

A	area (m^2)
A_{ff}	free flow area (m^2)
A_{fr}	frontal area (m^2)
a_λ	spectral absorption coefficient (m^{-1})
B	magnetic field strength (Tesla)
C	cost (\$)
	channel loft parameter
	thermal capacity rate (W/K)
C_1, C_2	constants
C_r	matrix capacity rate (W/K)
c_p	specific heat at constant pressure ($\text{J/kg} \cdot \text{K}$)
D	diameter (m)
d	tube diameter (m)
d_h	hydraulic diameter (m)
E_x	axial electric field (V/m)
E_y	transverse electric field (V/m)
f	friction factor
	denotes function
G	mass flux ($\text{kg/m}^2 \cdot \text{s}$)
H	spacing between electrode walls (m)
H_c	enthalpy of the core relative to the wall (J/kg)
h	enthalpy (J/kg)
	convective heat transfer coefficient ($\text{W/m}^2 \text{K}$)
$\bar{h}$	length averaged convective heat transfer coefficient ($\text{W/m}^2 \text{K}$)
h_o	total enthalpy (J/kg)
$\dot{I}rrev_E$	energy measure of irreversibility rate (W)
$\dot{I}rrev_{E,Av}$	availability energy measure of irreversibility rate (W)
$\dot{I}rrev_{E,IS}$	ideal system energy measure of irreversibility rate (W)
$\dot{I}rrev_S$	entropy measure of irreversibility rate (W/K)
J_x	axial current density (A/m^2)
J_y	transverse current density (A/m^2)

K	electrical loading parameter
k	thermal conductivity ($\text{W/m} \cdot \text{K}$)
	reaction rate coefficient
L	length (m)
λ	distance from the electrode wall to the center of the MHD channel (m)
M	Mach number
	mass flow rate (kg/s)
$\dot{m}$	mass flow rate (kg/s)
N	number of heat exchanger passes
N_{tu}	number of heat exchanger transfer units
n	species concentration (m^{-3} or cm^{-3})
P	pressure (atm or N/m^2)
	power (W)
Pr	Prandtl number
q	heat transfer rate (W)
q''	heat transfer rate per unit area (W/m^2)
q'''	heat transfer rate per unit volume (W/m^3)
R	radius (m)
	specific gas constant ($\text{J/kg} \cdot \text{K}$)
R_{foul}	fouling thermal resistance ($\text{m}^2 \text{ K/W}$)
$R_{contact}$	contact thermal resistance ($\text{m}^2 \text{ K/W}$)
Re	Reynolds number
r	flow passage spacing (m)
	boundary layer electrical resistance ($\text{ohms} \cdot \text{m}^2$)
S_D	design strength (psi)
St	Stanton number
T	temperature (K)
T_{ex}	exhaust stream temperature (K)
T_f	final state point temperature (K)
T_{fs}	final state point temperature of an isentropic process (K)
$T_{\lambda m}$	log-mean temperature (K)
T_0	temperature of the environment (K)
T_{WF}	weighting factor temperature (K)

t	thickness (m)
U	overall thermal conductance ($\text{W/m}^2 \text{ K}$)
u	velocity (m/s)
V	volume
V_{arc}	voltage drop across on arc (V)
V_{drop}	total boundary layer voltage drop on one wall (V)
V_{fd}	fluid dynamic voltage drop on one wall (V)
$\dot{W}_k$	rate of work (W)
w	distance from insulator wall to the center of the MHD channel (m)

Greek Symbols

α	absorptivity
	ratio of heat transfer area of one side to total heat exchanger volume (m^2/m^3)
β	Hall parameter
γ	ratio of fin area to total heat transfer area
Δ	denotes difference of a quantity
	thermal boundary layer thickness (m)
Δ_2	enthalpy thickness of a boundary layer (m)
δ	momentum boundary layer thickness for 99% of core velocity (m)
δ_o	velocity displacement thickness for constant density (m)
δ_2	momentum thickness of momentum boundary layer (m)
δx	small axial step (m)
ϵ	heat exchanger effectiveness
	emissivity
η	efficiency
λ	wavelength ($\text{\AA}$)
μ	viscosity ($\text{kg/m} \cdot \text{s}$)
ρ	density (kg/m^3)
σ	electrical conductivity (mhos/m)
	ratio of free-flow area to frontal area
	Stephan/Boltzman constant

τ	shear stress (N/m^2)
ω	ratio of thermal boundary layer thickness to momentum boundary layer thickness

Subscripts

Air Sep.	air separation plant
air	air
av	average
c	cold side
	core
cer	ceramic
coal	coal
comm	common side
Comp	compressor
e	equilibrium
g	gas
h	hot side
i	inner side
in	inlet
K	potassium
max	maximum
min	minimum
MHD	MHD
o	outer side
out	outlet
p	pass (of heat exchanger)
si	inner shell
so	outer shell
t, th	thermal
turb	turbine
w	wall

CHAPTER 1

INTRODUCTION

1.1 Motivation

In many areas of the United States, the restricted availability of cooling water can be a severe limitation to the siting of an electric power plant. For utilities which are faced with such siting problems, the low cooling water requirement of a Brayton cycle plant is very attractive, but the need for a clean fuel and the relatively low cycle efficiencies usually associated with these plants have removed them from consideration for baseload applications. Gas turbine/steam combined-cycle plants overcome the problem of low thermal efficiency and can also use coal as a fuel by first converting it to a low-Btu fuel gas, but they have a cooling water requirement which is considerably greater than that of a Brayton cycle alone. MHD/steam combined cycle plants, which use the Brayton cycle MHD plant with a steam bottoming plant, have a cooling water requirement which is slightly greater than that of gas turbine/steam plants even though they also have a modestly higher thermal efficiency. As can be seen in Table 1.1, both combined-cycle systems have a significantly lower ratio of cooling tower heat rejection to net power output compared to the conventional supercritical steam plant. Even a 50% reduction in cooling water requirement from that of a conventional steam plant, though, may not be sufficient to enable a large power plant to be sited in dry regions of the country. However, the combination of a Brayton cycle MHD plant with a gas turbine bottoming plant results in a system which retains the advantages of high thermal efficiency and the direct use of coal while adding the unique capability of being able to eliminate the requirement for cooling water. This MHD/gas turbine combined-cycle system, which is specifically designed for areas which have little or no available cooling water, is thoroughly investigated in this study in order to compare its thermal and economic performance with the MHD/steam system and to assess its potential performance with second generation MHD plant components.

Table 1.1

Cooling Tower Heat Loads of Electrical Power Generation Systems

<u>Study</u>	<u>Coal Thermal Input</u>	<u>Cooling Tower Heat Rej.</u>	<u>Net Electrical Power</u>	<u>η th</u>	<u>Cooling Tower Heat Rej. $\div$ Net Elec. Power</u>
EPRI/Fluor, Combined-Cycle Gas Turbine, Case MXSC ⁽¹⁾	2985 MW	929 MW	1212 MW	40.6%	0.767
EPRI/Fluor, Combined-Cycle Gas Turbine, Case EAHC ⁽¹⁾	2995 MW	978 MW	1213 MW	40.5%	0.806
ECAS II - GE, Combined-Cycle Gas Turbine ⁽²⁾	1447 MW	424 MW	585 MW	39.6%	0.725
EPRI/Westinghouse, MHD/Steam Combined- Cycle, Plant Design 1 ⁽³⁾	1966 MW	734 MW	859 MW	43.7%	0.855
PSPEC - Avco, MHD/Steam Combined-Cycle, Reference Plant 1 ⁽⁴⁾	2200 MW	817 MW	936 MW	42.5%	0.873
PSPEC - G. E., MHD/Steam Combined- Cycle, Base Case 1 ⁽⁵⁾	2872 MW	1093 MW	1189 MW	41.4%	0.919
3675psig/1000°F/1000°F Supercritical Steam Plant (oil-fired, no SO ₂ or NO _x control)	1877 MW	1048 MW	737 MW	39.2%	1.422

1.2 Previous Work

The MHD/steam system has been investigated by many system studies in recent years⁽²⁾ - (6). The following salient conclusions which are applicable to the present work can be drawn from these studies.

- Combined-cycle thermal efficiencies are in the range of 37% to 45% for first generation systems, and in the range of 45% to 50% for second generation systems.

- Although considerable uncertainty exists in their capital cost and cost of electricity, MHD/steam systems appear to be economically competitive with both conventional power generation systems and other advanced technology systems.

- Seed regeneration is preferable to flue gas scrubbers as a means of sulfur emissions control, especially when low sulfur coal is used as the fuel.

- Use of a directly-fired high temperature air heater substantially improves the thermal efficiency of the MHD/steam system over that which would be obtained if an indirectly-fired air heater or oxygen enrichment were used.

- Oxygen enrichment appears to be an attractive alternative to the use of a high temperature air heater for early MHD/steam systems, offering greater reliability at nearly the same cost of electricity.

- Staged combustion and slow cooling of the combustion gases in the radiant boiler can control nitric oxide (NO_x) emissions to meet current emission standards.

- Diffuser pressure recovery is expected to be in the range of 50% to 70%.

- Western subbituminous coal is optimally dried to 5% moisture or less; Illinois #6 coal is optimally dried to less than 2% moisture.

1.3 Objectives and Modeling Philosophy

The primary objective of this work is to provide a valid assessment of the thermal and economic performance of the MHD/gas turbine combined-cycle system. Included in this assessment is an analysis, beyond the scope of a standard system study, of some important considerations of the system.

The system and individual components are modeled in sufficient detail so that the objective of an accurate assessment of cost and performance will be assured. Previous system studies and component studies are utilized, when appropriate, as a source of cost, performance, and design information.

1.4 Cases Considered in System Study

Three cases are considered in this work. Case I is an MHD/gas turbine system which closely parallels that of the recent EPRI/Westinghouse MHD/steam system study ⁽³⁾ for the purpose of providing a direct comparison between the two systems. This case uses a directly-fired high temperature air heater to preheat the oxidizer. Case II is an MHD/gas turbine system which has some advanced technology MHD components, which has been designed to meet the latest emissions standards, and which uses Montana Rosebud coal which is appropriate for the water-limited regions of the country. Case III uses oxygen enrichment of the oxidizer to replace the directly-fired air heater used in the first two cases.

The addition of a small vapor cycle bottoming plant to the MHD/gas turbine system is included as a variation to all three cases. This variation is appropriate for locations where a limited amount of cooling water is available.

One major distinction between Case I and Cases II and III is in the method of calculating the cost of electricity of the systems. Case I follows the method used in the MHD/steam system study ⁽³⁾ and calculates the "first year commercial operation" cost of electricity. Cases II and

III follow the method more commonly used in recent system studies (4) (5) and calculate a "levelized" cost of electricity.

1.5 Report Organization

This report is divided into five chapters plus nine appendices. Following this introductory chapter is Chapter 2, which gives descriptions of the MHD/gas turbine combined-cycle systems considered in the three cases. Chapter 3 presents the thermal and economic performance of the three cases, including the vapor cycle variations. The results for Case I are compared with the performance of an MHD/steam system. The response of the system performance to several important variables is presented, and the results of an investigation into some of the more important sources of irreversibility in the combined cycle system are given. Results of a detailed analysis of nitric oxide decomposition for NO_x control are given for Cases II and III. In Chapter 4, models of the system components are presented. Conclusions are presented in Chapter 5. The appendices include a thorough description of the cost analysis as well as a presentation of detailed investigations which supplement the system study work.

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CHAPTER 2

SYSTEM DESCRIPTION

2.1 System Overview

In the MHD/gas turbine combined-cycle system, the MHD plant is nearly identical to that in an MHD/steam plant with the main difference being that the diffuser and radiant heater are cooled by compressed air instead of mixed phase water. A schematic of this system for Base Case I is shown in Figure 2.1. The gas turbine plant in this system, which uses only air as the working fluid, consists of two separate gas turbine cycles to maximize efficiency and receives its energy input from the exhaust gases of the MHD plant. The first turbine cycle (designated turbine cycle A) has one reheat and has the higher turbine inlet temperature. The second cycle (turbine cycle B) has a lower turbine inlet temperature and does not utilize reheat. The air of turbine cycle B is used to cool the diffuser of the MHD plant. MHD combustion air is used to cool the radiant heater. For the vapor cycle bottoming plant variations on the base cases, the ammonia working fluid is heated by the low temperature energy of the MHD and gas turbine exhaust streams which is not utilized by the gas turbine plant.

In the following section of this chapter, the system flow train will be described in greater detail for each base case and variation. In section 2.3, a succinct description will be given of the important components in the system.

2.2 Description of System Flow Train

Base Case I

The MHD plant in Base Case I is very similar to that of Plant Design 1 of the EPRI/Westinghouse (EPRI/W) MHD/steam system study.⁽³⁾ In particular, the following features are common to both plants: a directly-fired high temperature air heater with an air outlet temperature of 1644 K; a diffuser with a pressure recovery of 0.50; the use of Illinois #6 coal; a thermal input of 1970 MW_t; a combustor

Figure 2.1 System Flow Schematic for Base Case I (and Base Case II).

stoichiometry (defined as $[\text{Air/Fuel}]_{\text{actual}}/[\text{Air/Fuel}]_{\text{stoichiometric}}$) of 0.95; and the use of flue gas desulfurization to limit the sulfur emissions to 1.2 lb/MBtu.

The MHD combustion air for Base Case I is compressed in an inter-cooled compressor and heated in three different heat exchangers, as shown in Figure 2.1. The air first is heated in the high effectiveness recuperator HXM1 by the exhaust gas of turbine cycle A and then is heated to 1000 K by the radiant heater. The oxidizer is finally heated to its maximum temperature of 1644 K in the directly-fired high temperature air heater, after which it flows to the two-stage combustor. The first combustor stage receives 54% of the oxidizer and the total coal input, resulting in a first stage stoichiometry of 0.50. This stoichiometry is held to a low value to maximize the amount of slag which is rejected in the first stage. The second combustor stage receives the remaining oxidizer and the potassium seed. This produces a combustion plasma with a stoichiometry of 0.95 and a potassium seeding level of 1% by mass. The plasma is accelerated to a Mach number of 0.9 in the nozzle before entering the MHD channel. Energy is then extracted from the gas in the form of DC electrical power by charged particle interaction with a 6T magnetic field at a relatively high electrical load parameter of 0.8. The gas exits the MHD channel and enters the diffuser with sufficient total pressure so that with a pressure recovery of 0.50 the static pressure at the end of the diffuser is 1.03 atm. The next component in the flow train is the radiant heater, which slowly cools the gas to nominally 2000 K. During this slow cooling process the NO concentration in the gas is reduced to a level which enables it to meet the applicable NO_x emission standard. Following the radiant heater, the gas flows through the directly-fired high temperature air heater (HTAH) system. Immediately downstream of the HTAH, secondary combustion air is added to the flow, raising the stoichiometry of the gas to 1.05 (5% excess air). The secondary air is taken from the reheat turbine exhaust of turbine cycle A, thus directly regenerating part of the energy of the turbine exhaust gas. The combustion gas next enters the gas turbine plant where it first transfers energy in the metal tube recuperator HXA3 to heat the

intermediate pressure air for the reheat turbine and then transfers energy in recuperator HXA2 to heat air for the high pressure turbine of turbine cycle A. The final heat exchanger in the gas turbine plant in which energy is transferred from the MHD gas is recuperator HXB2, which heats the air of turbine cycle B. After exiting the gas turbine plant, the MHD gas flows through the electrostatic precipitator in which coal ash and potassium seed are removed and transported to the seed recovery system. Part of the gas is then diverted for coal drying, after which the gas streams are recombined and passed through the flue gas desulfurization process before being exhausted to the stack.

The gas turbine plant in Base Case I consists of two Brayton cycles, turbine cycle A which uses reheat and exhaust gas regeneration and turbine cycle B which uses only exhaust gas regeneration. The air for turbine cycle A is compressed in an intercooled compressor and then heated by exhaust gas regeneration in the finned circular tube recuperator HXA1. The air is next heated in the bare tube recuperator HXA2 to 1040 K and expanded in the high pressure turbine. Then the air is reheated in the bare tube recuperator HXA3 and expanded in the reheat turbine. A fraction of the reheat turbine exhaust flow is extracted for use as secondary combustion air and the rest of the exhaust flow passes through recuperators HXA1 and HXM1 before being sent to the stack. The air for turbine cycle B is compressed in an intercooled compressor and then heated by regeneration of the turbine exhaust gas energy in recuperator HXB1. The air is next heated by the MHD gas in recuperator HXB2 and then heated to its maximum temperature by cooling the diffuser of the MHD plant. After being expanded through the turbine, the air passes through recuperator HXB1 and is exhausted to the stack.

The MHD channel and combustor for Base Case I are cooled by means of liquid-coupled heat exchangers, in which the energy transferred to the pressurized coolant is dumped to the atmosphere by dry, air-cooled heat exchangers. The MHD/gas turbine system for Base Case I therefore requires no cooling water since a vapor cycle plant is not used and since the compressor intercoolers and the MHD channel and combustor cooling systems utilize air-cooled heat exchangers.

Case I, Variation 1

In this variation, which has the lowest cooling water requirement of the four variations considered for Case I, an ammonia vapor cycle bottoming plant utilizes energy remaining in the MHD exhaust gas after leaving the gas turbine plant. The energy remaining in the gas turbine exhaust streams is not utilized by the vapor cycle plant in this variation. Figure 2.2 gives a schematic diagram of the system. Three minor changes make the configuration of the MHD plant different from that of Base Case I: 1) the combustor is cooled by supercritical ammonia from the vapor cycle plant; 2) the MHD air compressor is not intercooled, eliminating intercooler ICM; 3) recuperator HXMI is eliminated, so the MHD combustion air is now heated first in the radiant heater. Also in this variation the MHD combustor pressure has been lowered slightly, resulting in a shorter channel.

The configuration of the gas turbine plant is the same as in Base Case I, although the effectiveness of the heat exchangers and the pressure ratios of the turbines and compressors are slightly changed. The main change in the system from Base Case I occurs downstream of the gas turbine plant, in the addition of the ammonia bottoming plant. The remainder of the gas which is not required for coal drying now passes through the ammonia heater before going to the cleanup process and the stack. The ammonia plant itself uses a supercritical vapor cycle and regenerates the energy in the ammonia turbine exhaust to reduce the amount of energy rejected to the cooling tower.

Case I, Variation 2

This variation has the second lowest cooling water requirement of the four variations for Case I. As shown by the system schematic in Figure 2.3, the vapor cycle bottoming plant utilizes the energy of the MHD gas and the turbine cycle B exhaust gas. The MHD plant for this variation is exactly the same as for Variation 1, but the gas turbine plant now is different. In turbine cycle B, the compressor intercooler ICB and the recuperator HXB1 have been eliminated. The turbine exhaust gas of this

Figure 2.2 System Flow Schematic for Case I, Variation 1 (and Case II, Variation 1).

Figure 2.3 System Flow Schematic for Case I, Variation 2 (and Case II, Variation 2).

cycle now flows directly to the ammonia bottoming plant. The configuration of turbine cycle A is unchanged. In this variation, the ammonia flow rate is approximately 50% greater than in Variation 1, and therefore the cooling water requirement is increased by roughly the same factor.

Case I, Variation 3

This variation, schematically shown in Figure 2.4, has the second highest cooling water requirement of the four variations for Case I. In this system, the vapor cycle bottoming plant utilizes the energy of the MHD gas and the turbine cycle A exhaust gas. The MHD plant is the same as in the previous two variations, and the turbine cycle B configuration is the same as in Base Case I and Variation 1, but the compressor intercooler ICA for turbine cycle A has been eliminated. After the reheat turbine exhaust gas of cycle A passes through recuperator HXA1, it then flows to an ammonia heater in the vapor cycle bottoming plant. The energy input to the bottoming plant is greater than in Variation 2, largely due to the greater air flow in turbine cycle A relative to turbine cycle B.

Case I, Variation 4

This variation, schematically shown in Figure 2.5, has the highest cooling water requirement of the four variations for Case I. In this system the vapor cycle plant utilizes the energy of all three exhaust streams: the MHD gas, and the turbine exhaust from cycles A and B. The MHD plant is the same as in the other Case I variations. For the gas turbine plant, the configuration of turbine cycle A is the same as in Variation 3 and the configuration of turbine cycle B is the same as in Variation 2. In this variation therefore, none of the compressors are intercooled and the recuperators HXM1 and HXB1 which are used in Base Case I are now eliminated. The size of the bottoming plant in this variation is approximately 2 1/2 times greater than in Variation 1.

Figure 2.4 System Flow Schematic for Case I, Variation 3 (and Case II, Variation 3).

Figure 2.5 System Flow Schematic for Case I, Variation 4 (and Case II, Variation 4).

Base Case II

The MHD plant for Base Case II contains components that are currently anticipated to be part of a second generation MHD system. A directly-fired air heater is used to preheat air to 1800 K, instead of the 1644 K level used in Case I. The diffuser is taken to have a pressure recovery of 0.65, instead of 0.50 for Case I. Potassium seed regeneration is used to control sulfur emissions, instead of flue gas desulfurization used in Case I. And Montana Rosebud coal is used, instead of Illinois #6 coal, with a slightly higher thermal input.

The flow train of the MHD plant for Base Case II is the same as that for Base Case I. The gas turbine plant configuration also is the same as that for Base Case I, with heat exchanger effectivenesses and turbine and compressor pressure ratios being somewhat different. As in Base Case I, this system has no cooling water requirement.

Case II, Variations 1-4

The flow train configurations for Variations 1 through 4 are the same as their respective variations in Case I. As in Case I, the combustor pressure for the vapor cycle variations is slightly lower than the Base Case II combustor pressure.

Base Case III

Base Case III considers oxygen enrichment of the MHD oxidizer, which makes high combustion flame temperatures possible without the need to preheat the oxidizer to a high temperature in the directly-fired high temperature air heater. This is viewed as an attractive alternative for early MHD systems because the directly-fired high temperature air heater is presently considered to be an advanced technology component. The oxygen mole fraction in the oxidizer is 0.35. This value was determined in other work ⁽⁴⁾ to approximately maximize MHD/steam system performance.

A system schematic for Base Case III is shown in Figure 2.6. The MHD plant is similar to that for Base Case II, with the main differences between the two being that the directly-fired HTAH has been eliminated and a separate air separation facility for oxygen enrichment has been added. Because of oxygen enrichment, the MHD gas flow rate is considerably lower than that of Case II even though the coal type and combustion stoichiometry are the same. The diffuser and radiant heater for Base Case III cool the MHD gas to a lower temperature than in Case I and II in order to reduce the severity of the gas side environment in the downstream heat exchangers. For partially the same reason, and for the additional reason of preventing nitric oxide formation, the secondary combustion air is not added until after the combustion gas exits the recuperator HXA2.

The configuration of the gas turbine plant is the same as in Base Case I and II, but the effectivenesses of the heat exchangers are generally higher. One other difference from the previous base cases is that in Base Case III the air flow rate in turbine cycle B is greater than that for turbine cycle A. This is due to the higher MHD gas temperature at the inlet of recuperator HXB2 as well as the greater amount of energy transferred to turbine cycle B air in the diffuser.

Case III, Variation 1

Only one vapor cycle variation was considered for Case III, and this involved the utilization by the ammonia bottoming plant of the energy in the exhaust of the MHD and turbine cycle A gas streams. For this variation, which is shown in Figure 2.7, the MHD plant is the same as that for Base Case III with the exception that the combustor is cooled by supercritical ammonia, and that the recuperator HXM1 and the MHD air compressor intercooler ICM are eliminated. Turbine cycle A configuration is changed from Base Case III by the removal of the intercooler for the compressor. The cooling water requirement for this variation is nearly the same as that for Variation 2 of Case II.

2.3 Component Descriptions

A brief description of the system components is presented in this section. Chapter 4 contains a more detailed presentation of these components and other parts of the system.

Figure 2.6 System Flow Schematic for Base Case III

Figure 2.7 System Flow Schematic for Case III, Variation 1

MHD Channel

The MHD channel is of the segmented Faraday design. The channel loft is specified by $C \equiv u\Delta u/(\Delta P/\rho) = 0.10$. Constraints on the electrical parameters which were used in the channel design are $E_y < 4.0$ kV/m for the transverse field, $E_x < 3.5$ kV/m for the axial field, and $J_y < 1.0$ A/cm² for the average transverse current density. The axial field constraint was taken from the EPRI/W system study.⁽³⁾ The other constraints were taken from reference 4. The electrical load parameter, defined using the average E_y between the anode and cathode of an electrode pair and which includes the electrode voltage drops, was set at 0.80 except when it was forced to a lower value by the transverse field constraint. The wall temperature of the MHD channel is 2000 K.

Magnet

The magnet is assumed to have a constant field strength of 6T except at the MHD channel entrance and exit. At each end, the field is tapered to a field strength of 5T over 1.5 m of axial length.

Combustor and Nozzle

The combustor is of a two-stage design with a stoichiometry of 0.50 in the first stage. The second stage stoichiometry is 0.95 for Case I and 0.935 for Cases II and III. The liquid slag rejection in the first stage is taken to be 95%. This yielded overall ash rejection values of 81%, 85%, and 92% for Cases I, II, and III respectively. The combustor heat loss is set at 4% of the coal HHV thermal input, and the combustor $\Delta P/P$ is set at 0.047. Both of these values are the same as those which were used in the EPRI/W study.⁽³⁾ The coal thermal input to the combustor, based on the HHV, is 1970 MW for Case I and 2140 MW for Cases II and III. The combustor pressure ranged from 6.2 atm for the Case I vapor cycle variations to 7.0 atm for Case III.

The seeded combustion gas is accelerated to Mach 0.9 in the nozzle, with a nozzle cooling loss of 3 MW.

Diffuser

The diffuser is conical in shape and has a pressure recovery of 0.50 for Case I and a recovery of 0.65 for Cases II and III. The wall temperature of the ceramic-lined diffuser is set at 2000 K. Air-cooling

of the diffuser is accomplished by compressed air flowing axially through the annular passage formed by the diffuser wall on the inner side and a metal shell on the outer side. Periodically spaced metal plates extend between the inner and outer annular surfaces to limit stresses imposed on the surfaces by the compressed air.

Radiant Heater

The radiant heater is cylindrical in shape and has a diameter of 16 m for Cases I and II and 14 m for Case III. Its axial length ranges from 45 m for Base Case I to 55 m for Base Case III. The diffuser, which is 8 m in diameter at the exit, is joined to the radiant heater by an abrupt expansion which results in the separation of the gas from the walls at the radiant heater inlet. This establishes a recirculation zone which will extend approximately half way into the radiant heater. The wall temperature of the ceramic-lined heater is specified to be 1600 K at the inlet, rising to 2000 K in the region of flow reattachment and falling to 1900 K at the heater exit. The purpose of the hot wall temperature, which limits the wall heat flux, is twofold: it provides a long residence time for NO decomposition, and it minimizes the problem of frozen seed in the directly-fired HTAH by providing it with a high gas inlet temperature of nominally 2000 K. The air cooling design of the radiant heater is the same as the design for the diffuser.

High Temperature Air Heater

The design of the directly-fired HTAH closely follows the FY-80 design by FluidDyne Engineering Corp. for a baseload MHD system.⁽⁷⁾ This is a periodic flow system which has 30 active heater units with beds of cored ceramic bricks. At any one time, 20 units operate on the gas cycle, 8 units operate on the air cycle, and 2 units operate on the pressurization/depressurization cycle. The air outlet temperature for Case I is 1644 K and the gas outlet temperature is 1496 K. The gas and air inlet temperatures are 1996 K and 1000 K respectively. The air outlet temperature for Case II is 1800 K and the gas outlet temperature is 1389 K. The gas and air inlet temperatures for Case II are 2000 K and 1000 K respectively. The high temperature air heater is not used in Case III.

Compressors and Intercoolers

The compressors in this study have a compressor efficiency of 88%. The compressors, when intercooled, have one intercool stage with two compression stages.

The intercoolers are finned circular tube air/air heat exchangers. Draft fans are used to provide the required ambient air flow for intercooling. The intercooler effectivenesses range from 0.52 to 0.74 and the capacity ratio (minimum capacity rate/maximum capacity rate) is fixed at 1.0.

Turbines

The turbines in this study have a turbine efficiency of 0.90. Since the turbine inlet temperatures range from 887 K to 1107 K (1137°F to 1533°F), uncooled turbine blades should be feasible.

DC/AC Inversion Units

The conversion of the MHD generator power from DC to AC will first involve DC current consolidation and voltage stepup, and then conversion to AC power by inverters. The electrical conversion efficiency of this process is taken to be 98.5%, which is the value used in the EPRI/W study.⁽³⁾

Generators

The efficiency of the generators is taken to be 98.5%, which is the value used in the EPRI/W study.⁽³⁾

Heat Exchangers

The recuperative heat exchangers used in the gas turbine plant can be divided into two categories, MHD gas/air and air/air heat exchangers. The air/air heat exchangers are tube and shell units with finned circular tubes. These units have effectivenesses ranging from 0.70 to 0.92. The MHD gas/air heat exchangers are also tube and shell units but with bare circular tubes. Fins are not used because of the slag and seed present in the MHD gas. Superalloy metals may be used in some or all of the tube passes of these heat exchangers. Effectivenesses of recuperators HXA2 and

HXB2 range from 0.60 to 0.72, while the effectiveness of HXA3 is held to low values of 0.29 to 0.40.

Coal Dryers

The coal drying units have an assumed drying efficiency (defined as $100 \times \text{heat input to the coal} \div \text{heat input to the dryer}$) of 50%, using MHD exhaust gas as the thermal energy source. The MHD gas exits from coal dryers at a temperature of 400 K. For Case I, Illinois #6 coal is dried to 0% moisture as in the EPRI/W study.⁽³⁾ For Cases II and III, Montana Rosebud coal is dried to 5% moisture.

Flue Gas Desulfurization

The flue gas desulfurization process used for Case I is the same as that used in the EPRI/W study.⁽³⁾ The process has a sulfur removal efficiency of 95%. The auxiliary power requirement for processing 87% of the flue gas for Case I is 17.5 MW.

Seed Recovery System

This system separates the seed from the ash and produces dried seed which is sent either to the combustor as in Case I or to the seed regeneration system as in Cases II and III. The energy requirement for a system which handles a seed flow of 10 kg/s is 0.32 MW_e of electrical power and 34.7 MW_t of steam.

Seed Regeneration

The seed regeneration system used for Cases II and III is the Westinghouse process,⁽⁸⁾ which is a two step reduction/oxidation scheme producing K₂CO₃ from K₂SO₄. For a system which handles a K₂SO₄ flow rate of 10 kg/s, the energy requirement is 7.91/MW_e of electrical power and 147 MW_t of coal input. The process generates 38.2 MW_t of steam which can be used to meet the steam energy requirement of the seed recovery system.

Ammonia Vapor Cycle Bottoming Plant

The ammonia vapor cycle bottoming plant is patterned after that used in the ECAS II closed-cycle gas turbine system (NASA Modified Case 4).⁽²⁾ The pinch point temperature difference for the ammonia heater is 18 K, the condenser outlet temperature is 360 K (90°F), the condenser $\Delta P/P$ is 0.09, the turbine efficiency is 0.83, and the pump efficiency is 0.70. The effectiveness of the condensed ammonia preheater, which regenerates energy in the superheated ammonia turbine exhaust, was set at 0.80. The ammonia turbine inlet pressure for all cases is 20 MPa (2900 psia) and the condenser pressure is 1.27 MPa (185 psia).

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CHAPTER 3

SYSTEM THERMAL AND ECONOMIC PERFORMANCE

3.1 Performance Modeling and Optimization Procedure

In modeling the thermal performance of the MHD/gas turbine system, an effort was made to account for all important phenomena which affect the system performance. The thermochemical and transport properties of the combustion gas and of air were modeled as accurately as possible, including multiphase models where appropriate. A quasi 1-D model for the MHD channel was used which considered both Joulean heating in the electrode wall boundary layer and extended Hartmann flow in the insulating wall boundary layer. A comprehensive analysis was made of radiative energy transport in the radiant heater. The gas turbine plant required less detail in modeling; however each component was treated in the depth necessary to allow an accurate assessment of thermal performance and cost to be made. For the heat exchangers, this involved a calculation of the hot and cold side heat transfer coefficient and friction factor and the minimum allowable tube wall thickness for each pass.

The optimization procedure consisted not only of determining the combination of values for system variables which minimized the cost of electricity (COE), but also determining the system configuration which would minimize the COE. Four system configurations were examined before the selection of the configuration which was used in the three base cases, and this configuration was changed in the optimization of the vapor cycle bottoming plant variations. The optimization of the system for the three base cases and their vapor cycle variations involved running over 2000 individual cases in which the values for 37 variables were varied. By interactively performing the optimization, it was possible to make an "educated" selection of the values of the variables. This capitalized on the learning process which occurred as trends and relationships between variables were recognized. Of the 37 system variables, 4 were for the MHD channel. These were the electrical load parameter, the combustor pressure, the channel inlet velocity, and the channel loft parameter. Of the remaining variables, which were for the gas turbine plant, 10 had a relatively greater effect on the system performance than the others.

These important variables were the effectivenesses of the heat exchangers and the pressure ratios of the compressors for cycles A and B and the reheat turbine for cycle A. The other variables had either a small effect on the system performance or had an optimum which was not sensitive to the values of the other variables. These less critical variables consisted mostly of the heat exchanger pressure drops, the number of passes in individual heat exchangers, and the number of compressor intercool stages. The optimized cases presented in this study, therefore, represent the result of a thorough parametric search and the best known performance for the system configuration.

3.2 Case I System Results

The system state points for Base Case I are presented in Figure 3.1 and the principal mass flows are summarized in Table 3.1. Both of the gas turbine cycles have a mass flow which is greater than that of the MHD plant, with turbine cycle A having the greatest mass flow rate.

The MHD channel performance is given in Table 3.2. The maximum axial field for the Base Case I channel was 2.9 kV/m, which is less than the constraint of 3.5 kV/m used in this study. The enthalpy extraction for the 16.4 m channel was 22.8% and the DC power output was 642 MW. Judged on the basis of the magnitude of the channel enthalpy extraction, this Base Case I channel is conservative in design compared to channels considered in other system studies. In particular, the Base Case I channel enthalpy extraction is significantly lower than the 24.9% enthalpy extraction of the channel in the Base Case 1 system of the GE PSPEC study⁽⁵⁾, and it is also lower than the 23.5% enthalpy extraction of the Reference Plant 1 channel of the Avco PSPEC study.⁽⁴⁾

The power generation and consumption for the MHD/gas turbine system are summarized in Table 3.3. In this table it can be seen that for Base Case I, the gas turbine plant generates 336 MW of power which is a little more than half of the power generated by the MHD channel. And in the gas turbine plant, turbine cycle A produces roughly three times the power output of turbine cycle B. The net MHD plant power (defined as MHD channel power - MHD compressor power) is 483 MW, or approximately 50% greater than the gas turbine plant power.

The design parameters for the heat exchangers in the gas turbine plant are presented in Table 3.4. The effectiveness of most of the heat exchangers has been optimized at values around 0.7, typifying a modest rather than high performance design. This indicates that the cost of the

Table 3.1

Case I - Summary of Principal Mass Flows

	<u>Base Case</u>	<u>Var. 1</u>	<u>Var. 2</u>	<u>Var. 3</u>	<u>Var. 4</u>
Coal Flow (as received) (kg/s)	79	79	79	79	79
MHD Compressor Air Flow (kg/s)	611	611	611	611	611
MHD Gas Combustion Flow (kg/s)	685	685	685	685	685
Secondary Combustion Air Flow (kg/s)	64	64	64	64	64
Turbine Cycle "A" Air Flow (kg/s)	1180	1172	1161	1199	1194
Turbine Cycle "B" Air Flow (kg/s)	806	808	808	808	809
Vapor Cycle Ammonia Flow (kg/s)	0	180	280	369	477
System Flow Schematic Fig. No.	2.1	2.2	2.3	2.4	2.5
(Page No.)	7	11	12	14	15

Table 3.2

Case I - MHD Channel Parameters

<u>Parameter</u>	<u>Base Case</u>	<u>Vapor Cycle Variations</u>
Coal Thermal Input to Combustor	1970 MW	1970 MW
Combustor Pressure	6.5 atm	6.2 atm
Channel Inlet Velocity	835 m/s	834 m/s
Channel Outlet Velocity	695 m/s	699 m/s
Maximum Axial Field Strength	2.9 kV/m	3.1 kV/m
Channel Length	16.4 m	14.5 m
Gross Power Output	642.4 MW _{DC}	625.1 MW _{DC}
Channel Heat Loss	75.1 MW	69.0 MW
Enthalpy Extraction	22.8 %	22.1 %
MHD Cycle Efficiency ($\frac{P_{MHD,AC} - P_{Comp}}{\dot{q}_{in}}$)	24.5 %	23.3 %
MHD Channel Isentropic Efficiency	79.2 %	79.2 %

Table 3.3

Case I Summary of Power Generation and Consumption

<u>Power Generation</u> (MW_{AC})	<u>Base Case</u>	<u>Var. 1</u>	<u>Var. 2</u>	<u>Var. 3</u>	<u>Var. 4</u>
MHD Power	632.8	615.7	615.7	615.7	615.7
Net MHD Power ($P_{\text{MHD}} - P_{\text{Comp}}$)	483.0	458.1	458.1	458.1	458.1
Turbine Cycle A Power	253.3	256.5	254.2	224.2	223.3
Turbine Cycle B Power	<u>82.8</u>	<u>94.1</u>	<u>78.1</u>	<u>92.3</u>	<u>79.7</u>
Total Gas Turbine Plant Power	336.1	350.6	332.3	316.5	303.0
Ammonia Plant Power	0.0	77.5	112.1	159.1	195.4
 <u>Power Consumption</u> (MW_{AC})					
MHD Compressor Power	149.8	157.6	157.6	157.6	157.6
Auxiliary Power	39.5	42.6	43.1	43.7	44.0
 Net Power Generation (MW_{AC})	781.9	843.6	859.4	890.0	912.5

Table 3.4

Base Case I Gas Turbine Plant Heat Exchanger Design Parameters

	<u>HXA1</u>	<u>HXA2</u>	<u>HXA3</u>	<u>HXB1</u>	<u>HXB2</u>	<u>HXM1</u>	<u>ICA</u>	<u>ICB</u>	<u>ICM</u>
Effectiveness	0.80	0.70	0.29	0.70	0.70	0.92	0.68	0.68	0.68
Heat Load (MW)	269	452	208	87	219	44	102	55	48
Capacity Ratio	0.96	0.71	0.74	1.00	1.00	0.55	1.00	1.00	1.00
$\Delta P/P$, Shell Side	.012	.020	.020	.010	.020	.008	.010	.010	.010
$\Delta P/P$, Tube Side	.012	.040	.035	.010	.035	.008	.015	.015	.015

heat exchanger surface area is sufficiently great to cause the trade off of system cost versus performance to optimize at only a moderate level of heat exchanger effectiveness. This trade off in particular applies to the heat exchangers HXA2, HXA3, and HXB2 which determine the heat input to turbine cycles A and B, and therefore greatly affect the power output of these cycles.

An energy flow diagram for Base Case I is shown in Figure 3.2 and a tabular summary of the energy balance for this case can be found in Table 3.5. From these, it can be seen that a large amount of energy, 689 MW, is rejected by the gas turbine and MHD exhaust streams. The temperature of these gas streams is quite high, being nearly 200 K or more above ambient temperature, as expected for Brayton cycles.

The thermal performance of Base Case I appears in Table 3.6. The system thermal efficiency of 39.7% compares favorably with the 36% figure of conventional steam plants and has the advantage of requiring no cooling water since only Brayton cycles are used in this system.

The exclusive use of Brayton cycles, however, limits the thermal performance of the system to efficiencies which are substantially lower than an MHD/steam system because of the Brayton cycle's inability to utilize low temperature energy. And it is due to the desire to improve the MHD/gas turbine system's thermal performance that the vapor cycle variations to the base case have been considered.

The system state points for vapor cycle Variation 1, which has an ammonia bottoming plant to utilize the energy in the MHD exhaust gas stream, are presented in Figure 3.3. The principal mass flows for all of the vapor cycle variations are summarized in Table 3.1. In the optimized configuration for Variation 1, the MHD gas outlet temperature from the gas turbine plant (after HXB2) is over 50 K greater than in the base case. The gas turbine plant mass flow rates for the vapor cycle variations are essentially unchanged from Base Case I. The ammonia bottoming plant mass flow increases by over a factor of 2.5 from Variation 1 to Variation 4.

The MHD channel performance for the vapor cycle variations, which differs slightly from that of the base case, appears in Table 3.2. Due to the increased efficiency of utilization of the energy in the MHD exhaust stream with the addition of the vapor cycle plant, the system performance is optimized with a slightly lower combustor pressure and shorter channel.

The power generation and consumption for Variations 1 - 4 are summarized in Table 3.3. As the vapor cycle power increases from Variation 1

Figure 3.2 Base Case I Energy Flow Diagram

Table 3.5

Case I Energy Balance Summary

<u>Energy In (MW)</u>	<u>Base Case</u>	<u>Var. 1</u>	<u>Var. 2</u>	<u>Var. 3</u>	<u>Var. 4</u>
Coal to Combustor	1970	1970	1970	1970	1970
<u>Energy Out (MW)</u>					
Net Electrical Power	781.9	843.6	859.4	890.0	912.5
Auxiliary Power (less ID fan Power)	32.6	34.3	34.7	35.3	35.6
Conversion Losses	12.5	13.5	13.7	14.1	14.6
Coal Dryers + Misc. Heat Losses	89.8	89.8	89.8	89.8	89.8
Intercooler Energy Rejection	207.3	157.5	102.9	39.4	0.0
Turbine Cycle A Exhaust	226.4	252.5	263.2	78.6	79.2
Turbine Cycle B Exhaust	152.8	146.4	56.3	163.7	56.7
MHD Exhaust	309.6	151.8	153.7	155.9	154.0
(Total Exhaust Stack Losses, A + B + MHD)	(688.8)	(550.7)	(473.2)	(398.2)	(289.9)
Cooling Towers	0	211.6	327.3	434.2	558.6
Generator (+ Combustor and Nozzle) Energy Rejection	157.1	69.0	69.0	69.0	69.0

Table 3.6

Case I. System Thermal Performance

	<u>Base Case</u>	<u>Var. 1</u>	<u>Var. 2</u>	<u>Var. 3</u>	<u>Var. 4</u>
Net Electrical Power Output (MW)	781.9	843.6	859.4	890.0	912.5
Cooling Tower Heat Rejection (MW)	0.0	212	327	434	559
Overall Thermal Efficiency (%)	39.7	42.8	43.6	45.2	46.3
Heat Rate (Btu/kWhr)	8599	7970	7824	7554	7368

State Point	T(K)	P(atm)	M(kg/s)	State Point	T(K)	P(atm)	M(kg/s)	State Point	K(T)	P(atm)	M(kg/s)
<u>MHD Air/Combustion Gas</u>				<u>Turbine Cycle "A" Air</u>				<u>Vapor Cycle Ammonia</u>			
1	298	1.00	610.5	19	400	0.92	157.1	33	594	5.43	807.7
2	549	7.17	610.5	20	380	0.92	592.3	34	816	5.20	807.7
3	1000	6.71	610.5	21	395	1.00	749.4	35	930	5.09	807.7
4	1644	6.51	610.5	<u>Turbine Cycle "B" Air</u>				36	641	1.01	807.7
5	1644	6.51	321.3	22	298	1.00	1171.9	37	477	1.00	807.7
6	1644	6.51	289.2	23	476	9.00	1171.9	<u>Vapor Cycle Ammonia</u>			
7	2779	6.20	685.1	24	709	8.87	1171.9	38	304	12.5	180.0
8	2634	3.86	685.1	25	1040	8.56	1171.9	39	313	201.4	180.0
9	2197	0.84	685.1	26	880	4.07	1171.9	40	342	199.4	50.9
10	2260	1.03	685.1	27	1040	3.91	1171.9	41	342	199.4	129.1
11	2000	1.03	685.1	28	767	1.02	64.3	42	620	197.4	129.1
12	1500	1.00	685.1	29	767	1.02	1107.6	43	620	197.4	50.9
13	1608	1.00	749.4	30	522	1.00	1107.6	44	620	197.4	180.0
14	1395	0.98	749.4	<u>Turbine Cycle "B" Air</u>				45	375	14.0	180.0
15	930	0.96	749.4	31	298	1.00	807.7	46	325	13.8	180.0
16	708	0.94	749.4	32	429	5.50	807.7				
17	708	0.94	157.1								
18	708	0.94	592.3								

Figure 3.3 State Point Diagram for Case I, Variation 1

to Variation 4, the total gas turbine plant power decreases. This is due to the change which is made in the configuration of each gas turbine cycle when its exhaust energy is utilized by the vapor cycle plant. In the base case the system performance is optimized by the use of intercooling and exhaust gas regeneration, whereas for the variations which utilize the turbine exhaust energy, the system performance is optimized by reverting to a simpler Brayton cycle which eliminates intercooling, and for turbine cycle B, also exhaust gas regeneration. For Variations 1 - 4, the vapor cycle plant accounts for 7% to 18% of the total power generation. The MHD compressor power is greater for the variations because of the elimination of the intercool stage for these systems. This, combined with the shorter MHD channel, accounts for the 25 MW reduction in net MHD plant power from the base case for the vapor cycle systems.

The design parameters for the gas turbine plant heat exchangers in the Variation 1 system are given in Table 3.7. The changes in heat exchanger effectivenesses from Base Case I are generally small.

An energy flow diagram for Variation 1 is shown in Figure 3.4, with an energy balance summary for this and the other vapor cycle variations appearing in Table 3.5. Comparing this energy flow diagram with Figure 3.2 for Base Case I, it can be seen that the energy in the MHD gas which flows out of turbine plant B is about 50 MW greater for Variation 1 than for Base Case I, while the MHD exhaust energy which goes to the stack is 150 MW less than that for Base Case I.

The thermal performance of the four vapor cycle systems appears in Table 3.6. The thermal efficiency of these systems increases from 42.8% to 46.3% as the cooling tower heat rejection increases from 212 MW to 559 MW. Thus, the addition of a vapor cycle bottoming plant offers a significant improvement in the thermal performance, but at the expense of imposing a cooling water requirement and additional complexity on the MHD/gas turbine system.

The economic performance of Base Case I and the four vapor cycle variations is presented in Table 3.8. The rules used in calculating the cost of electricity (COE) are given in Table 3.9 and a description of the COE calculation procedure is given in Appendix A. The COE of Base Case I is 46.9 mills/kWhr, while the vapor cycle variation with the greatest amount of cooling tower heat rejection has a COE which is 4.4 mills/kWhr

Table 3.7

Case I, Variation 1 Gas Turbine Plant Heat Exchanger Design Parameters

	<u>HXA1</u>	<u>HXA2</u>	<u>HXA3</u>	<u>HXB1</u>	<u>HXB2</u>	<u>ICA</u>	<u>ICB</u>
Effectiveness	0.84	0.68	0.29	0.78	0.66	0.70	0.70
Heat Load (MW)	287	432	213	138	193	104	54
Capacity Ratio	0.96	0.71	0.75	1.00	1.00	1.00	1.00
$\Delta P/P$, Shell Side	.015	.020	.020	.013	.020	.010	.010
$\Delta P/P$, Tube Side	.015	.040	.035	.013	.043	.015	.015

Figure 3.4 Case I, Variation 1 Energy Flow Diagram

Table 3.8

Case I System Economic Performance

	<u>Base Case</u>	<u>Var. 1</u>	<u>Var. 2</u>	<u>Var. 3</u>	<u>Var. 4</u>
Total Constructed Cost (mid-1978 \$)	851×10^6	877×10^6	874×10^6	894×10^6	901×10^6
Unit Capital Cost (\$/kW)	1089	1040	1017	1005	987
COE _{capital} (mills/kWhr)	31.95	30.51	29.83	29.48	28.96
COE _{fuel} *	10.03	9.30	9.12	8.81	8.59
COE _{O & M} *	4.89	4.89	4.89	4.89	4.89
COE _{total}	46.87	44.70	43.84	43.18	42.45
Cooling Tower Heat Rejection (MW)	0.0	212	327	434	559

* Not levelized

Table 3.9

Case I Economic Rules for Cost of Electricity Calculation

Contingency Factor	10%
A & E Service Factor	10%
Indirect Labor Factor (applied only to direct labor costs)	90%
Construction Time	6 1/2 yr
Factor for Escalation and Interest During Construction	1.699
Escalation Rate	6 1/2 %
Fixed Charge Rate	18%
Plant Capacity Factor	0.70
Coal Cost	\$ 1.166/MBtu
Method of COE Calculation	First year commercial operation (Fuel and O & M costs are not levelized)

lower. The total constructed cost of the system, which includes interest and inflation during construction, does not vary greatly between the base case and the variations. This is a result of the reduction in the cost of the gas turbine plant, due to its increasing simplicity, as the size of the vapor cycle plant increases from Variation 1 to Variation 4. The unit capital cost, on the other hand, decreases as the cooling tower heat rejection increases because of the increase in system power output.

From Table 3.8, it is apparent that the contribution to the total COE from the capital cost, COE_{capital} , is over three times the value of the fuel cost contribution, COE_{fuel} . And as the thermal efficiency of the system increases with the increasing size of the vapor cycle bottoming plant, the amount of reduction in COE_{capital} is roughly twice the reduction in COE_{fuel} . The fact that the system is optimized at a modest length of the MHD channel and modest effectivenesses of the gas turbine plant heat exchangers is indicative of the relatively greater sensitivity of the COE to the unit capital cost than to the fuel cost.

3.3 Comparison with MHD/Steam System

The EPRI/W Plant Design 1 MHD/steam system⁽³⁾ which is used in this comparison with the MHD/gas turbine Case I systems is similar to the matured technology design studied in ECAS⁽²⁾, but more conservative in performance and cost. So that the comparison with the Case I systems be as direct as possible, some adjustments were made to the MHD/steam system performance. The adjustments were necessary because the design and performance of the MHD channel for the MHD/steam system were different from those of this study. Since the MHD channel models which were used were quite similar, the difference appears to be mainly due to the transport properties of the MHD gas. The source of this difference may be, at lower temperatures, due to the modeling of the phosphorous thermochemistry, and at higher temperatures, due to the modeling of the iron thermochemistry. This study uses a value for the electron affinity of PO_2 of 2.82 eV,⁽⁹⁾ and does not include the FeO_2^- ion in the equilibrium model due to the lack of JANAF⁽¹⁰⁾ thermochemical data and the likelihood that it is present in insignificant concentrations.⁽¹¹⁾ Therefore, the higher performance of the MHD generator in the present study was applied to the MHD/steam study, resulting in an improvement in its performance and costs. The revised

MHD/steam power output and cost results are given in Table 3.10, and the procedure used in calculating the revised results is given in Appendix B.

Comparison of the MHD/steam system performance with that of Base Case I, also shown in Table 3.10, shows that the trade off for the elimination of the 722 MW cooling tower heat rejection of the MHD/steam system is a 7% increase in COE, slightly over 3 mills/kWhr, and a 5.0 percentage point decrease in thermal efficiency. The non-utilization of the exhaust gas energy, which enables the MHD/gas turbine system to have no cooling water requirement, has a lesser effect on the COE than on the thermal efficiency. This is a result of the 5.5% lower capital cost of the Base Case I MHD/gas turbine system. The relative contributions of capital and fuel to the COE are nearly the same for both systems.

The division of power generation for the MHD/steam system, given in the upper half of Table 3.10, shows that while the net MHD plant power of 470 MW is slightly less than the 483 MW of the MHD/gas turbine system due to a non-intercooled compression process, the steam plant power of 465 MW is substantially larger than the 336 MW power output of the gas turbine plant. The inability of the gas turbine plant to be as efficient as the steam plant in bottoming the MHD plant is the cause of the 11% decrease in thermal efficiency. This point is also borne out in a comparison of the total exhaust stack losses for the MHD/steam plant shown in the energy balance summary of Table 3.11 with those of the MHD/gas turbine system in Table 3.5. The MHD/steam stack losses of 295 MW are less than half of those for the Base Case I system. The difference in the auxiliary power requirements for the two systems, seen in Table 3.10, is due to the power requirements of the boiler feedwater pumps and the circulating water pumps in the MHD/steam system, as well as a somewhat higher draft fan power requirement.

A comparison of the cost accounts for the MHD/steam and Base Case I MHD/gas turbine systems is given in Table 3.12. The three accounts which show the greatest difference in cost between the two systems are: MHD Topping Cycle Equipment, Turbogenerator Units, and Boiler Plant Equipment. For the MHD/gas turbine system, the account which is equivalent to the Boiler Plant Equipment account includes coal and flue gas handling equipment, controls and instrumentation, and the gas turbine plant heat

Table 3.10

Comparison of MHD/Steam System Thermal and
Economic Performance* with MHD/Gas Turbine System Performance

	<u>MHD/Steam</u>	<u>MHD/Gas Turbine</u> <u>Base Case I</u>
Coal Thermal Input (MW)	1966.4	1970.0
Net Electrical Power Output (MW)	878.1	781.9
Net MHD Power (MW)	470.1	483.0
Bottoming Plant Power (MW)	465.3	336.1
Auxiliary Power Requirement (MW)	57.3	39.5
Thermal Efficiency (%)	44.7	39.7
Cooling Tower Heat Rejection (MW)	722	0
Total Constructed Cost (mid-1978 \$)	899×10^6	851×10^6
Unit Capital Cost (\$/kW)	1024	1089
COE _{capital} (mills/kWhr)	30.06	31.95
COE _{fuel}	8.92	10.03
COE _{O & M}	4.87	4.89
COE _{total}	43.85	46.87

*Adjusted for improved MHD generator performance.

Table 3.11

MHD/Steam Energy Balance SummaryEnergy In

Coal Energy In (MW)	1966.4
---------------------	--------

Energy Out

Electrical Power (MW)	878.1
-----------------------	-------

Auxiliary Power (MW)	57.3
----------------------	------

Conversion Losses (MW)	14.4
------------------------	------

Stack and Misc. Thermal Losses (MW)	295.0
-------------------------------------	-------

Cooling Towers (MW)	721.6
---------------------	-------

Table 3.12

Comparison of Cost Accounts⁺ for MHD/Gas Turbine
Base Case and MHD/Steam^{*} Plants

<u>MHD/Steam</u>			<u>Base Case I MHD/Gas Turbine</u>		
Civil and Structures	-	\$ 40.2 x 10 ⁶	Civil and Structures	-	\$ 36.2 x 10 ⁶
Boiler Plant Equipment	-	165.3	Coal & Flue Gas Handling; Instrumentation (131.7) Gas Turbine Plant Heat Exchangers (100.1)	-	231.8
Turbogenerator Units	-	52.8	Gas Turbine/Compressor/ Generator Sets	-	19.6
Accessory Electric	-	32.6	Accessory Electric	-	28.7
Miscellaneous Power Plant	-	12.3	Miscellaneous Power Plant	-	11.7
MHD Topping Cycle Equipment	-	343.7	MHD Topping Cycle Equipment	-	285.6
Transmission Plant	-	<u>11.2</u>	Transmission Plant	-	<u>10.0</u>
		\$ 658.1 x 10 ⁶			\$ 623.6 x 10 ⁶

+ Costs include material plus direct and indirect labor.

* Includes minor adjustments to account for improved MHD channel design.

exchangers. This account is more costly than that for the MHD/steam system because of the high cost of the gas turbine plant heat exchangers. The other two accounts have a lower cost for the MHD/gas turbine system than for the MHD/steam system. The gas turbine/compressor/generator units cost less than the steam turbogenerator units both because of a smaller power output and a lower cost per unit power output. The MHD Topping Cycle Equipment account has a lower cost for the MHD/gas turbine system primarily because of less expensive MHD air preheat components. The effectiveness, and therefore cost, of the HTAH is lower due to a higher gas inlet temperature to the heater in the MHD/gas turbine system. The low temperature air heater in this system is the air-cooled radiant heater and has a much lower cost than the MHD gas/air, metal tube low temperature air heater in the MHD/steam system. The four other cost accounts in Table 3.12 show only small differences between the two systems, arising mainly because of the difference in net electrical power output. A detailed presentation of the equipment and labor costs of the MHD/steam and Base Case I systems appears in Appendix C.

While the Base Case I system has a thermal and economic performance which is lower than that of the MHD/steam system, the Case I vapor cycle variations have performances which approach and even exceed that of the steam-bottomed system. Variation 2 has a thermal efficiency of 43.6% and a COE of 43.84 mills/kWhr, which closely matches the MHD/steam figures of 44.7% and 43.85 mills/kWhr. Yet this variation has 330 MW of cooling tower heat rejection, which is less than half of the MHD/steam system 720 MW value. Both vapor cycle Variations 3 and 4 have thermal and economic performances which exceed the MHD/steam system performance while still having a lower amount of cooling tower heat rejection. A graphic comparison of the performance of the MHD/gas turbine and MHD/steam systems is shown in Fig. 3.5, in which the MHD/gas turbine results are shaded.

The equipment and labor costs used in the economic analysis of the MHD/gas turbine systems were based on the costs used in the EPRI/W study⁽³⁾ when possible and appropriate, with minor adjustments being made for differences in component size or performance. For components which were not the same for the two systems, an uncertainty was introduced to the cost comparison of the two systems which was much greater than that for the

Figure 3.5 Performance Comparison of MHD/Gas Turbine System with MHD/Steam System.

similar components. The estimated relative uncertainty in the COE comparison of the MHD/steam and MHD/gas turbine systems is 2 mills/kWhr, and is due mainly to the costing of the non-similar components. Therefore, the 3 mill/kWhr difference between the Base Case I COE and the MHD/steam COE is only 1 mill/kWhr greater than the relative COE uncertainty.

3.4 Case II System Results

Base Case II, along with its four vapor cycle bottoming plant variations, includes many features currently envisioned for a second generation MHD plant: a directly-fired HTAH with an air outlet temperature of 1800 K, a diffuser pressure recovery of 0.65, and sulfur control accomplished by means of seed regeneration. The differences between Case I and Case II in the system modeling and the economic rules are presented in Table 3.13. The calculation of a levelized COE, in which the fuel and operating costs are adjusted upwards to account for the inflation of these costs over a plant's lifetime, and the use of a plant availability factor of 0.65 are typical of recent MHD system studies. The specification of a fuel cost of \$1.35/MBtu was based on projected costs for Western coal⁽¹²⁾ in the time period during which second generation MHD plants are expected to operate. The non-levelized O & M cost for Base Case II, excluding the seed reprocessing O & M, is 2.7 mills/kWhr. The entire contribution to the COE of the seed recycle and regeneration system was calculated separately and was based on the cost and design of the Westinghouse process,⁽⁸⁾ selected on the basis of lowest annual cost. The Case II systems were designed to meet an SO₂ emission level of 0.2 lb/MBtu and an NO_x emission level of 0.5 lb/MBtu. The combustor stoichiometry of 0.935 enables the systems to meet this NO_x standard.

The system state points for Base Case II are presented in Figure 3.6, and the state points for the vapor cycle Variation 1 are presented in Figure 3.7. The principal mass flows for Case II are summarized in Table 3.14. As in Case I, the largest mass flow is the turbine cycle A air, though it is smaller for Case II than for Case I while the other mass flows are somewhat larger. This is a result of the higher air preheat temperature for Case II which has the effect of reducing the energy input to the high temperature turbine cycle. The other mass flows for Case II are larger because of the 9% increase in thermal input to the system.

Table 3.13

System Differences Between Case I and Case II

	<u>Case II</u>	<u>Case I</u>
Coal Type	Montana Rosebud	Illinois #6
Final % Moisture in Coal After Drying	5%	0%
Combustor Stoichiometry	0.935	0.95
Combustor Thermal Input	2140 MW	1970 MW
Seeding Level	0.7 wt % K	1.0 wt % K
Air Preheat Temperature	1800 K	1644 K
Diffuser Pressure Recovery	0.65	0.50
Sulfur Control Method	Seed regeneration	Flue gas scrubbing
COE Calculation Method	Levelized (levelization factor = 1.882)	First year commercial operation (non- levelized)
Coal Cost	\$ 1.35/MBtu	\$ 1.166/MBtu
Availability Factor	0.65	0.70
Contingency Factor for Unconventional Components	0.20	0.10
Factor for Escalation and Interest During Construction	1.679	1.699

State Point	T(K)	P(atm)	M(kg/s)	State Point	T(K)	P(atm)	M(kg/s)	State Point	T(K)	P(atm)	M(kg/s)
<u>MHD Air/Combustion Gas</u>				14	1516	1.00	813.7	26	779	1.02	79.7
1	298	1.00	647.7	15	1306	0.98	813.7	27	779	1.02	957.8
2	462	7.87	647.7	16	896	0.96	813.7	28	558	1.01	957.8
3	549	7.81	647.7	17	662	0.94	813.7	29	499	1.00	957.8
4	1000	7.36	647.7	18	662	0.94	351.7	<u>Turbine Cycle "B" Air</u>			
5	1800	7.14	647.7	19	662	0.94	462.0	30	298	1.00	879.7
6	1800	7.14	346.4	<u>Turbine Cycle "A" Air</u>				31	440	6.20	879.7
7	1800	7.14	301.3	20	298	1.00	1037.5	32	571	6.12	879.7
8	2829	6.80	734.0	21	496	11.00	1037.5	33	805	5.86	879.7
9	2684	4.28	734.0	22	703	10.83	1037.5	34	928	5.74	879.7
10	2209	0.79	734.0	23	1060	10.46	1037.5	35	622	1.01	879.7
11	2260	1.03	734.0	24	862	4.18	1037.5	36	492	1.00	879.7
12	2000	1.03	734.0	25	1060	4.02	1037.5				
13	1389	1.00	734.0								

Figure 3.6 State Point Diagram for Base Case II

State Point	T(K)	P(atm)	M(kg/s)	State Point	T(K)	P(atm)	M(kg/s)	State Point	T(K)	P(atm)	M(kg/s)
<u>MHD Air/Combustion Gas</u>				19	400	0.92	293.4	33	607	6.12	883.5
1	298	1.00	647.7	20	380	0.92	520.3	34	829	5.86	883.5
2	560	7.70	647.7	21	398	1.00	813.7	35	966	5.72	883.5
3	1000	7.25	647.7	<u>Turbine Cycle "A" Air</u>				36	650	1.01	883.5
4	1800	7.03	647.7	22	298	1.00	1009.5	37	481	1.00	883.5
5	1800	7.03	346.4	23	492	11.00	1009.5	<u>Vapor Cycle Ammonia</u>			
6	1800	7.03	301.3	24	720	10.83	1009.5	38	304	12.5	170.0
7	2828	6.70	734.0	25	1060	10.46	1009.5	39	313	201.4	170.0
8	2683	4.22	734.0	26	862	4.18	1009.5	40	342	199.4	55.2
9	2214	0.79	734.0	27	1060	4.02	1009.5	41	342	199.4	114.8
10	2260	1.03	734.0	28	777	1.02	79.7	42	620	197.4	114.8
11	2000	1.03	734.0	29	777	1.02	929.8	43	620	197.4	55.2
12	1389	1.00	734.0	30	532	1.00	929.8	44	620	197.4	170.0
13	1516	1.00	813.7	<u>Turbine Cycle "B" Air</u>				45	375	14.0	170.0
14	1312	0.98	813.7	31	298	1.00	883.5	46	325	13.8	170.0
15	933	0.96	813.7	32	437	6.20	883.5				
16	711	0.94	813.7								
17	711	0.94	293.4								
18	711	0.94	520.3								

Figure 3.7 State Point Diagram for Case II, Variation 1

Table 3.14

Case II Summary of Principal Mass Flows

	<u>Base Case</u>	<u>Var. 1</u>	<u>Var. 2</u>	<u>Var. 3</u>	<u>Var. 4</u>
Coal Flow (kg/s)	103	103	103	103	103
MHD Compressor Air Flow (kg/s)	648	648	648	648	648
MHD Combustion Gas Flow (kg/s)	734	734	734	734	734
Secondary Combustion Air Flow (kg/s)	80	80	80	80	80
Turbine Cycle A Air Flow (kg/s)	1038	1010	962	1088	976
Turbine Cycle B Air Flow (kg/s)	880	884	888	884	887
Vapor Cycle Ammonia Flow (kg/s)	0	170	298	352	468
System Flow Schematic Fig. No.	2.1	2.2	2.3	2.4	2.5
(Page No.)	7	11	12	14	15

The MHD channel performance is summarized in Table 3.15. The channel length for the optimized vapor cycle variation systems is slightly shorter than the base case channel length. The channel enthalpy extraction for Case II is greater than that for Case I because of the increased pressure ratio for the channel, which is a result of the increased diffuser pressure recovery and higher channel inlet temperature. The MHD cycle efficiency shows an improvement of 11% over Case I because of the higher air preheat temperature and the greater channel enthalpy extraction.

Power generation and consumption for the Case II systems are presented in Table 3.16. As expected due to the improved MHD plant design, the net MHD plant power for Base Case II was increased by 12% over Base Case I, an amount greater than the thermal input increase of 9%. The power generation split between the MHD plant and the gas turbine plant was changed, with the MHD plant accounting for a slightly greater percentage of the total power generation in Base Case II (62%) than in Base Case I (59%). The auxiliary power requirement for Case II is less than in Case I because of the lower power requirement for sulfur emission control.

The gas turbine plant heat exchanger design parameters for Base Case II and Variation 1 are given in Tables 3.17 and 3.18 respectively. The design parameters are generally similar to those of Case I, with some minor differences being that the intercooler effectivenesses and the turbine cycle B recuperator effectivenesses are slightly higher for Case II. This small upward shift can be attributed to the fact that the system thermal performance is given greater weight in Case II than in Case I due to the levelization of the fuel cost in calculating the COE, which nearly doubles its value.

Energy flow diagrams for Base Case II and Case II, Variation 1 are shown in Figures 3.8 and 3.9. A tabular summary of the energy balances for the Case II systems is given in Table 3.19. Because of the greater energy requirement for the drying of Montana Rosebud coal, the MHD gas stack loss for Base Case II is slightly less than for Base Case I, even though the MHD gas flow rate is greater. The increased coal drying requirement also accounts for the fact that the maximum vapor cycle heat rejection to the cooling towers is less for Case II than for Case I.

Table 3.15

Case II MHD Channel Parameters

	<u>Base Case</u>	<u>Vapor Cycle Variations</u>
Combustor Coal Thermal Input	2140 MW	2140 MW
Combustor Pressure	6.8 atm	6.7 atm
Channel Inlet Velocity	844 m/s	844 m/s
Channel Outlet Velocity	694 m/s	695 m/s
Maximum Axial Field Strength	3.2 kV/m	3.2 kV/m
Channel Length	16.7 m	16.1 m
Gross Power Output	744.3 MW _{DC}	739.1 MW _{DC}
Channel Heat Loss	86.3 MW	84.2 MW
Channel Enthalpy Extraction	23.5 %	23.3 %
MHD Cycle Efficiency ($\frac{P_{MHD,AC} - P_{Comp}}{q_{in}}$)	27.2 %	26.4 %
MHD Channel Isentropic Efficiency	79.0 %	79.0 %

Table 3.16

Case II Summary of Power Generation and Consumption

	<u>Base Case</u>	<u>Var. 1</u>	<u>Var. 2</u>	<u>Var. 3</u>	<u>Var. 4</u>
<u>Power Generation (MW_{AC})</u>					
MHD Channel Power	733.1	728.1	728.1	728.1	728.1
Net MHD Plant Power ($P_{\text{MHD}} - P_{\text{Comp}}$)	570.9	553.0	553.0	553.0	553.0
Turbine Cycle A Power	249.3	245.5	234.0	218.7	200.7
Turbine Cycle B Power	<u>103.6</u>	<u>116.7</u>	<u>102.6</u>	<u>108.8</u>	<u>106.4</u>
Total Gas Turbine Plant Power	352.9	362.2	336.6	327.5	307.1
Ammonia Plant Power	0.0	73.4	128.5	151.8	201.7
<u>Power Consumption (MW_{AC})</u>					
MHD Compressor Power	162.2	175.1	175.1	175.1	175.1
Auxiliary Power	30.9	33.6	34.2	34.7	35.2
Net Power Generation (MW _{AC})	892.9	955.0	983.9	997.6	1026.6

Table 3.17

Base Case II Gas Turbine Plant Heat Exchanger Design Parameters

	<u>HXA1</u>	<u>HXA2</u>	<u>HXA3</u>	<u>HXB1</u>	<u>HXB2</u>	<u>HXM1</u>	<u>ICA</u>	<u>. ICB</u>	<u>ICM</u>
Effectiveness	0.78	0.68	0.32	0.72	0.72	0.90	0.70	0.70	0.70
Heat Load (MW)	225	414	233	119	221	59	102	63	54
Capacity Ratio	0.94	0.87	0.94	0.99	1.00	0.68	1.00	1.00	1.00
$\Delta P/P$, Shell Side	.015	.020	.020	.013	.020	.008	.010	.010	.010
$\Delta P/P$, Tube Side	.015	.035	.040	.013	.043	.008	.015	.015	.015

Table 3.18

Case II, Variation 1 Gas Turbine Plant Heat Exchanger Design Parameters

	<u>HXA1</u>	<u>HXA2</u>	<u>HXA3</u>	<u>HXB1</u>	<u>HXB2</u>	<u>ICA</u>	<u>ICB</u>
Effectiveness	0.86	0.64	0.31	0.80	0.68	0.72	0.72
Heat Load (MW)	243	383	226	156	211	102	65
Capacity Ratio	0.93	0.90	0.97	0.99	1.00	1.00	1.00
$\Delta P/P$, Shell Side	.015	.020	.020	.013	.020	.010	.010
$\Delta P/P$, Tube Side	.015	.035	.040	.013	.043	.015	.015

Figure 3.8. Base Case II Energy Flow Diagram

Figure 3.9 Energy Flow Diagram for Case II, Variation 1

Table 3.19

Case II Energy Balance Summary

<u>Energy In (MW)</u>	<u>Base Case</u>	<u>Var. 1</u>	<u>Var. 2</u>	<u>Var. 3</u>	<u>Var. 4</u>
Coal to Combustor	2140	2140	2140	2140	2140
<u>Energy Out (MW)</u>					
Electrical Power	892.9	955.0	983.9	997.6	1026.6
Auxiliary Power (less ID fan power)	22.9	24.4	24.9	25.4	25.9
Conversion Losses	14.1	15.0	15.5	15.7	16.2
Coal Dryer + Misc. Heat Losses	140.3	140.3	140.3	140.3	140.3
Intercooler Energy Rejection	218.6	167.0	97.1	62.1	0.0
Turbine Cycle A Exhaust	195.6	221.6	210.2	68.0	62.6
Turbine Cycle B Exhaust	173.2	164.0	62.0	161.7	62.0
MHD Exhaust	306.9	168.1	170.9	170.5	171.4
(Total Exhaust Stack Losses, A + B + MHD)	(675.7)	(553.7)	(443.1)	(400.2)	(286.7)
Cooling Towers	0.0	200.4	351.0	414.5	550.8
Generator (+ Combustor and Nozzle) Energy Rejection	175.5	84.2	84.2	84.2	84.2

The thermal performance of the Case II systems is presented in Table 3.20. In this table both a station thermal efficiency and an overall thermal efficiency are presented. The difference between the two is that the system energy input used in the calculation of the station efficiency is only the combustor coal input, which strongly affects the overall plant size. The station efficiency does not consider the 90 MW_t coal input required by the seed reprocessing plant, which does not affect the size of any other system components. (For Case I, the station thermal efficiency is the same as the overall thermal efficiency since the only coal input is that to the combustor). The station efficiency is a truer indicator of the performance of the thermodynamic cycles, and can have a stronger effect on the COE, while the overall efficiency is a truer indicator of the system's efficiency in utilizing an energy resource. The station thermal efficiency of Base Case II is 41.7%, an increase of 2.0 percentage points from Base Case I. This increase is partly due to the improved MHD plant performance, and partly due to the use of low sulfur Montana Rosebud coal and seed regeneration. The overall thermal efficiency is lower than the station efficiency by 1.7 percentage points. The vapor cycle variation 4, which has the greatest cooling water requirements, has a station thermal efficiency of 48.0% and an overall thermal efficiency of 46.0%. Thus in comparison to Case I, the Case II systems have station thermal efficiencies which are about 2 percentage points higher, but overall efficiencies which are nearly the same. This paradox is a result of the different quantity and type of energy requirement for sulfur control for the two cases. Seed regeneration requires more energy than does flue gas scrubbing because of its need for a separate coal input, though it has a substantially smaller electrical energy requirement than does flue gas scrubbing. Thus for an overall thermal efficiency, the effect of the improved performance of the MHD plant in Base Case II is offset by the increased energy requirement for sulfur control, resulting in very similar overall efficiencies for Case I and Case II. For a station efficiency, only the electrical requirement for sulfur control is considered. Thus the station efficiencies for Case II are greater than for Case I because of both the improved MHD plant performance and the lower auxiliary power requirement for sulfur control.

Table 3.20

Case II System Thermal Performance

	<u>Base Case</u>	<u>Var. 1</u>	<u>Var. 2</u>	<u>Var. 3</u>	<u>Var. 4</u>
Net Electrical Power Output (MW)	892.9	955.0	983.9	997.6	1026.6
Cooling Tower Heat Rejection (MW)	0.0	200	351	414	551
Overall Thermal Efficiency (%)	40.0	42.8	44.1	44.7	46.0
Station Thermal Efficiency* (%)	41.7	44.6	46.0	46.6	48.0
Station Heat Rate* (Btu/kWhr)	8180	7648	7423	7321	7115

* does not include 90 MW_t coal input for seed reprocessing.

The economic performance of the Case II systems is presented in Table 3.21. With the exception of those changes listed in the bottom part of Table 3.13, the economic rules used in the COE calculation are the same as those in Table 3.9. The levelized COE of Base Case II is 58.34 mills/kWhr, while vapor cycle Variation 4 has a COE which is 5.5 mills/kWhr less. The total constructed cost for the Case II systems is roughly $\$ 50 \times 10^6$ greater than that for Case I. The increase in the HTAH cost and in the contingency factor for unconventional components were two major contributors to the total constructed cost increase. Since the increase in electrical power output is proportionally greater than the cost increase however, the unit capital cost for Case II systems is about 8% less than that for Case I. Included as a separate item in the breakdown of the COE for Case II is the contribution of seed reprocessing. This item, which includes both the capital and operating cost of the seed reprocessing plant, represents about 4% of the total COE for Case II. The levelized fuel cost portion of the Base Case II COE is 36% of the total, which is in contrast to the 21% portion which the non-levelized fuel cost represents in Base Case I. Also in contrast to Case I is the fact that the decrease in COE_{fuel} is slightly greater than the decrease in $COE_{capital}$ as the amount of cooling tower heat rejection increases from 0 to 559 MW. In Case I, the decrease in $COE_{capital}$ was twice the decrease in COE_{fuel} . Case II, however, does show the same approximately linear relationship between COE and thermal efficiency and the amount of cooling tower heat rejection, as shown in Figure 3.10.

If the COE for Base Case II were calculated using all of the same rules as in Base Case I and using the same O & M cost, then the COE would be 42.0 mills/kWhr. This represents a decrease in the COE from Base Case I of 4.9 mills/kWhr. About 40%, or 2.1 mills/kWhr is due to the improved performance of the MHD plant while the remaining 2.8 mills/kWhr of the decrease is due to the lower cost of sulfur emissions control with Montana Rosebud coal and seed regeneration.

3.5 Case III System Results

Base Case III and its one vapor cycle variation are MHD/gas turbine systems which use oxygen enrichment instead of high temperature air preheat to obtain a high plasma temperature in the coal combustor. The

Table 3.21

Case II System Economic Performance

	<u>Base Case</u>	<u>Var. 1</u>	<u>Var. 2</u>	<u>Var. 3</u>	<u>Var. 4</u>
Total Constructed Cost (mid-1978 \$)	898×10^6	927×10^6	929×10^6	944×10^6	945×10^6
Unit Capital Cost (\$/kW)	1006	970	944	946	921
COE _{capital} (mills/kWhr)	29.90	28.90	28.11	28.22	27.46
COE _{fuel}	20.78	19.44	18.87	18.61	18.08
COE _{seed reprocessing}	2.58	2.41	2.34	2.31	2.24
COE _{O & M}	5.08	5.08	5.08	5.08	5.08
COE _{total}	58.34	55.83	54.40	54.22	52.86
Cooling Tower Heat Rejection (MW)	0	200	351	414	551

* levelized by factor of 1.882

Figure 3.10 Variation of Case II Performance with Cooling Tower Heat Rejection.

State Point	T(K)	P(atm)	M(kg/s)	State Point	T(K)	P(atm)	M(kg/s)	State Point	T(K)	P(atm)	M(kg/s)
<u>MHD Oxidizer/Combustion Gas</u>				13	896	0.99	477.3	24	811	1.02	79.7
1	298	1.00	395.4	14	1136	0.99	557.0	25	811	1.02	959.8
2	452	7.64	395.4	15	678	0.97	557.0	26	540	1.01	959.8
3	533	7.58	395.4	16	678	0.97	325.7	27	507	1.00	959.8
4	980	7.35	395.4	17	678	0.97	231.3	<u>Turbine Cycle "B" Air</u>			
5	980	7.35	211.4	<u>Turbine Cycle "A" Air</u>				28	298	1.00	1203.9
6	980	7.35	184.0	18	298	1.00	1039.5	29	483	10.40	1203.9
7	2868	7.00	477.3	19	488	11.00	1039.5	30	628	10.30	1203.9
8	2743	4.43	477.3	20	742	10.89	1039.5	31	865	9.99	1203.9
9	2313	0.81	477.3	21	1060	10.56	1039.5	32	1096	9.37	1203.9
10	2240	1.03	477.3	22	824	3.46	1039.5	33	665	1.01	1203.9
11	1852	1.03	477.3	23	1060	3.34	1039.5	34	521	1.00	1203.9
12	1440	1.01	477.3								

Figure 3.11 State Point Diagram for Base Case III

method of oxygen enrichment used in Case III, which is similar to the method used in the PSPEC studies,^{(4) (5)} consists of using an air separation facility to produce an atmospheric pressure gas stream which is 70% O₂. This oxygen-enriched stream is mixed with an ambient air stream to produce an oxidizer containing 35% O₂ by volume. This oxidizer is then compressed, preheated to 980 K, and then injected into the two-stage coal combustor. With the exception of the changes required for the use of oxygen enrichment, the Case III systems have the same component specifications, parameters, and economic rules as the Case II systems.

The system state points for Base Case III are presented in Figure 3.11, and the state points for Variation 1 are presented in Figure 3.12. The principal mass flows for Case III are summarized in Table 3.22. Two major differences from Case II in the mass flows can be seen, one being that the MHD oxidizer and combustion gas flow rates are reduced by more than one third, and the other being that the turbine cycle B air flow rate is nearly 50% greater. The reduced MHD oxidizer and gas flow rates are a direct result of the oxygen enrichment. The increased turbine cycle B flow is a result of an increased energy input to the cycle. In the energy flow diagram for Base Case III, Figure 3.13, it can be seen that the increase is due primarily to a large energy flow to the cycle from the diffuser and radiant heater, and secondarily due to energy input to the MHD gas stream just upstream of turbine plant B from the secondary combustion air.

The MHD channel performance is summarized in Table 3.23. The gross power output of the channel is 497 MW, which is significantly less than the 744 MW channel output for Base Case II. Since the enthalpy extraction of the channel is only 10% lower than that in Case II, it follows that the reduced channel power is primarily caused by a reduced energy input to the channel. In the energy flow diagram it can be seen that the energy flow to the combustor from the radiant heater and turbine plant A is only 286 MW, compared to the 1117 MW for Case II. This reduced energy inflow to the combustor, and therefore the channel, is a result of both the lower level of oxidizer preheat and the lower oxidizer mass flow rate. The MHD cycle efficiency for Case III also is much lower than in Case II, 15% compared to 27%. This is due not only to the reduced amount of MHD exhaust energy regeneration caused by eliminating the HTAH, but also by the 67 MW power requirement of the air separation facility.

State Point	T(K)	P(atm)	M(kg/s)	State Point	T(K)	P(atm)	M(kg/s)	State Point	T(K)	P(atm)	M(kg/s)
<u>MHD Oxidizer/Combustion Gas</u>				<u>Turbine Cycle "A" Air</u>				<u>Vapor Cycle Ammonia</u>			
1	298	1.00	395.4	20	298	1.00	1064.6	37	304	12.5	298.0
2	557	7.59	395.4	21	623	11.00	1064.6	38	313	201.4	298.0
3	980	7.35	395.4	22	767	10.89	1064.6	39	342	199.4	181.1
4	980	7.35	211.4	23	1060	10.56	1064.6	40	342	199.4	55.1
5	980	7.35	184.0	24	831	3.58	1064.6	41	342	199.4	61.8
6	2868	7.00	477.3	25	1060	3.46	1064.6	42	620	197.4	298.0
7	2743	4.43	477.3	26	807	1.03	79.7	43	375	14.0	298.0
8	2313	0.81	477.3	27	807	1.03	984.9	44	325	13.8	298.0
9	2240	1.03	477.3	28	653	1.02	984.9				
10	1852	1.03	477.3	29	365	1.00	984.9				
11	1441	1.01	477.3	<u>Turbine Cycle "B" Air</u>							
12	929	0.99	477.3	30	298	1.00	1233.4				
13	1165	0.99	557.0								
14	722	0.97	557.0								
15	722	0.97	279.9								
16	722	0.97	277.1								
17	400	0.95	279.9								

Figure 3.12 State Point Diagram for Case III, Variation 1

Table 3.22

Case III Summary of Principal Mass Flows

	<u>Base Case</u>	<u>Variation 1</u>
Coal Flow (kg/s)	103	103
MHD Compressor Oxidizer Flow (kg/s)	396	396
MHD Combustion Gas Flow (kg/s)	478	478
Secondary Combustion Air Flow (kg/s)	80	80
Turbine Cycle A Air Flow (kg/s)	1040	1065
Turbine Cycle B Air Flow (kg/s)	1204	1233
Vapor Cycle Ammonia Flow (kg/s)	0.0	298
System Flow Schematic Fig. No.	2.6	2.7
(Page No.)	18	19

Figure 3.13 Base Case III Energy Flow Diagram

Table 3.23

Case III MHD Channel Parameters

Coal Combustor Thermal Input	2140 MW
Combustor Pressure	7.0 atm
Channel Inlet Velocity	837 m/s
Channel Outlet Velocity	681 m/s
Maximum Axial Field Strength	2.2 kV/m
Channel Length	14.8 m
Gross Power Output	496.7 MW _{DC}
Channel Heat Loss	76.5 MW
Channel Enthalpy Extraction	21.3 %
MHD Cycle Efficiency	$\frac{P_{\text{MHD,AC}} - P_{\text{Comp}} - P_{\text{Air sep.}}}{q_{\text{in}}}$ 15.3 %
MHD Channel Isentropic Efficiency	78.9 %

The power generation and consumption for Case III are summarized in Table 3.24. In the comparison of this table with Table 3.16 for Case II, it is evident that the use of oxygen enrichment has shifted the majority of the power production from the MHD plant to the gas turbine plant. For Base Case III, the net MHD plant power is 328 MW, or one third less than the total gas turbine plant power of 487 MW. While the turbine cycle A power has not changed much, the power output of turbine cycle B is about double that in Case II. The ammonia plant for the vapor cycle Variation 1 is the same size as that of Case II's Variation 2.

The gas turbine plant heat exchanger design parameters for Base Case III and Variation 1 are given in Tables 3.25 and 3.26 respectively. All of the heat exchangers have a higher effectiveness than in Case II, with significant increases in particular for the three MHD gas/air recuperators, HXA2, HXA3, and HXB2. Since the capacity ratios for these three heat exchangers are much lower than in Case II, the increases in size and cost of these heat exchangers are not as substantial as they would be if the capacity ratios were the same.

A tabular summary of the energy balances for the two Case III systems is presented in Table 3.27. Comparison with Table 3.19 for Case II shows that the total exhaust stack losses are not much different, but that the modest increase in intercooler energy rejection and the addition of the air separation plant power requirement for Case III comprise the main differences.

The thermal performance for Case III is presented in Table 3.28 and the economic performance in Table 3.29. The Base Case III power output of 789 MW and the station thermal efficiency of 36.9% represents a substantial decrease in thermal performance from Base Case II's 893 MW and 41.7%. The economic performance of Base Case III, however, shows only a modest decrease from that of Base Case II. Its COE of 60.8 mills/kWhr is 2.5 mills/kWhr, or 4%, greater than that for Base Case II. The $\$109 \times 10^6$ lower total constructed cost of the Base Case III system accounts for the relatively smaller change in COE compared to that in thermal efficiency. It is also for this same reason that COE_{capital} for Base Case III is smaller than that for Base Case II, while the COE_{fuel} is nearly 3 mills/kWhr larger. The addition of the vapor cycle bottoming plant in Case III produces improvements in thermal efficiency and COE which are almost identical to Case II.

Table 3.24

Case III Summary of Power Generation and Consumption

	<u>Base Case</u>	<u>Variation 1</u>
<u>Power Generation</u> (MW_{AC})		
MHD Channel Power	489.2	489.2
Net MHD Plant Power ($P_{MHD} - P_{Comp} - P_{Air Sep.}$)	327.6	318.5
Turbine Cycle A Power	257.9	222.6
Turbine Cycle B Power	<u>228.6</u>	<u>235.2</u>
Total Gas Turbine Plant Power	486.5	457.8
Ammonia Plant Power	0.0	128.5
<u>Power Consumption</u> (MW_{AC})		
MHD Compressor	94.6	103.7
Air Separation Plant	67.0	67.0
Auxiliary Power	24.7	27.8
<u>Net Power Generation</u> (MW_{AC})	789.4	877.0

Table 3.25

Base Case III Gas Turbine Plant Heat Exchanger Design Parameters

	<u>HXA1</u>	<u>HXA2</u>	<u>HXA3</u>	<u>HXB1</u>	<u>HXB2</u>	<u>HXM1</u>	<u>ICA</u>	<u>ICB</u>	<u>ICM</u>
Effectiveness	0.84	0.78	0.40	0.80	0.90	0.92	0.74	0.74	0.74
Heat Load (MW)	278	371	277	182	311	33	108	121	33
Capacity Ratio	0.94	0.59	0.57	0.99	0.52	0.41	1.00	1.00	1.00
$\Delta P/P$, Shell Side	.010	.020	.020	.010	.020	.008	.008	.008	.008
$\Delta P/P$, Tube Side	.010	.030	.035	.010	.030	.008	.015	.015	.015

Table 3.26

Case III, Variation 1 Gas Turbine Plant Heat Exchanger Design Parameters

	<u>HXA1</u>	<u>HXA2</u>	<u>HXA3</u>	<u>HXB1</u>	<u>HXB2</u>	<u>ICB</u>
Effectiveness	0.84	0.76	0.40	0.80	0.86	0.72
Heat Load (MW)	164	350	276	228	304	112
Capacity Ratio	0.93	0.57	0.56	0.99	0.51	1.00
$\Delta P/P$, Shell Side	.010	.020	.020	.010	.020	.008
$\Delta P/P$, Tube Side	.010	.030	.035	.010	.030	.015

Table 3.27

Case III Energy Balance Summary

<u>Energy In (MW)</u>	<u>Base Case</u>	<u>Variation 1</u>
Combustor Coal Energy In	2140	2140
<u>Energy Out (MW)</u>		
Electrical Power	789.4	877.0
Auxiliary Power (less ID fan power)	21.3	23.8
Air Separation Plant Power	67.0	67.0
Conversion Losses	13.5	15.0
Coal Dryer + Misc. Heat Losses	127.3	127.3
Intercooler Energy Rejection	262.8	112.4
Turbine Cycle A Exhaust	203.6	66.9
Turbine Cycle B Exhaust	272.7	277.5
MHD Exhaust	216.9	133.1
(Total Exhaust Stack Losses: A + B + MHD)	(693.2)	(145.6)
Cooling Towers	0.0	351.0
Generator (+ Combustor and Nozzle) Energy Rejection	165.5	76.5

Table 3.28

Case III System Thermal Performance

	<u>Base Case</u>	<u>Variation 1</u>
Net Electrical Power Output (MW)	789.4	877.0
Cooling Tower Heat Rejection (MW)	0	351
Overall Thermal Efficiency (%)	35.6	39.6
Station Thermal Efficiency* (%)	36.9	41.0
Station Heat Rate* (Btu/kWhr)	9252	8328

*does not include 90 MW_t coal input for seed reprocessing

Table 3.29

Case III System Economic Performance

	<u>Base Case</u>	<u>Variation 1</u>
Total Constructed Cost (Mid-1978 \$)	789. x 10 ⁶	831. x 10 ⁶
Unit Capital Cost (\$/kW)	1000	948
COE _{capital} (mills/kWhr)	29.79	28.17
COE _{fuel} *	23.52	21.17
COE _{seed reprocessing} *	2.41	2.17
COE _{O & M} *	5.08	5.08
COE _{total}	60.80	56.59
Cooling Tower Heat Rejection (MW)	0	351

* levelized by factor of 1.882

The results of Case III show that the use of oxygen enrichment to obtain greater reliability for early MHD plants produces a system which has nearly the same unit capital cost as a system with a directly-fired HTAH, but a substantially lower thermal efficiency. The increase in COE due to oxygen enrichment is a strong function of the cost of coal and the method of COE calculation (levelized or non-levelized). Since a high cost coal and a levelized COE are used in Case III, the increase in COE, though small, is greater than it would be if a non-levelized COE and low cost of coal were used.

3.6 Sensitivity of System Performance to Key Variables

In this section the response of the system performance to the variation of important system variables is presented. For each of the eight variables investigated, the dependence of the cost of electricity and station thermal efficiency of the Base Case II system is given. First however, the relationship between the cost of electricity and thermal efficiency will be presented so that the results of this section can be more easily understood.

A great aid to the understanding of the dynamics of the system economic performance can be obtained by investigating the relationship between the plant capital cost and the thermal efficiency in the determination of the COE. The total COE can be expressed as the sum of its constituent parts:

$$COE_{total} = COE_{capital} + COE_{fuel} + COE_{O \& M} + COE_{seed}$$

To a close approximation, the contribution from O & M and seed reprocessing can be assumed to be constant, and the following expressions can be used for the capital and fuel contributions:

$$COE_{capital} = \frac{C_1 \cdot CCAP}{\eta} ; COE_{fuel} = \frac{C_2}{\eta}$$

where CCAP is the total capital cost of the plant and η is the station thermal efficiency. For the COE to decrease with increasing thermal efficiency, then

$$dCOE/d\eta = \frac{1}{\eta} \left[C_1 \frac{dCCAP}{d\eta} - COE_{\text{capital}} - COE_{\text{fuel}} \right] < 0.$$

This can be manipulated to give,

$$\frac{dCCAP/CCAP}{d\eta/\eta} < 1 + \frac{COE_{\text{fuel}}}{COE_{\text{capital}}}$$

Therefore, the COE will continue to decrease with an increasing thermal efficiency as long as the relative increase in the total capital plant cost is less than $1 + COE_{\text{fuel}}/COE_{\text{capital}}$ times the relative increase in η . The ratio of COE_{fuel} to COE_{capital} has a value of 0.3 for the Case I systems and a value of 0.7 for the Case II systems. This higher value for the Case II systems indicates that the thermal performance of the system is given greater weight in the Case II optimization, based on minimizing the COE, than in Case I. In the remainder of this section, the interaction of the total plant cost and the thermal efficiency will be brought out in the response of the system to changes in important variables.

Turbine Cycle A Turbine Inlet Temperature

The response of the system performance to the turbine inlet temperature of turbine cycle A is shown in Fig. 3.14. The data points are given for actual case runs, and approximate correlation bands are also shown which were obtained by a polynomial curvefit of the data. The purpose of the correlation bands is only to show trends in the behavior of COE and thermal efficiency; the band width was arbitrarily chosen to be ± 0.1 data units and does not represent a margin of uncertainty or error.

The thermal efficiency is seen to be a strong function of the turbine inlet temperature, and this is due to the strong effect which this variable has on the gas turbine plant efficiency. As the turbine inlet temperature

Figure 3.14 Variation of System Performance with Turbine Cycle A Turbine Inlet Temperature.

is reduced from 1100 K to 950 K, the gas turbine plant efficiency is reduced from 26.5% to 23.3%.

The COE is also substantially influenced by the turbine inlet temperature. For lower values of this variable the COE increases as the turbine inlet temperature is lowered because the reduction in plant cost, due mainly to the reduction in cost of the heat exchangers HXA2 and HXA3, is not greater than the reduction in thermal efficiency by the factor $1 + \text{COE}_{\text{fuel}}/\text{COE}_{\text{capital}}$, or roughly 1.7. The COE increases at high values of the turbine inlet temperature because the total plant cost increases at a rate greater than 1.7 times the increase in thermal efficiency as the turbine inlet temperature is raised. The data points show that there is little increase in COE from 1040 K to 1000 K. This is due to a reduction in tube wall thickness for HXA2 and HXA3 which causes a significant decrease in the cost of these heat exchangers. A similar cost reduction does not occur between 1000 K and 950 K. This type of non-smooth behavior will be seen for the other gas turbine plant variables also, and is caused by the fact that the tube wall thickness of the heat exchangers is limited to standard sizes and therefore cannot be smoothly varied.

Turbine Cycle A Compressor Pressure Ratio

The response of the system performance to the compressor pressure ratio of turbine cycle A is shown in Fig. 3.15. Compared to the turbine inlet temperature, the compressor pressure ratio has a much weaker effect on thermal efficiency, but a nearly comparable effect on COE. The maximum thermal efficiency occurs around a pressure ratio of 13, while the minimum COE occurs at a value of 11. As the pressure ratio decreases, the size of the heat exchangers increases due to a drop in both the mass flow per unit area and the convective heat transfer coefficient. This effect tends to outweigh the effect of a decreasing tube wall thickness and results in an increasing plant cost with decreasing pressure ratio at values lower than 11. This increasing plant cost combined with a decreasing thermal efficiency in the same region produces a moderate increase in COE as the pressure ratio falls below 11. At pressure ratios greater than 11, the increasing tube wall thickness dominates the decreasing

Figure 3.15 Variation of System Performance with Turbine Cycle A Compressor Pressure Ratio.

heat exchanger size in the effect on total plant costs, causing the plant costs to increase as the pressure ratio is increased above a value of 11. This increase in plant costs combined with an efficiency which decreases slightly beyond a pressure ratio of 13 results in the increased COE at high pressure ratios.

Turbine Cycle B Compressor Pressure Ratio

The response of the system performance to the compressor pressure ratio of turbine cycle B is shown in Fig. 3.16. Compared to the turbine cycle A compressor pressure ratio, this variable has a slightly greater effect on the thermal efficiency and a slightly lower effect on the COE. The COE rises moderately for a decreasing pressure ratio at low values for the same reason as for turbine cycle A - a result of both an increasing cost of heat exchangers and a decreasing thermal efficiency. An increase in tube wall thickness at a pressure ratio of 6.4 is the cause of a jump in COE at that value. The slow rise in COE for pressure ratios greater than this value occurs because, even though the total plant cost is decreasing, the plant cost does not decrease at a rate in excess of 1.7 times the rate of decrease in thermal efficiency.

Recuperator HXA2 Effectiveness

The effectiveness of recuperator HXA2 affects the system performance and state points in a way which is different from the other heat exchanger effectivenesses. Unlike most other heat exchangers, the recuperator HXA2 has an air outlet temperature which is specified independently (since the outlet temperature is the same as the cycle A turbine inlet temperature). The HXA2 effectiveness instead determines the cycle A air mass flow rate, in addition to the gas outlet temperature. This effectiveness therefore determines the energy input to turbine cycle A and also influences the energy input to turbine cycle B.

The response of the system performance to the HXA2 effectiveness is shown in Fig. 3.17. The effectiveness has a moderate influence on the thermal efficiency, displaying a nearly linear relationship. The thermal efficiency increases with increasing HXA2 effectiveness because of an increasing total energy input to the gas turbine cycles. The effect of the

Figure 3.16 Variation of System Performance with Turbine Cycle B Compressor Pressure Ratio.

Figure 3.17 Variation of System Performance with Recuperator HXA2 Effectiveness.

HXA2 effectiveness would be much greater if turbine cycle B were not present, since this cycle tends to moderate the influence on system performance of any change in the design of turbine cycle A.

The COE is seen to be only moderately affected by the HXA2 effectiveness over most of the range of values. For effectivenesses less than 0.78, the COE is nearly constant because the effect on COE of a change in thermal efficiency is nearly offset by a change in total plant cost. As the effectiveness rises beyond 0.78, the rapidly increasing heat exchanger cost is not offset by the increasing thermal efficiency and the COE therefore rises more quickly.

Recuperator HXB2 Effectiveness

The response of the system performance to the effectiveness of recuperator HXB2 is shown in Fig. 3.18. The effectiveness strongly affects the thermal efficiency, the reason being that the HXB2 effectiveness has a moderate influence on the energy input to turbine cycle B and a large influence on the efficiency of turbine cycle B. Increasing the effectiveness from 0.55 to 0.80 produces a 100 K increase in the turbine inlet temperature.

The COE is also seen to be strongly affected by the HXB2 effectiveness due to the fact that both the total plant cost and the thermal efficiency are strongly influenced by this variable. As the HXB2 effectiveness increases from the low value of 0.50, the plant cost exhibits a slow increase at first, with the increase becoming more rapid beyond an effectiveness of 0.65. Thus the substantial rise in thermal efficiency in the lower range of HXB2 effectiveness is not significantly offset in its effect on COE by the rise in the plant cost. But for HXB2 effectivenesses higher than 0.75, the effect of the increasing thermal efficiency is dominated by the effect of the rapid rise in the plant cost, resulting in a corresponding, rapid rise in COE.

Recuperator HXA1 Effectiveness

The response of the system performance to the effectiveness of recuperator HXA1, which regenerates the energy in the exhaust gas of turbine cycle A, is shown in Fig. 3.19. The small effect of this variable

Figure 3.18. Variation of System Performance with Recuperator HXB2 Effectiveness.

Figure 3.19 Variation of System Performance with Recuperator HX1 Effectiveness.

on thermal efficiency is due mainly to the fact that recuperator HX_{M1} and gas turbine cycle B act to reduce the influence which the HX_{A1} effectiveness has on the system performance. This small influence on thermal efficiency, combined with the relatively small capital cost of HX_{A1}, account for the relatively small effect which the HX_{A1} effectiveness has on the COE.

The response of the system to the effectiveness of the other air/air heat exchangers is not presented because they also have a small influence on the system performance. The other heat exchangers have an even smaller capital cost than that of HX_{A1}, and an influence on thermal efficiency which is roughly of the same magnitude as that of HX_{A1}.

MHD Channel Electrical Load Parameter

The response of the system performance to the electrical load parameter is shown in Fig. 3.20. While this variable strongly affects the channel length, as shown in the figure, it has only a weak influence on the thermal efficiency and the COE. Part of the reason for this weak influence is that this variable is actually a maximum value for the load parameter since the transverse field (E_y) constraint forces the load parameter to a lower value at the front end of the channel. For high values of the load parameter, the transverse field constraint is enforced over more than 75% of the channel. One other reason why the thermal efficiency is only weakly affected by the load parameter is that the gas turbine plant partially offsets any change in the MHD plant thermal performance.

The primary reason for the minor effect of the load parameter on the COE is that the rate of change plant costs is very nearly 1.7 times the rate of change in thermal efficiency as the load parameter is varied, so that their effects on the COE nearly balance.

MHD Combustor Pressure

The response of the system performance to the combustor pressure is shown in Fig. 3.21. As does the load parameter, this variable also strongly affects the MHD channel length and weakly affects the thermal efficiency and the COE. Increasing the combustor pressure from 6.2 atm

Figure 3.20 Variation of System Performance with Electrical Load Parameter.

Figure 3.21. Variation of System Performance with Combustor Pressure.

to 7.4 atm increases the enthalpy extraction of the channel by 9%. The thermal efficiency, however, is increased by less than 2%. This low increase in efficiency is due to the offsetting effect of the gas turbine plant and the fact that the MHD air compressor power requirement increases as the combustor pressure is raised.

The small variance in the COE over the range of combustion pressures included in Fig. 3.21 is due to the balancing changes in the total plant cost and the thermal efficiency, the same reason as for the electrical load parameter.

3.7 Analysis of System Irreversibilities

An analysis of the sources of irreversibility in the MHD/gas turbine system is a very useful tool for gaining further insight into the system and for guiding further investigations of the system. This analysis pinpoints the losses in the power generation system and measures their magnitudes so that proper, well informed decisions can be made on improving the performance of the system, whether it be by changing the system configuration or by improving component performance. In the following parts of this section, the method used in evaluating the irreversibilities will be delineated and the results of the irreversibility analysis for six MHD/gas turbine systems will be presented.

The energy measure of irreversibility of a process is a measure of the difference between the actual work input or output of a process and the amount of work input or output which would occur if the process were performed reversibly, which is the best possible performance allowed by the Second Law of Thermodynamics. For a process in which there is work output, the energy measure of irreversibility, Irrev_E , is equal to the difference between the maximum possible work output and the actual work output. For a process in which there is work input, Irrev_E is equal to the difference between the actual work input and the minimum possible work input. This definition of Irrev_E , however, actually encompasses two different ways of performing this measure.[†] One way, which is called the availability function measure of irreversibility, $\text{Irrev}_{E,Av}$, is a measure of the difference between the actual work input (or output) and

[†] Much of the material in this section is taken from reference 13.

the ideal work input (or output) obtained if the process were performed reversibly between the same initial and final state points as the actual process. The other way of performing this measure is called the ideal system measure of irreversibility, $\dot{I}_{rev_{E,IS}}$, and is particularly useful for processes which are ideally represented as being adiabatic and isentropic. This measure of irreversibility is equal to the difference between the actual work input (or output) and the ideal work input (or output) obtained if the process were adiabatic and isentropic. With this measure the final state point of the ideal process is different from that of the actual process. Components which perform processes that are often evaluated by the ideal system measure of irreversibility are pumps, compressors, turbines, motors and generators. The usual definition of the efficiency of these devices uses an adiabatic and isentropic process as the standard against which the real devices are compared.

The two energy measures of irreversibility, $\dot{I}_{rev_{E,Av}}$ and $\dot{I}_{rev_{E,IS}}$, can be related to each other by means of another type of irreversibility measure called the entropy measure of irreversibility, $\dot{I}_{rev_S}$. Using a rate basis, this measure is defined as

$$\dot{I}_{rev_S} \equiv \text{Creation of Entropy} - \left(\sum_{in} \dot{q}/T - \sum_{out} \dot{q}/T \right)$$

where

$\dot{q}$ = heat transfer per unit time

T = temperature at which energy is transferred into or out of system.

The relationship between $\dot{I}_{rev_S}$ and $\dot{I}_{rev_{E,Av}}$ is

$$\dot{I}_{rev_{E,Av}} = \dot{I}_{rev_S} T_0$$

where

T_0 = temperature of the environment.

The relationship between $\dot{I}_{rev_S}$ and $\dot{I}_{rev_{E,IS}}$ is

$$\dot{I}_{rev_{E,IS}} \approx \dot{I}_{rev_S} \left(\frac{T_f + T_{fs}}{2} \right)$$

where

$$\begin{aligned} T_f &= \text{temperature at final state point of actual process} \\ T_{fs} &= \text{temperature at final state point of ideal, isentropic process.} \end{aligned}$$

The two energy measures of irreversibility, therefore, are equal to the entropy measure of irreversibility multiplied by the appropriate temperature. This temperature can be considered to be a weighting factor such that,

$$\dot{\text{Irrev}}_E = T_{WF} \dot{\text{Irrev}}_S$$

where

$$\begin{aligned} \dot{\text{Irrev}}_E &= \dot{\text{Irrev}}_{E,AV} \quad \text{for } T_{WF} = T_0 \\ \dot{\text{Irrev}}_E &\approx \dot{\text{Irrev}}_{E,IS} \quad \text{for } T_{WF} = \left(\frac{T_f + T_{fs}}{2} \right) \end{aligned}$$

In the irreversibility analysis, $\dot{\text{Irrev}}_S$ and either $\dot{\text{Irrev}}_{E,AV}$ or $\dot{\text{Irrev}}_{E,IS}$ along with T_{WF} were calculated for each of the system components which was considered. The expressions which were used to calculate these measures of irreversibility are presented below.

Gas Turbine (turbine expansion process):

$$\dot{\text{Irrev}}_{E,IS} = \dot{W}_{k_{turb}} (1/\eta_{turb} - 1)$$

$$\dot{\text{Irrev}}_S = \dot{\text{Irrev}}_{E,IS}/T_{WF}$$

where

$$\dot{W}_{k_{turb}} = \text{actual rate of work output from turbine}$$

$$T_{WF} = \frac{T_f + T_{fs}}{2}$$

Compressor (compression process):

$$\dot{\text{Irrev}}_{E,IS} = \dot{W}_{k_{comp}} (1 - \eta_{comp})$$

$$\dot{\text{Irrev}}_S = \dot{\text{Irrev}}_{E,IS}/T_{WF}$$

where

$\dot{W}_{k, \text{comp}}$ = actual rate of work input to compressor

$$T_{WF} = \frac{T_f + T_{fs}}{2}$$

MHD Generator ("turbine" expansion process):

$$\dot{I}_{\text{rev}}_{E, IS} = \dot{W}_{k, \text{MHD}} (1/\eta_{\text{MHD, turb}} - 1)$$

$$\dot{I}_{\text{rev}}_S = \dot{I}_{\text{rev}}_{E, IS} / T_{WF}$$

where

$\dot{W}_{k, \text{MHD}}$ = actual DC power output from MHD generator

$\eta_{\text{MHD, turb}}$ = "turbine" efficiency of MHD generator

$$T_{WF} = \frac{T_f + T_{fs}}{2} \quad \text{where } T_f = \text{mean gas outlet temperature from MHD generator}$$

Heat Exchanger (heat transfer process):

$$\dot{I}_{\text{rev}}_S = q [1/T_{\ell m, c} - 1/T_{\ell m, h}]$$

$$\dot{I}_{\text{rev}}_{E, Av} = T_0 \dot{I}_{\text{rev}}_S$$

where

$$T_{\ell m} = \frac{T_{in} - T_{out}}{\ln(T_{in}/T_{out})}$$

c, h denote cold side and hot side

Stack (mixing of an exhaust stream with the atmosphere, for like gases with different temperatures; composition difference is not considered).

$$\dot{I}_{\text{rev}}_S = \dot{m} c_p [T_{\text{ex}}/T_0 - 1 - \ln(T_{\text{ex}}/T_0)]$$

$$\dot{I}_{\text{rev}}_{E, Av} = T_0 \dot{I}_{\text{rev}}_S$$

where

$$\begin{aligned} T_{ex} &= \text{temperature of exhaust stream} \\ \dot{m} &= \text{exhaust stream mass flow rate} \end{aligned}$$

Intercooler, Generator & Combustor (Base Case only) Coolers
(heat transfer plus mixing with atmosphere):

$$\begin{aligned} \dot{I}_{rev_S} &= q[1/T_0 - 1/T_{lm,H}] \\ \dot{I}_{rev_{E,Av}} &= T_0 \dot{I}_{rev_S} \end{aligned}$$

Diffuser (fluid dynamic process):

$$\begin{aligned} \dot{I}_{rev_S} &= \dot{m} [c_p \ln(T_{out,ad}/T_{in}) - R \ln(P_{out}/P_{in})] \\ \dot{I}_{rev_{E,IS}} &= T_{WF} \dot{I}_{rev_S} \end{aligned}$$

where

$$\begin{aligned} T_{out,ad} &= \text{gas exit temperature from the diffuser if the} \\ &\quad \text{diffuser were adiabatic.} \\ T_{in} &= \text{gas inlet temperature to the diffuser} \\ R &= \text{gas constant} \\ P_{in}, P_{out} &= \text{inlet and outlet diffuser pressures} \\ T_{WF} &= T_{WF,MHD} \text{ (same } T_{WF} \text{ as for MHD generator)} \end{aligned}$$

Air Separation Facility (separation of air into two mixtures of different compositions):

$$\begin{aligned} \dot{W}_{k_{min}} &= -\dot{m}_{O_2} R_{O_2} T \ln(P_{O_2 i}/P_{O_2 f}) - \dot{m}_{N_2} R_{N_2} T \ln(P_{N_2 i}/P_{N_2 f}) \\ \dot{I}_{rev_{E,IS}} &= \dot{W}_{k_{air \text{ sep.}}} - \dot{W}_{k_{min}} \\ \dot{I}_{rev_S} &= \dot{I}_{rev_{E,IS}}/T_{WF} \end{aligned}$$

where

$$\begin{aligned} \dot{W}_{k_{min}} &= \text{minimum rate of work required for air separation} \\ \dot{m}_{O_2}, \dot{m}_{N_2} &= \text{mass flow rate of oxygen and nitrogen in air} \end{aligned}$$

$$\begin{aligned}
 R_{O_2}, R_{N_2} &= \text{gas constants for oxygen and nitrogen} \\
 P_{O_2 i}, P_{N_2 i} &= \text{initial partial pressures of oxygen and nitrogen in air} \\
 P_{O_2 f}, P_{N_2 f} &= \text{final partial pressures of oxygen and nitrogen after separation} \\
 \dot{W}_{k \text{ air sep.}} &= \text{actual work input to air separation plant} \\
 T_{WF} &= T_0 \text{ (a convenient assumption)}
 \end{aligned}$$

The results of the analysis for the base case and one vapor cycle variation for Cases I, II, and III are shown in Tables 3.30, 3.31, and 3.32 respectively. This analysis did not attempt to point out every source of irreversibility, but chose to focus on the largest irreversibilities. The ideal system measure of irreversibility was used for compressors, turbines, the MHD generator, and the diffuser. The motivation for using this measure rather than the availability function measure was that it gave a better indication of the improvement in system performance which would result if the irreversibilities in these components could be eliminated. For an illustrative example of this point, consider the irreversibility of the turbine expansion process in the turbine in cycle B for Base Case I. The ideal system measure of irreversibility uses a temperature weighting factor of 580 K, resulting in an energy irreversibility of 28.2 MW. The availability function measure, on the other hand, would use $T_{WF} = T_0 = 298 \text{ K}$ and yield an energy irreversibility of 14.5 MW, or nearly half of that obtained by the ideal system measure. If the irreversibility in this turbine could be eliminated, giving a turbine efficiency of 100%, the actual improvement in system performance would be 25.7 MW if the mass flow rate and heat exchanger effectivenesses of turbine cycle B remained the same. Thus, the improvement in system performance which would occur by eliminating the irreversibility is much closer to $\dot{I}_{rev, IS}$ than to $\dot{I}_{rev, AV}$. The reason why the system improvement is somewhat less than $\dot{I}_{rev, IS}$ is that the turbine outlet

Table 3.30 Case I Component Irreversibilities

Component	Base Case			Variation 2		
	$\text{Irrev}_S (10^4 \text{ W/K})$	$T_{WF} \text{ (K)}$	$\text{Irrev}_E \text{ (MW)}$	$\text{Irrev}_S (10^4 \text{ W/K})$	$T_{WF} \text{ (K)}$	$\text{Irrev}_E \text{ (MW)}$
Stack, MHD gas	24.6	298	73.4	3.5	298	10.5
Stack, A turbine cycle	18.7	298	55.7	24.4	298	72.7
Stack, B turbine cycle	11.7	298	34.9	1.9	298	5.7
Compressor, MHD	4.2	425	18.0	3.5	534	18.9
Compressor, A turbine cycle	8.9	441	39.1	8.6	442	38.1
Compressor, B turbine cycle	5.0	410	20.4	4.2	500	21.2
High pressure turbine, A turbine	3.0	861	25.8	3.1	852	26.6
Reheat turbine, A turbine cycle	5.1	765	39.2	4.9	772	37.6
Turbine, B turbine cycle	4.9	580	28.2	4.7	611	28.7
MHD Generator	7.9	2149	168.7	7.6	2166	164.6
Intercooler, MHD	3.2	298	9.4	---	---	---
Intercooler, A turbine cycle	7.5	298	22.4	7.4	298	22.2
Intercooler, B turbine cycle	3.5	298	10.5	---	---	---
Recuperator HXA1	5.0	298	14.9	5.1	298	15.4
" HXA2	12.4	298	37.0	12.1	298	36.1
" HXA3	8.9	298	26.5	9.3	298	27.6
" HXB1	1.7	298	4.9	---	---	---
" HXB2	4.7	298	14.0	6.4	298	19.0
" HXM1	0.4	298	1.2	---	---	---
HTAH	9.3	298	27.6	9.3	298	27.7
Diffuser, Fluid Dynamic Process	2.8	2149	60.1	2.8	2166	61.5
Diffuser, Heat Transfer	5.3	298	15.9	6.9	298	20.4
Radiant Heater	27.6	298	82.2	26.7	298	79.5
Combustor Cooling	23.6	298	70.4	15.1	298	45.1
MHD Generator Cooling	22.1	298	65.8	20.3	298	60.5
Ammonia Condenser	---	---	---	3.6	298	10.7
Ammonia Heater, MHD gas	---	---	---	4.1	298	12.3
Ammonia Heater, B turbine exhaust	---	---	---	4.8	298	14.2

Table 3.31 Case II Component Irreversibilities

Component	Base Case			Variation 2		
	$\text{Irrev}_S (10^4 \text{ W/K})$	$T_{WF} (K)$	$\text{Irrev}_E (MW)$	$\text{Irrev}_S (10^4 \text{ W/K})$	$T_{WF} (K)$	$\text{Irrev}_E (MW)$
Stack, MHD gas	21.8	298	65.0	4.2	298	12.4
Stack, A turbine cycle	15.6	298	46.5	18.6	298	55.5
Stack, B turbine cycle	13.5	298	40.2	2.1	298	6.2
Compressor, MHD	4.5	430	19.5	3.9	544	21.0
Compressor, A turbine cycle	8.2	456	37.3	7.6	454	34.5
Compressor, B turbine cycle	5.5	412	22.7	4.8	513	24.8
High Pressure Turbine, A turbine cycle	3.0	851	25.9	2.8	851	24.0
Reheat Turbine, A turbine cycle	4.7	767	36.2	4.4	766	33.7
Turbine, B turbine cycle	5.3	610	32.5	5.5	633	34.6
MHD Generator	9.1	2173	197.9	9.0	2178	196.7
Intercooler, MHD	3.6	298	10.7	---	---	---
Intercooler, A turbine cycle	7.8	298	23.3	7.3	298	21.8
Intercooler, B turbine cycle	3.7	298	13.5	---	---	---
Recuperator HXA1	4.0	298	12.0	3.0	298	8.8
" HXA2	9.5	298	28.2	9.2	298	27.5
" HXA3	7.8	298	23.2	7.3	298	21.8
" HXB1	2.3	298	6.9	---	---	---
" HXB2	3.8	298	11.2	7.2	298	21.6
" HXM1	0.6	298	1.6	---	---	---
HTAH	8.7	298	25.8	8.7	298	25.8
Diffuser, Fluid Dynamic Process	1.9	2173	40.6	1.9	2178	42.1
Diffuser, Heat Transfer	8.4	298	24.9	8.4	298	24.9
Radiant Heater	27.9	298	83.2	27.9	298	83.2
Combustor Cooling	25.7	298	76.7	16.2	298	48.4
MHD Generator Cooling	25.4	298	75.8	24.8	298	73.9
Ammonia Condenser	---	---	---	3.8	298	11.4
Ammonia Heater, MHD exhaust	---	---	---	3.7	298	11.1
Ammonia Heater, B turbine exhaust	---	---	---	6.1	298	18.0

Table 3.32 Case III Component Irreversibilities

Component	Base Case			Variation 1		
	$\dot{I}rrev_S (10^4 W/K)$	$T_{WF} (K)$	$\dot{I}rrev_E (MW)$	$\dot{I}rrev_S (10^4 W/K)$	$T_{WF} (K)$	$\dot{I}rrev_E (MW)$
Stack, MHD gas	13.0	298	38.6	2.9	298	8.6
Stack, A turbine cycle	16.7	298	49.6	2.2	298	6.5
Stack, B turbine cycle	22.8	298	68.0	23.6	298	70.4
Compressor, MHD	2.7	425	11.4	2.3	541	12.4
Compressor, A turbine cycle	8.2	452	37.1	7.1	604	42.6
Compressor, B turbine cycle	8.9	441	39.4	9.0	440	39.6
High Pressure Turbine, A turbine	3.8	811	30.9	3.8	818	30.7
Reheat Turbine, A turbine cycle	4.0	801	32.1	4.3	793	33.8
Turbine, B turbine cycle	9.3	668	62.2	5.4	677	36.8
MHD Generator	5.8	2294	132.8	5.8	2294	132.8
Intercooler, MHD	2.1	298	6.4	---	---	---
Intercooler, A turbine cycle	8.0	298	24.0	---	---	---
Intercooler, B turbine cycle	8.0	298	24.0	7.8	298	23.3
Recuperator HXA1	4.2	298	12.5	1.1	298	3.4
" HXA2	7.4	298	22.0	8.6	298	25.7
" HXA3	12.7	298	37.8	12.5	298	37.2
" HXB1	3.0	298	8.9	3.3	298	9.9
" HXB2	1.9	298	5.5	3.1	298	9.1
" HXM1	4.1	298	12.1	---	---	---
HTAH	---	---	---	---	---	---
Diffuser, Fluid Dynamic Process	1.1	2294	24.6	1.1	2294	24.6
Diffuser, Heat Transfer	11.5	298	34.4	11.5	298	34.4
Radiant Heater	21.9	298	65.2	21.9	298	65.2
Combustor Cooling	25.8	298	76.8	16.3	298	48.5
MHD Generator Cooling	22.7	298	67.5	22.7	298	67.5
Ammonia Condenser	---	---	---	3.9	298	11.5
Ammonia Heater, MHD exhaust	---	---	---	4.1	298	12.1
Ammonia Heater, A turbine exhaust	---	---	---	6.5	298	19.3
Air Separation Facility	21.9	298	65.2	21.9	298	65.2

temperature is lower for $\eta_{\text{turb}} = 100\%$, resulting in a reduced amount of exhaust energy regeneration for the turbine cycle. The changes in turbine cycle B state points which occur as a result of the 100% turbine efficiency are shown in Fig. 3.22. In fact, it can be seen that if recuperator HXB1 were not present in the system and the turbine exhaust went directly to the stack, then the improvement in system performance would be equal to $\dot{\text{Irrev}}_{\text{E,IS}}$. The ideal system measure of irreversibility, therefore, is a better indicator of the potential improvement in system performance than the availability function measure for components whose processes can be ideally represented as being adiabatic and isentropic.

Observations which have been made on the basis of the results of the irreversibility analysis are presented below:

- The largest source of irreversibility in the system is the MHD generator, with $\dot{\text{Irrev}}_{\text{E,IS}} = 169 \text{ MW}$ for Base Case I. The losses in the MHD generator are primarily due to the presence of boundary layers and to Joulean heating. The use of hotter walls in the MHD generator would eliminate some of the losses, but a substantial amount of the losses are currently unavoidable. Reduction of the irreversibilities in the MHD generator is limited primarily by material and fundamental design constraints rather than by cost considerations.
- The cooling of the combustor and MHD generator represent large losses of availability, although the irreversibility of the generator cooling is actually already accounted for in $\dot{\text{Irrev}}_{\text{E,IS}}$ for the MHD generator. The reason for the high values of irreversibility is that heat is being transferred across a very large temperature difference and then dumped to the atmosphere. The exception to this is the combustor cooling for the vapor cycle variations in which the energy is transferred to supercritical ammonia for use in the vapor cycle plant. For these cases, $\dot{\text{Irrev}}_{\text{E}}$ for combustor cooling is roughly 25 MW less. The ways in which the irreversibility in cooling the combustor and generator could be reduced would be for the coolant to be at a much higher temperature, thus reducing the temperature difference, or for the combustor and generator to have hotter walls, thus reducing the total amount of heat transfer. Materials and design constraints are currently a problem for the implementation of either

State Point	T(K) $\eta_{\text{turb}}=90\%$	T(K) $\eta_{\text{turb}}=100\%$	P(atm)	M(kg/s)
30	298	298	1.00	805.7
31	439	439	6.00	805.7
32	544	518	5.94	805.7
33	798	790	5.73	805.7
34	880	872	5.68	805.7
35	589	552	1.01	805.7
36	485	473	1.00	805.7

Fig. 3.22 Comparison of State Points for Turbine Efficiencies of 90% and 100% for Base Case I, Turbine Cycle B.

of these methods, especially for the MHD generator. Reduction of the combustor irreversibility by reducing the amount of heat loss is an anticipated improvement in advanced combustor designs.

- The radiant heater is the second largest source of irreversibility for the Case I and Case II systems. Both the large amount of heat transfer and the large temperature difference across which it occurs are reasons for this large value. Reduction of the amount of heat transfer is not a feasible method of reducing $\dot{I}_{rev_E}$ for the radiant heater, since one of its functions is to reduce the gas temperature to a level which is acceptable to the downstream components. The possibility does exist, however, for the reduction of the temperature difference across which the heat transfer occurs. If it were feasible to use silicon carbide tubes embedded in the ceramic heater wall to carry the air coolant, the air could be heated to 1300 K or higher. This would reduce the value of $\dot{I}_{rev_E}$ by 20 MW.
- The fluid dynamic process in the diffuser is a substantial source of irreversibility in the system. The potential does exist, though, for considerable improvement. The value for the diffuser pressure recovery used in Case I, $C_p = 0.50$, is very conservative. Advanced diffuser designs are anticipated to have pressure recoveries of 0.65 or higher. Raising the assumed value of pressure recovery from the Case I value to 0.65 for Case II resulted in nearly a 40% reduction in $\dot{I}_{rev_E}/\dot{m}$. The reason the ideal system measure of irreversibility is used for the diffuser is that the diffuser performance influences the energy output from the MHD generator for the same combustor pressure. As the diffuser performance improves, the pressure ratio across the MHD generator increases, resulting in an increased enthalpy extraction. It is also for this reason that T_{WF} for the diffuser was chosen to be the same as that for the MHD generator.
- The values of $\dot{I}_{rev_E}$ for the MHD and gas turbine stacks for the base cases are substantial, but also are much less than the rates of energy being discharged by these stacks. For example, Fig. 3.2 shows that the MHD gas stack loss is 310 MW, whereas the value of $\dot{I}_{rev_E}$ for this stack is only 73 MW. This marked contrast shows the value of a Second Law (irreversibility) analysis over a First

Law (energy balance) analysis in properly indicating the sources of losses. However, a comparison of $\dot{I}_{rev_E}$ for the MHD stack with the $\dot{I}_{rev_E}$ for other components clearly indicates that the MHD stack is an important loss. The vapor cycle bottoming plant, which utilizes the exhaust stream energy, clearly reduces the value of $\dot{I}_{rev_E}$ for the MHD and turbine cycle B stacks. Thus, in the case of the stack irreversibilities, a great reduction in the size of the losses is very feasible. On the other hand, this reduction works against the main design objective of the MHD/gas turbine system - that of minimizing or eliminating the cooling water requirement. The proper trade off of low cooling water requirement versus increased system performance will have to be determined by criteria other than those considered in this section.

- The intercoolers have a relatively low value of $\dot{I}_{rev_E}$. This is another clear example of the inadequacy of First Law analysis, for the total energy flow exhausted by the intercoolers is five times the sum of $\dot{I}_{rev_E}$ for the intercoolers.
- The turbines have a higher value of $\dot{I}_{rev_E}$ than the compressors even though the turbine efficiency is greater than the compressor efficiency. This is due to the higher T_{WF} for the turbines.
- The gas turbine plant recuperators are only a moderate source of irreversibility. Recuperators HXA2 and HXA3 have the largest losses of the six, HXA3 because of its very low value of effectiveness and HXA2 because of its large heat load.

The irreversibility analysis of the MHD/gas turbine system has pinpointed components in which large losses occur, and which therefore are prime candidates for further development. It is interesting to note from the MHD/gas turbine system capital cost summary, which appears in Appendix A, that the components which have the largest values of $\dot{I}_{rev_E}$ have a low or modest capital cost, including the combustor, MHD generator, diffuser, and radiant heater. This generally indicates that the limitations to improved performance are constraints on materials and design and an early stage of development rather than the limitation of high cost. Thus, the improved thermal performance of the MHD/gas turbine system will be highly dependent on the effort spent on advancing component

development and overcoming material and design constraints.

3.8 Investigation of NO_x Emissions

3.8.1 Introduction

An important investigation which was performed as part of the MHD/gas turbine system study was a detailed modeling of the gas radiation heat transfer and nitric oxide decomposition in the radiant heater. Since the high temperature and long residence time of the gas in the combustor virtually assures that an equilibrium concentration of nitric oxide (on the order of 10,000 ppm) will exist in the gas at that location, the task of meeting the NO_x emissions standard requires that either conditions be provided for the decomposition of NO to a low level, or that the NO be removed from the gas stream by a separate process. This investigation considers the decomposition method of NO_x control which, if designed properly, can essentially assure complete compliance to the NO_x emissions standard.

The decomposition of NO to a level which meets the applicable standard can be accomplished by a combination of staged combustion and slow cooling of the MHD combustion gas in the radiant heater. The staged combustion is carried out in a simple manner by burning fuel-rich in the combustor and then adding secondary combustion air at a point downstream of the radiant heater where the NO formation rates are very slow. The slow cooling of the MHD gas, however, is not as straightforward. The temperature-time history strongly affects the final NO concentration of the gas, and since the gas cooling rates are not radially uniform in the cylindrical radiant heater, the temperature-time history of the gas must be known as a function of radius in the heater. In this investigation, a two-dimensional gas radiation model was used to calculate the radially dependent gas cooling rates. The NO concentrations and decomposition rates were then also calculated as a function of radius and were used to obtain an average NO concentration at the end of the radiant heater. In the following subsections are presented: the models for molecular and atomic gas radiation, the model for radiative energy exchange in the radiant heater, the NO_x kinetics model, and the results of the NO_x analysis for the radiant heater.

3.8.2 Band and Line Radiation Models

The gas radiation model in this investigation considered band radiation due to the CO_2 , H_2O , and CO molecular species, and line radiation from the potassium doublet line. Data for the band radiation were taken from Ludwig, et al.⁽¹⁴⁾ Figure 3.23 shows the spectral variation of the absorption coefficient which is based on this data. The line radiation for potassium is partially based on the experimental data obtained by Russian investigators.⁽¹⁵⁾ The behavior of the wings of the potassium line outside the region of the Russian data is presently unknown, so three different models for the absorption coefficient were used with the intention of determining the effect of this uncertainty. The model which is considered to represent the upper limit of the absorption coefficient in the wings is:

$$a_\lambda = \frac{2.2 \times 10^{-16}}{(\lambda - 7699)^{1.0}} \frac{2050}{T_g} \frac{P}{P_o} n_K, \lambda > 8199 \text{ \AA}$$

$$a_\lambda = \frac{4.8 \times 10^{-15}}{(7665 - \lambda)^{1.65}} \frac{2050}{T_g} \frac{P}{P_o} n_K, \lambda < 7165 \text{ \AA}$$

where

$$\begin{aligned} a_\lambda &= \text{spectral absorption coefficient, cm}^{-1} \\ \lambda &= \text{wavelength, \AA} \\ T_g &= \text{gas temperature, K} \\ P_o &= 1 \text{ atm} \\ P &= \text{gas pressure, atm} \\ n_K &= \text{potassium atom number density, cm}^{-3} \end{aligned}$$

This model is denoted "Russian/slope 1" in this work. The model which is considered to represent the lower limit of the absorption coefficient in the wings has the absorption coefficient equal to the value which is given by the Lorentzian model for the potassium line for the spectral regions outside of that covered by the Russian data. This model is denoted the "Russian/Lorentzian" model in this work. The third model which was used yields values for the absorption coefficient which are between the previous two models. The absorption coefficient outside the Russian data is given by:

$$a_\lambda = \frac{1.1 \times 10^{-13}}{(\lambda - 7699)^{2.0}} \frac{2050}{T_g} \frac{P}{P_o} n_K, \lambda > 8199 \text{ \AA}$$

Figure 3.23 Spectral Absorption Coefficient vs. Wavenumber.

$$a_{\lambda} = \frac{4.2 \times 10^{-14}}{(7665 - \lambda)^{2.0}} \frac{2050}{T_g} \frac{P}{P_o} n_K, \lambda < 7165 \text{ }^{\circ}\text{A}.$$

This model is denoted "Russian/slope 2."

For a bimodal particle size distribution such as calculated by Im and Chung⁽¹⁶⁾, it was found that emission and absorption by the entrained slag droplets was negligible. This result is based on the theory of McGregor⁽¹⁷⁾ which predicts a cutoff of emission and absorption by particles for wavelengths greater than $\pi\sqrt{2}$ times the particle diameter. Since most of the particles for the bimodal size distribution have a radius of less than 0.1 μm , the contribution of these particles to the radiative flux is very small. However, recent experiments⁽¹⁸⁾ do not support McGregor's theory and suggest that emission and absorption from small particles follows Mie theory. In this case, results by Ahluwalia and Im⁽¹⁹⁾ indicate that the radiation heat flux to the wall will be roughly 15% to 20% greater than the flux calculated in this study.

3.8.3 Radiative Energy Exchange Model

This model consists of three key parts, each employing simplifications to allow a manageable solution to the problem of radiative energy exchange in the radiant heater. The first part simplifies the treatment of the spectrally dependent absorption coefficient, the second part simplifies the treatment of radiative energy exchange in the cylindrical geometry, and the third part simplifies the treatment of fluid flow and mass transfer in the radiant heater.

The motivation for the first part of the model arises from the spectral dependence of emission and absorption of radiation by the gas. An exact calculation of the radiative energy transfer requires an integration over all wavelengths of the spectral radiation intensity, which could involve an extremely great computational effort. A good approximation to the exact solution, however, can be made by dividing the emission and absorption characteristics of the entire spectrum into a few representative groups and calculating the average values for each group. To calculate the total radiation intensity, a summation of the

intensities for each of the groups is then performed instead of an integration of spectral intensities over all wavelengths. The groups selected for this model, shown on the right side of Fig. 3.23, are:

1) the transparent group for spectral regions having absorption coefficients with a value less than 0.1 m^{-1} , 2) the near optically thick group for spectral regions having absorption coefficients between 0.1 m^{-1} and 1 m^{-1} , and 3) the very optically thick group for spectral regions having absorption coefficients greater than 1 m^{-1} .

The motivation for the second part of the model comes from the dependence of the energy absorbed by a unit volume of gas on the entire temperature field within the radiant heater. The value of this energy absorption term is needed in order to perform an energy balance on the unit volume of gas, and thus determine its cooling rate. The exact calculation of this term would require an iterative, three-dimensional computational procedure. However, several approximations can be made which can greatly simplify this calculation. For radiation in the transparent spectral group, the "transparent approximation" can be used which assumes that emission and absorption by the gas along an optical pathlength is negligible, resulting in the simplification that the radiation intensity for this group which is incident on the unit volume of gas is only a function of the radiant heater wall temperature and not dependent on the three dimensional gas temperature field. For radiation in the near optically thick and very optically thick spectral groups, two approximations can be made about the gas temperature field which greatly simplify the problem. The first approximation is that the temperature distribution is axisymmetric. The second approximation is that the effect of the axial temperature gradient on the gas cooling rate can be neglected since it has a much smaller magnitude than the radial temperature gradient, especially in the vicinity of the wall. The problem of calculating the energy absorption term in a three dimensional temperature field for the two optically thick spectral groups has thus been simplified to the calculation of the absorption term in a cylinder with only a radial temperature gradient. Calculation of this term was performed by applying the equation of radiative energy transfer along an optical path, which was defined by using the mean beam length approximation,

to evaluate the radiation intensity impinging on the unit volume of gas for each of the two spectral groups.

In the third part of the model, simplifications were made regarding the gas flow field in the radiant heater. The separation zone in the front section of the heater was assumed to extend 24 m into the heater, roughly six step heights. From the point of reattachment, plug flow was assumed for the remainder of the heater. In earlier work by Hals, et al.,⁽²⁰⁾ the effects of convective heat transfer, turbulence, and bouyancy were found to be small.

3.8.4 NO_x Kinetics Model

The model for the NO_x kinetics uses the extended-Zeldovich reaction mechanism which has been used in previous nitric oxide work for an MHD/steam system by Pepper.⁽⁶⁾ This mechanism is:

In this previous work, Pepper presents the following expression for the time rate of change of the NO concentration:

$$n_{\text{NO}} \cong 2(1 - \alpha^2) \left[\frac{R_1}{1 + \alpha K_1} \right]$$

where

$$\begin{aligned} \alpha &= [\text{NO}]/[\text{NO}]_e \\ R_1 &= k_{1b} [\text{N}]_e [\text{NO}]_e \\ K_1 &= R_1/(R_2 + R_3) \\ R_2 &= k_{2b} [\text{N}]_e [\text{O}_2]_e \\ R_3 &= k_{3b} [\text{N}]_e [\text{OH}]_e \end{aligned}$$

and the subscript e denotes equilibrium. The best rate constant data for the extended Zeldovich mechanism are for the forward reactions. The reverse rate constants for R₁, R₂, and R₃ are obtained by dividing the forward rate constants for these reactions by the equilibrium constants.

The forward rate constants, and resultant reverse rate constants are:

$$k_{1f} = 1.84 \times 10^{14} \exp(-76.25/RT) \text{ cm}^3/(\text{mole} \cdot \text{s}) \text{ from Ref. 21}$$

$$k_{2f} = 2.36 \times 10^9 T \exp(-38.64/RT) \text{ cm}^3/(\text{mole} \cdot \text{s}) \text{ from Ref. 22}$$

$$k_{3f} = 1.74 \times 10^{14} \exp(-49.20/RT) \text{ cm}^3/(\text{mole} \cdot \text{s}) \text{ from Ref. 23}$$

$$k_{1b} = 3.96 \times 10^{13} \exp(-1.00/RT) \text{ cm}^3/(\text{mole} \cdot \text{s})$$

$$k_{2b} = 1.04 \times 10^{10} T \exp(-6.55/RT) \text{ cm}^3/(\text{mole} \cdot \text{s})$$

$$k_{3b} = 6.32 \times 10^{13} \exp(-1.25/RT) \text{ cm}^3/(\text{mole} \cdot \text{s})$$

The equilibrium constants used to derive the backward rate constants were those presented in reference 23.

In the model for NO decomposition in the radiant heater, the following assumptions were made: 1) The NO concentration profile at the entrance of the radiant heater is greater than the equilibrium NO concentration profile by a constant factor; this factor was determined from a NO kinetics analysis of the gas from the combustor (where equilibrium is assumed) to the exit of the diffuser. 2) The gas flows through the radiant heater with a radially flat velocity profile and with no mixing. In the separation zone, the gas flowing in the core maintains a flat velocity profile while the gas in the recirculation region flows downstream in the vicinity of the core and flows upstream near the wall. An approximate analysis of radiant heat transfer in the separation zone was performed for this flow pattern so that the effect of the recirculation region on the radiant cooling of the core gas could be included. In the separation zone, the gas flowing in the core did not mix with the gas in the recirculation region. 3) No radial diffusion of NO occurs. 4) The NO neither decomposes nor forms downstream of the radiant heater, due to the rapid gas cooling which occurs in the downstream heat exchangers. Actually, the NO concentration in the gas will continue to decrease slowly as the gas flows from the radiant heater to the high temperature air heater at a nearly constant temperature. But upon

entering the high temperature air heater, any further decrease in the NO concentration is prevented by the very rapid cooling rates in the heater beds. The assumption that no NO is formed downstream of the radiant heater, even after the addition of the secondary combustion air, is supported by calculations with the NO_x kinetics model which show the additional NO formed after secondary combustion to be less than 1 ppm for Cases II and III. This is because the NO kinetics are virtually frozen at the temperature of the secondary combustion.

The second and third assumptions given above tend to make the NO decomposition model yield a conservative estimate, or an upper bound, on the average NO concentration at the exit of the radiant heater since mixing and diffusion would act to reduce the high NO concentration which is calculated for the gas near the wall in most cases. This point will become more clear in the next section. The fourth assumption also gives a conservative estimate of the final NO concentration. Because the main objective of the NO_x emissions investigation is to ensure that the plant design will meet the applicable NO emissions standard, the above assumptions are not unreasonable.

3.8.5 Results of NO_x Investigation

For the Case II systems, the average gas temperature and NO concentration in the radiant heater as a function of residence time are shown in Fig. 3.24. These results are for a wall temperature of $2000 \text{ K} \leq T_w \leq 1900 \text{ K}$, as specified in section 2.3, and the Russian/slope 2 potassium line radiation model. The radiant heater for the Case II systems was designed to have a residence time of 2.4 seconds, giving an average outlet temperature of 2000 K and an average NO concentration of 380 ppm. The 1979 NSPS for subbituminous coal is 0.5 lb/MBtu of NO_x , which is equivalent to a concentration of 408 ppm for the Case II conditions. Thus, compliance with this standard is obtained with the Case II radiant heater design. Results of an NO_x investigation for the oxygen enriched Case III systems showed that the average outlet NO concentration for a similarly designed radiant heater was 354 ppm.

The effects of combustor stoichiometry, wall temperature, and the potassium line radiation model were also studied in this investigation for the Case II systems. The combustor stoichiometry has the greatest

Figure 3.24 Average [NO] and Gas Temperature vs. Residence Time for Base Case II.

influence of these three variables on the NO concentration. The variation of the average NO concentration with combustor stoichiometry is shown in Fig. 3.25 for the Case II radiant heater design. The figure shows that the 0.5 lb/MBtu NO_x emissions standard is met by stoichiometries of 0.935 and lower. It is interesting to note the stoichiometric level which would be necessary to meet a NO_x level of 0.4 lb/MBtu and 0.2 lb/MBtu since these levels are considered to be possible standards in the future. The 0.4 lb/MBtu level could be met by a Case II radiant heater design with a combustor stoichiometry of 0.92 or lower, while the 0.2 lb/MBtu level would require a stoichiometry slightly lower than 0.88 or a stoichiometry of 0.88 in conjunction with a longer residence time. An analysis of the Base Case II MHD/gas turbine system with a stoichiometric level of 0.90 instead of 0.935 showed that the system performance for the 0.90 level is only moderately reduced from that for the 0.935 level. The thermal efficiency was 0.5 percentage points lower and the COE was 0.6 mills/kWhr higher. Thus the penalty incurred on the system for meeting at least a 0.4 lb/MBtu standard with the Case II radiant heater design would not be very large.

The effects of wall temperature and potassium line model on nitric oxide concentrations are much more dependent on the details and assumptions in the radiative energy exchange model and the NO decomposition model than is the effect of combustor stoichiometry. Both of these parameters strongly affect the radial temperature and NO concentration profiles, though in different ways. The potassium line model has a moderate influence on the absorption coefficient of the transparent and near optically thick spectral groups, and thus predominantly affects the radial temperature profile in the region which is 1 m or more in from the wall. The Russian/slope 1 model yields the highest absorption coefficients and therefore results in the most rapid gas cooling, while the Russian/Lorentzian model yields the lowest absorption coefficients and therefore the slowest gas cooling. The wall temperature, on the other hand, influences the cooling rate of the entire gas flow region, though particularly the region which is nearest the wall. Radial profiles of gas temperature and NO concentration for the three potassium line models and the three wall temperature specifications are shown in Figs. 3.26 and 3.27. The profiles in Fig. 3.26, for a residence time of

Figure 3.25 Variation of Average [NO] with Combustor Stoichiometry for Case II Radiant Heater Design.

Figure 3.26 Radial Temperature and [NO] Profiles at 1.05s for Variation in Wall Temperature and Potassium Line Model.

Figure 3.27 Radial Temperature and [NO] Profiles at 2.45s for Variation in Wall Temperature and Potassium Line Model.

1.05 seconds, all have the lowest NO concentrations existing at or near the wall. This is due to the fact that the equilibrium NO concentration decreases with temperature and therefore the concentrations in the vicinity of the wall are driven toward lower values than are the concentrations in the higher temperature regions. For the Case II combustion gas, for example, the equilibrium concentrations of NO at 2200 K, 2100 K, and 2000 K are respectively 940 ppm, 400 ppm, and 145 ppm. For the 1500 K and 1700 K wall temperatures the temperature gradient of the gas near the wall is very steep, due to the ability of the very optically thick spectral group to radiate a great amount of energy to the wall in this region. But since the mean penetration distance of the radiation in this spectral group is small, being only 0.5 m, regions much further than this distance from the wall are not cooled significantly by radiation which is in the very optically thick spectral group and thus are cooled at a much slower rate. The rise in NO concentration at the wall for the 1500 K and 1700 K wall temperatures is a result of the very rapid gas cooling which occurs at the wall, since the gas is quickly cooled to a temperature at which the NO decomposition reaction mechanism is effectively "frozen". The NO concentration at the wall thus remains at a relatively high level. This freezing process is shown graphically in Figs. 3.28 and 3.29 which give the development of the temperature and NO concentration profiles with time. In Fig. 3.28, the steepening gas temperature profiles at the wall show the rapid cooling in that region. In Fig. 3.29, the slow NO decomposition rate near the wall can be seen to produce the rise in the NO concentration profile at that location.

The profiles of the gas temperature at 2.45 seconds in Fig. 3.27 show the expected effects which are produced by increasing the wall temperature: higher temperature profiles and more uniform temperature profiles. For all three wall temperatures, the Russian/slope 1 potassium line model yielded the lowest temperatures and the Russian/Lorentzian model yielded the highest temperatures. The NO concentration profiles are markedly different from those at 1.05 seconds for the 1500 K and 1700 K wall temperatures in that the NO concentration in the core is lower than the concentration near the wall. This is due to the low temperatures, and therefore slow NO decomposition, near the wall which enabled the concentration in the core to decrease to a lower level than

Figure 3.28 Development of Radial Temperature Profile.

Figure 3.29 Development of Radial [NO] Profile.

at the wall. For the highest wall temperature however, the opposite is true since the temperature near the wall was high enough to prevent the NO decomposition rate from becoming much slower than the rate in the core. The peculiar behavior of the NO concentration at 7.2 m from the center of the heater is primarily a result of the assumption that the NO concentration at the radiant heater entrance is greater than the equilibrium concentration at each radial location by a constant factor. Since the initial temperature at 7.2 m was lower than the temperature of the gas nearer the center, the initial NO concentration was lower. Also, because of the lower temperature, the NO decomposition rate at 7.2 m is lower than the rate at nearby, radially inward locations. However, the difference between these initial NO concentrations was not made up by the opposing difference in decomposition rates, and the result of this is the drop in the NO concentration profile at 7.2 m. This peculiarity, then, arises from the model assumptions and the assumed initial temperature profile, and it quite probably will not be seen in an actual device. However, the reduction in the NO decomposition rates throughout the entire heater caused by low wall temperatures is a very real phenomenon which results in higher NO concentrations at the exit of the radiant heater. This effect is clearly seen in Fig. 3.30, which shows the average NO concentration at the radiant heater exit as a function of residence time for the three wall temperature specifications. For a 2.5 second residence time, the difference between the NO concentrations for the variable temperature hot wall and the 1700 K wall ranges from 50 ppm to 95 ppm, depending on the potassium line model used. The difference between the NO concentrations for the hot wall and the 1500 K wall for the same residence time ranges from 110 ppm to 170 ppm. Evidence of the freezing of the NO decomposition reaction for the 1500 K and 1700 K wall temperatures can also be seen in Fig. 3.30, indicated by the flattening of the NO concentration vs. time profiles.

In conclusion, the NO_x emissions investigation showed that the 0.5 lb/MBtu NO_x standard could be met by the Case II radiant heater design. By lowering the combustor stoichiometry, lower levels of NO_x emissions could be obtained at a rather small penalty to the system performance. Furthermore, the wall temperature of the radiant heater influenced the NO_x emission levels. Wall temperatures of 1700 K

Figure 3.30 Average [NO] vs. Residence Time for Variation in Wall Temperature and Potassium Line Model.

and 1500 K were shown to produce a freezing of the NO decomposition mechanism after a residence time of 2.0 to 2.5 seconds. The potassium line radiation model also influenced the NO concentration, but to a lesser extent. For the 1700 K and 1500 K wall temperatures, the Russian/slope 1 model produced the fastest gas cooling rates and the highest average NO concentrations while the Russian/Lorentzian model produced the opposite results. The average NO concentration was only weakly affected by the choice of potassium line radiation model for the highest wall temperature.

CHAPTER 4

COMPONENT MODELS

4.1 Introduction

The component modeling was guided by the philosophy of employing the degree of detail necessary for an accurate assessment of system cost and performance. For some components, such as turbines, compressors, and generators, the performance can be adequately modeled by the use of a component efficiency. For other components, such as the MHD generator, a more detailed model is required in order to account for important phenomena which affect the cost and performance. And other parts of the power plant system, such as waste ash handling, have an insignificant effect on the system thermal performance, but do affect the system cost. These components will only have a cost model. Therefore, the degree of detail in the model of each component is determined by the role which is played by the component in the system and by the detail which is required to allow an accurate assessment of its effect on the system thermal performance.

The cost models for most of the power plant components are based on the cost figures which are presented in the EPRI/W study⁽³⁾ for the Plant Design 1 system. Most of the component cost models also include a functional dependence on appropriate parameters which relates the component's cost to its size. This functional dependence was usually taken from an Argonne National Laboratory report⁽²⁴⁾ on a systems analysis of MHD/steam systems. Some components, such as the diffuser, radiant heater, and gas turbine plant heat exchangers, required a detailed cost analysis. The costs for materials which were used in these components are presented in Table 4.1. Except where noted otherwise, the costs were taken from the EPRI/W study⁽³⁾ and are in mid-1978 \$. A list of costs for all of the following components and subsystems for Cases I, II, and III is given in Appendix D.

Table 4.1. Cost of Materials

<u>Material</u>	<u>Density(lb/ft³)</u>	<u>Cost (\$/lb)</u>	
<u>Piping and Tubing</u>			
Haynes 188	570	30.2	+
Hastelloy X	513	12.7	+
310 Stainless Steel	504	2.60	
Croloy 304 H	501	2.49	
Croloy 2 1/4 (T22)	501	1.02	
Croloy 1 1/4	501	0.84	
Low Carbon Steel	495	0.70	
<u>Plate, Sheet, and Structural Steel</u>			
Haynes 188	570	26.2	+
Hastelloy X	513	8.7	+
Alloy Steel	501	0.84	
Low Carbon Steel	495	0.70	
<u>Insulation and Refractories</u>	<u>Density</u>	<u>Cost(includes installation labor)</u>	
High density castable spinel for ceramic wall liner	167 lb/ft ³	219 \$/ft ³	*
Blazelite 2000 outer insulation (6" thickness)	55 lb/ft ²	10.2 \$/ft ²	*
Gas Manifold Insulation (12.5" thickness)	70 lb/ft ²	29.7 \$/ft ²	*
Air Manifold Insulation (6" thickness)	55 lb/ft ²	15.3 \$/ft ²	*

+ Cost from Stellite Div., Cabot Corp.⁽²⁵⁾, mid-1980 \$

* Cost based on Fluidyne Information⁽²⁶⁾, mid-1980 \$

4.2 Models

MHD Channel

A quasi one-dimensional model was used for the segmented Faraday MHD channel in which an integral model was used for the boundary layers. While not as accurate as a two-dimensional differential code, the quasi 1-D method has the great advantage of being computationally fast while still being able to consider important MHD boundary layer phenomena such as Joulean dissipation and extended Hartmann flow. A second order Euler predictor-corrector method with an axial stepsize of 0.1 m was found to give excellent accuracy in the solution of the ordinary differential equations of the model. The details of the MHD channel model, including the treatment of boundary layer voltage drops, electrode wall boundary layer heating due to Joulean dissipation, and extended Hartmann flow on the insulating wall, are given in Appendix E.

The constraints on the electrical parameters in the channel design were $E_y < 4.0$ kV/m for the transverse field, $E_x < 3.5$ kV/m for the axial field, and $J_y < 1.0$ A/cm² for the average transverse current density. The wall temperature of the channel was specified to be 2000 K.

The MHD channel had an aspect ratio of 1.0 (square in cross section), and the channel loft was specified by $C \equiv u\Delta u/(\Delta P/\rho) = 0.10$. This value for C results in a decrease in the Mach number of the core flow from the channel inlet to the channel outlet. Lower values of C result in a lower deceleration of the core flow (e.g. $C \approx 0.05$ gives constant Mach number flow, $C = 0.0$ gives constant velocity flow), while higher values of C result in greater deceleration of the core flow. The system performance in this study was found to gradually increase with increasing C , but an upper limit on its value was set at 0.10 because flow separation may be a problem for values that are significantly greater than this. The channel inlet Mach number was set at its upper constraint of $M = 0.9$, maximizing the system performance. The purpose of the upper constraint was to avoid transonic flow in the channel.

The electrical load parameter used in this study is defined by the average field between the cathode and anode of an electrode pair (which

is less than the field in the core):

$$K \equiv \frac{E_{yc} - V_D/H}{u_c B}$$

where

- E_{yc} = core value of E_y
- V_D = total voltage drop (anode + cathode)
- H = spacing between electrode walls
- u_c = core velocity
- B = magnetic field

The electrical load parameter for the channel was set at 0.8, but allowed to drop below this value in order to meet the transverse field (E_y) constraint. The value of 0.8 optimized the system performance for all of the MHD/gas turbine systems studied.

The cost formula for the MHD channel is based on the cost in the EPRI/W study, with roughly the same functional variation as in the Argonne study.

MHD Channel Cost =

$$2.4 \times 10^6 [(0.36(A_i + A_o)L/2)^{0.6} + 0.17 \dot{m}^{0.3}/A_i]/6.2$$

where

- A_i, A_o = channel inlet and outlet areas, m^2
- L = channel length, m
- $\dot{m}$ = mass flow rate in channel, kg/s.

Magnet

The superconducting magnet was specified to have a constant magnetic field of 6T, except at the MHD channel entrance and exit where the field is tapered to 5T over 1.5 m of axial length. The cost formula for the magnet is based on the cost in the EPRI/W study and the functional variation in the Argonne study.

$$\text{Magnet System Cost} = 60.9 \times 10^6 (3.8 \text{ EQTBV}^{0.48}/48.5)$$

where
$$\text{EQTBV} = L\pi/4[(R_i + R_o)^2 + (R_o - R_i)^2/3]$$

$$L = \text{MHD Channel length, m}$$

$$R_i = 0.43 + .707 \sqrt{A_i}$$

$$R_o = R_i + 0.07 L$$

$$A_i = \text{Area of MHD Channel Inlet, m}^2$$

Combustor and Nozzle

The combustor considered for the MHD/gas turbine system is of a two stage design. The stoichiometry in the first stage was set at 0.50 and the first stage liquid slag rejection was set at 0.95. This yielded overall ash rejection values of 81%, 85%, and 92% for Cases I, II, and III respectively. A lower first stage stoichiometry would increase the ash rejection, but since incomplete char burnout is a problem for very low stoichiometries, the value of 0.5 was used. The combustor heat loss was set at 4% of the coal HHV thermal input, the same value used in the EPRI/W study,⁽³⁾ with 3% occurring in the first stage and 1% in the second stage. The remainder of the MHD combustor air is added to the second stage, along with the potassium seed, to give an overall stoichiometry of 0.95 for Case I and 0.935 for Cases II and III. The potassium seeding level by weight is 1% for Case I and 0.7% for Cases II and III. The combustor pressure drop was set at $\Delta P/P = 0.047$, also the same value used in the EPRI/W study.⁽³⁾ Illinois #6 coal, dried to 0% moisture, was used for Case I, and Montana Rosebud coal, dried to 5% moisture, was used in Cases II and III. The coal compositions are given in Appendix F, and are the same

as those used in the PSPEC studies^{(4),(5)} with the exception that SO_3 was not included in the ash composition. The removal of the SO_3 is based on the belief that its inclusion in the ash composition is erroneous.⁽²⁷⁾

The chemical composition and temperature of the combustion gas in the first and second stages of the combustor were determined by the chemical equilibrium program EQUIL. This program is described in Appendix G. A simple non-ideal solution model is used in EQUIL to determine the equilibrium state of the liquid and vapor slag mixture. Based on experimental data by Byker,⁽²⁸⁾ the activity coefficients of $\text{CaO}_{(l)}$ and $\text{SiO}_{2(l)}$ for the first combustor stage were set at 0.005 and 1.0 respectively, and the activity coefficients of $\text{CaO}_{(l)}$ and $\text{SiO}_{2(l)}$ for the second stage were set at 0.01 and 0.5 respectively. The other components of the liquid slag solution were assumed to behave ideally, and therefore had activity coefficients of 1.0. In the first stage of the combustor, 95% of the liquid slag solution is rejected. Since the composition of this solution is not the same as the mineral matter input to the first stage due to slag vaporization, the composition of the slag carried over to the second stage will also be different from the mineral matter input. The energy of the slag rejected in the first stage is assumed to be unrecoverable, and therefore treated as an additional heat loss.

The nozzle, which is located at the combustor exit, accelerates the flow to Mach 0.9. An analysis of this process with a boundary layer code gave a heat loss of 3 MW and a value for the momentum thickness Reynolds number at the nozzle exit of 5000.

The cost model for the combustor and nozzle is based on the combustor and nozzle cost in the EPRI/W study, and uses approximately the same functional variation as that in the Argonne report. The cost formula, which includes direct and indirect labor, is:

$$\text{Combustor and Nozzle Cost} = 1.3 \times 10^6 (\dot{m}/690)^{0.6}$$

where $\dot{m}$ is the total mass flow rate at the exit of the combustor, kg/s.

Diffuser

The diffuser for the MHD/gas turbine system is conical in shape with ceramic lining on the side of the wall, or shell, which is exposed to the MHD exhaust gas. This shell is cooled by compressed air which flows along its outer side in an annular, concentric passage formed by an outer shell. The diffuser length ranges from 37 m for Case I to 68 m for Case III with corresponding length to inlet diameter ratios ranging from 12 to 25. The mean gas outlet velocity from the diffuser is nominally 90 m/s for Cases I and II and 40 m/s for Case III. The model for the diffuser consists of three parts: the MHD gas flow model, the air cooling model, and the cost model. The model for the MHD gas flow is approximate in nature, while the models for the air cooling and the cost are more detailed. A great deal of uncertainty exists about the character of the flow at the exit of the MHD channel due to secondary flows and the extended Hartmann effect in channel. This complex flow is carried into the diffuser and makes an accurate analytical prediction of flow inside the diffuser very difficult. For this reason, the approximate model is appropriate for the purposes of this study.

In the MHD gas flow model, the pressure recovery of the diffuser was set to a specified value, influenced in part by the flow blockage which the MHD channel model calculates for the channel exit. This variation with blockage, as well as the variation of pressure in the diffuser with axial distance, is based on an experimental study of the pressure recovery of conical diffusers by Dolan and Runstadler.⁽²⁹⁾ For Cases II and III, the pressure recovery was 0.65 with the diffuser having an inlet blockage of approximately 6%. The pressure recovery in Case I was set at 0.50, which is the value used in the EPRI/W study, independent of the inlet flow blockage. In determining the wall heat transfer and the gas cooling rate, the MHD gas flow in the diffuser was modeled as plug flow with the velocity being determined by the flow area and the gas density from the continuity equation. The expansion angle of the diffuser was $2\theta = 6.2^\circ$. The wall heat transfer rate was the sum of the convective and radiative heat transfer rates, where the radiative transfer rate accounted for both molecular band radiation from

CO_2 and H_2O and for line radiation from potassium atoms. In the calculation of the gas cooling rate, a radially uniform gas temperature profile was assumed. The details of the MHD gas flow model for the diffuser, as well as a presentation of the air cooling model, are presented in Appendix H.

The cost model for the diffuser consists of calculations for seven components of the diffuser cost which are summed to obtain the total diffuser cost. A description of the cost model is also presented in Appendix H.

Radiant Heater

The cylindrical radiant heater has a diameter of 16 m for Cases I and II and a diameter of 14 m for Case III. The gas flow in the radiant heater is separated from the wall in the front part of the heater due to the abrupt area expansion which exists at the junction of the ~8 m diffuser exit with the 16 m or 14 m heater entrance. The reattachment point was assumed to be located roughly 6 step heights from the radiant heater entrance. (1 step height = radiant heater radius - diffuser exit radius). The total length of the radiant heater ranges from 45 m for Case I to 55 m for Case III. The wall temperature is specified to be 1600 K at the entrance, rising to 2000 K in the region of flow reattachment and falling to 1900 K at the exit. The gas flow and heat transfer models will be only briefly described here since they have been presented in greater detail in Section 3.8 of this report.

In the gas flow model, the gas in the recirculation region is assumed to not mix with the gas in the core region, and the gas is assumed to flow with a uniform velocity (plug flow) downstream of the reattachment point. The velocity of the gas in the plug flow region is nominally 20 m/s. The gas heat transfer model assumes that energy exchange in the heater occurs only by means of radiative heat transfer. The model accounts for molecular band radiation by CO_2 , H_2O , and CO, and for line radiation by potassium.

The air cooling model for the radiant heater is the same as that for the diffuser (described in Appendix H) with the only difference being that the annular air flow passage is cylindrical in shape rather than conical in shape.

The cost model for the radiant heater is the same as that for the diffuser (see Appendix H), with the thickness of the outer shell and the insulation also being the same.

High Temperature Air Heater

The directly-fired high temperature air heater (HTAH), which has been described previously in Section 2.3 of this report, is similar to the FY-80 design by Fluidyne Engineering Corporation.⁽⁷⁾ Since the performance of the HTAH is fixed by the specification of the MHD air preheat temperature and by the results of the MHD/gas turbine system thermodynamic analysis, the sole function of the HTAH model is to determine the cost of the HTAH system. Cost information for the FY-80 design was also obtained from Fluidyne,⁽²⁶⁾ and was used in the derivation of an approximate functional relationship between the HTAH size and cost. For the purpose of obtaining this functional relationship, the cost of the HTAH was divided into three categories: 1) the cost of the ceramic matrix beds and the bed supports; 2) the cost of the steel heater vessels and the vessel insulation; 3) the cost of the manifolds, valves, and the heater support steel. The costs of these three categories were expressed as functions of the number of heat transfer units (nondimensional), N_{tu} , and the MHD exhaust gas mass flow rate, $\dot{m}_g$. While the size of the HTAH is actually a function also of the MHD combustion air flow rate, this is related to $\dot{m}_g$ by a constant factor for the same combustor stoichiometry. Therefore, a relative change in $\dot{m}_g$ will be accompanied by the same relative change in the air mass flow rate, allowing the functional relationship to be accurately written using only one of the mass flow rates. The first cost category, that for the beds and bed supports, was set to be directly proportional to the product of N_{tu} and $\dot{m}_g$ since the heat transfer area is directly proportional to this product. The second cost category, that for the vessel steel and insulation, was set to be proportional to the 1/2 power of both N_{tu} and $\dot{m}_g$. The third cost category, that for the manifolds, valves, and support steel, was set to be proportional to the 1/2 power of $\dot{m}_g$. This category is, to the first order, unaffected by changes in N_{tu} .

The resultant cost formula has a functional variation which is based on the cost of the Fluidyne FY-80 system and an overall cost which is based on the HTAH cost for the EPRI/W system.⁽³⁾ This formula,

which includes the cost of direct and indirect labor, is:

$$\begin{aligned} \text{Cost of HTAH system} = & [49 \times 10^6 (N_{tu, FY80}) (\dot{m}_g / \dot{m}_{g, FY80}) + \\ & 38 \times 10^6 (N_{tu} / N_{tu, FY80})^{0.5} (\dot{m}_g / \dot{m}_{g, FY80})^{0.5} + \\ & 65 \times 10^6 (\dot{m}_g / \dot{m}_{g, FY80})] (0.882) (1.084) \end{aligned}$$

where

$N_{tu, FY80}$ = number of heat transfer units for FY-80 design (=3.64)

$\dot{m}_{g, FY80}$ = gas mass flow rate for FY-80 design (=860 kg/s).

The factor of 0.882 near the end of the expression deflates the mid-1980 dollars of the FY-80 case to mid-1978 \$ which are used in this study. The last term in the expression adjusts the HTAH cost so that the formula yields the HTAH cost of the EPRI/W study for the N_{tu} and $\dot{m}_g$ corresponding to that study. The N_{tu} for the HTAH is calculated from the design specifications by the following formula:

$$N_{tu} = \frac{-\ln[(1-\epsilon')/(1-\epsilon' C_{min}/C_{max})]}{1 - C_{min}/C_{max}}$$

$$\epsilon' = \epsilon \left[1 - \frac{1}{9(C_r/C_{min})^{1.93}} \right]$$

where C_{min} = minimum flow-stream capacity rate ($\equiv \dot{m} c_p$).

C_{max} = maximum flow-stream capacity rate.

C_r = matrix capacity rate (= total matrix mass x matrix specific heat x cycles per unit time).

ϵ = actual heat exchanger effectiveness.

The pressure drops for the air side and gas side of the HTAH were specified to be the same as those in the EPRI/W study, namely

$$\Delta P / P_{air} = 0.03, \quad \Delta P / P_{gas} = 0.03.$$

Gas Turbine Plant Turbine/Generator/Compressor Sets

The thermal performance of the turbines, generators, and compressors was modeled by the use of an isentropic efficiency for each component. This study used a turbine efficiency of 0.90, a generator efficiency of 0.985, and a compressor efficiency of 0.88. The components were assumed to be adiabatic. The turbines, generators, and compressors were costed as a unit because cost data were available in the form of \$/kW power output from the unit. The cost formula for the turbine/generator/compressor set is based on the topping cycle cost for the ECAS II gas turbine/steam combined-cycle system⁽²⁾ and includes all labor costs.

$$\text{Cost of Turbine/Generator/Compressor Sets} = 103 \cdot \text{PGT}$$

where PGT = turbine shaft power output, kW.

This cost formula agrees well with current costs for gas turbine electrical generating units, accounting for inflation between mid-1978 dollars and current dollars.

Intercoolers

The intercoolers, like the gas turbine plant heat exchangers, have their thermal performance completely specified. They are designed and costed in exactly the same manner as the gas turbine plant heat exchangers, with the exception that the stream which flows inside the tubes is the hot side and the stream in crossflow over the tubes is the cold side.

DC/AC Inversion Units

The conversion of the MHD generator power from DC to AC involves two processes, one being a consolidation of the DC current from multiple pairs of electrodes, and the other being the conversion of the high voltage DC power to AC power by means of inverters. The electrical conversion efficiency of this two-step process was taken to be 98.5%, which is the value that was used in the EPRI/W study.⁽³⁾ The cost formula is also based on the EPRI/W study, with the functional variation taken from the Argonne study.

$$\text{Cost of DC/AC Inversion Units} = 52.3 \times 10^6 (\text{PWRDC}/621)^{0.51}$$

where PWRDC = MHD Generator Power, MW_{DC}.

MHD Air Compressor

The thermal performance of the MHD air compressor was modeled using an isentropic efficiency of 0.88. The cost of the MHD air compressor is based on the cost in the EPRI/W study, with an additional allowance of 30% to account for compressor sizing which is greater than that required. This allows some redundancy, which is a requirement if the plant is to continue operation during compressor unit maintenance or repairs. The allowance of 30% was taken from a compressor arrangement in the EPRI/W study.⁽³⁾ The cost formula, which includes direct and indirect labor, is

$$\text{MHD Air Compressor Cost} = 65.5 \cdot \text{PCMHD}$$

where PCMHD = compressor power requirement, kW.

Gas Turbine Plant Heat Exchangers

The model for the gas turbine plant heat exchangers consists of a design routine and a cost routine. The design routine employs the effectiveness- N_{tu} method of Kays and London.⁽³⁰⁾ In the MHD/gas turbine system analysis code, the heat exchanger effectiveness, pressure drops, outer tube diameters, heat transfer surface specifications, and number of passes are input variables, while the inlet and outlet temperatures, pressures, and mass flow rates are determined in the thermodynamic analysis of the system. Therefore, the function of the heat exchanger design routine is to determine the heat transfer surface area, tube wall thickness, and heat exchanger dimensions which will give the specified performance. The cost routine uses the results of the design routine to determine the heat exchanger cost.

For the three MHD gas/air heat exchangers (HXA2, HXA3, HXB2), because the tube material may be very costly, an analysis of the optimum tube material and wall thickness is determined for each pass. For the finned, circular tube air/air heat exchangers the tube material and wall thickness is the same for all of the passes since the service requirements do not vary significantly within the heat exchangers. The design procedure employed in the model for the gas turbine heat exchangers is given in Appendix I.

The cost model considers nine cost categories for the MHD gas/air heat exchangers and seven for the air/air heat exchangers. A description

of these cost categories and the calculation of the total heat exchanger cost is also given in Appendix I.

Flue Gas Desulfurization

The flue gas desulfurization process which is used in Case I is the same as that used in the EPRI/W study.⁽³⁾ The process has a sulfur removal efficiency of 95% and the auxiliary power requirement for processing 87% of the flue gas, as in the EPRI/W study, is 17.5 MW. The capital cost of the system is the same as in the EPRI/W study,

$$\text{Flue Gas Desulfurization Cost} = 29.43 \times 10^6.$$

Seed Recovery System

This system is based on the seed recovery system designed by Westinghouse.⁽³⁾⁽⁸⁾ Its function is to separate the water-soluble seed from the ash and to dry the seed for either recycle back to the combustor, as in Case I, or for delivery to the seed regeneration system, as in Cases II and III. The energy requirement of the seed recovery system per 10 kg/s seed flow is 0.32 MW_e of electrical power and 34.7 MW_t of steam.

In the Westinghouse analysis of the seed recovery system, both a capital cost and an annual cost are presented. For Case I, which had a seed recovery system of the same size as that of the EPRI/W MHD/steam system,⁽³⁾ the capital cost in the EPRI/W study was used directly,

$$\text{Seed Recovery System Cost} = 31.63 \times 10^6.$$

The operating and maintenance cost of this system was included in the O & M cost for the entire MHD/gas turbine system.

For Cases II and III, which did not attempt to compare the MHD/gas turbine system performance with the EPRI/W MHD/steam system performance, a separate seed reprocessing component of the total cost of electricity was calculated. An annual cost of the seed recovery system was calculated for these cases which was based on the annual cost presented in Reference 8. The annual cost used in the present study includes the amortized capital cost and the levelized operating and maintenance cost, with the exception that the electrical power requirement of the seed recycle system

is included in the overall auxiliary power requirement. The formula for the annual cost is,

Annual Cost of Seed Recovery System =

$$8.9 \times 10^6 [0.4(\dot{m}_{\text{seed}}/10)f_c f_{\text{level}} + 0.6(\dot{m}_{\text{seed}}/10)^{0.7}] \cdot 0.910$$

where $\dot{m}_{\text{seed}}$ = mass flow rate of seed to combustor, kg/s

f_c = plant capacity factor

f_{level} = levelization factor.

The first term in the brackets accounts for the levelized annual operating cost and the second term accounts for the annual capital cost. The 0.7 power in the second term scales the capital cost to the system size and was taken from the EPRI/W study.⁽³⁾ The 0.910 factor at the end of the expression deflates the annual cost from end-1979 \$, which were used in Reference 8, to mid-1978 \$.

The capital cost of the seed recovery system was also calculated. This cost was needed so that the total capital cost and the unit capital cost of the MHD/gas turbine system could be calculated. However, since the contribution of the seed recovery system capital cost to the cost of electricity is included in the seed reprocessing component, COE_{seed} , this capital cost is not included in the capital cost component, $\text{COE}_{\text{capital}}$. The capital cost formula for this system is,

Seed Recovery System Cost =

$$29.0 \times 10^6 (\dot{m}_{\text{seed}}/10)^{0.7} \cdot 0.910$$

Seed Regeneration System

The seed regeneration system which is used in Cases II and III is based on the Westinghouse process.⁽⁸⁾ This process uses a two step reduction/oxidation scheme to convert K_2SO_4 to K_2CO_3 , producing elemental sulfur as a by-product and having no waste streams which require special treatment or disposal. The energy requirement of the system per 10 kg/s K_2SO_4 flow is 7.91 MW_e of electrical power and 147 MW_t of coal. The process produces 38.2 MW_t of steam per 10 kg/s K_2SO_4 flow,

which can be used to meet the steam energy requirement of the seed recovery system.

As for the seed recovery system, an annual cost of the seed regeneration system is calculated in this study. This annual cost is also based on Reference 8 and includes the same costs as for the seed recovery system. The cost formula is,

Annual Cost of Seed Regeneration System =

$$8.0 \times 10^6 \left[0.45 \left(\frac{\dot{m}_{K_2SO_4}}{10} \right) f_c f_{level} + 0.55 \left(\frac{\dot{m}_{K_2SO_4}}{10} \right)^{0.7} \right] \cdot 0.910$$

where $\dot{m}_{K_2SO_4}$ = mass flow rate of K_2SO_4 to seed regeneration system, kg/s
 f_c = plant capacity factor
 f_{level} = levelization factor.

The first term in the brackets accounts for the levelized operating cost and the second term accounts for the capital cost. The 0.7 power factor in the second term and the 0.910 factor at the end have the same meaning as in the seed recycle system cost formula.

The capital cost of the seed regeneration system is given by,

Seed Regeneration System Cost =

$$23.6 \times 10^6 \left(\dot{m}_{K_2SO_4} / 10 \right)^{0.7} \cdot 0.910 .$$

As for the seed recycle system, the contribution of the capital cost of this system to the cost of electricity is included in the seed reprocessing component, COE_{seed} , and is not included in the capital cost component, $COE_{capital}$.

Ammonia Vapor Cycle Bottoming Plant

The ammonia vapor cycle bottoming plant is similar to the ammonia bottoming plant used in the ECAS II closed-cycle gas turbine system (NASA Modified Case 4).⁽²⁾ The pinch point temperature difference for the ammonia heater is 18 K, which gives a heater effectiveness in the vicinity of 0.9 for the vapor cycle variation systems. The condenser outlet temperature is 306 K (90°F), its pressure is 1.27 MPa (185 psia), and

its $\Delta P/P$ is 0.09. The ammonia turbine efficiency is 0.83 and the pump efficiency is 0.70. The effectiveness of the condensed ammonia preheater is 0.80. This preheater regenerates some of the energy of the superheated ammonia turbine exhaust, therefore reducing the amount of heat rejected to the cooling towers and reducing the cooling water requirement. The turbine inlet pressure for the supercritical ammonia system is 20 MPa (2900 psia). The turbine inlet temperature for most of the vapor cycle variation systems is 620 K. The thermodynamic characteristics of the ammonia cycle for this turbine inlet temperature, on a per unit mass of ammonia working fluid basis, are: turbine work output = 483 kJ/kg, pump work input = 45 kJ/kg, thermal energy input = 1616 kJ/kg, condenser energy rejection = 1178 kJ/kg, net work output = 438 kJ/kg, thermal efficiency = 27.1%. The turbine inlet temperatures for Case I, Variation 2 and Case I, Variation 4 are slightly lower than 620 K (604 K and 609 K respectively), and have slightly different thermodynamic characteristics than those just presented.

The cost model of the vapor cycle plant is based on the cost of the ammonia bottoming plant in the ECAS II closed-cycle gas turbine system (NASA Modified Case 4).⁽²⁾ The vapor cycle power equipment is costed on the basis of $\$/kW_e$ power output and the cooling tower heat rejection system is costed on the basis of $\$/kW_t$ of energy rejection. The cost formula is:

$$\text{Cost of Ammonia Vapor Cycle Bottoming Plant} = 265 \cdot PWR_{AMM} + 13.2 \cdot QREJ$$

where PWR_{AMM} = net power output from ammonia cycle, kW_e

$QREJ$ = cooling tower heat rejection, kW_t

The auxiliary power requirement for the cooling tower heat rejection system is 10 kW_e per MW_t of heat rejection. This figure is based on the Avco PSPEC study⁽⁴⁾ and includes the power requirement for circulating water pumps and cooling tower draft fans.

Stack and Exhaust Gas System

This system consists of the exhaust stacks, hot gas ducting, draft fans, and electrostatic precipitators. The isentropic efficiency of the draft fans was set to be 0.80, and the auxiliary power requirement for

the electrostatic precipitator (ESP) units is 0.4 MW_e . The cost formula for the stack and exhaust gas system, which is based on the costs in the EPRI/W study and includes all labor costs, is:

Cost of Stack & Exhaust Gas System =

$$0.0161 \dot{m}_{\text{gas}} + 0.095 \text{ PFAN} + 4.4 + 2.7 (\dot{m}_{\text{gas}}/743)^{0.5}$$

where $\dot{m}_{\text{gas}}$ = MHD exhaust gas mass flow rate, kg/s.

PFAN = draft fan power requirement, MW_e .

The first term is for the ESP cost, the second term is for the draft fan cost, the third term is for the stacks, and the fourth term is for the hot gas ducting.

Coal Processing System

This system includes the coal pulverizers and dryers and the coal feed system. The thermal performance of the coal dryers is given in Section 2.3 of this report. The cost of the coal processing system is based on the cost in the EPRI/W study with the functional variation being based on the Argonne study. The cost formula, which includes all labor costs, is:

$$\text{Cost of Coal Processing System} = 23.4 \times 10^6 (\dot{m}_{\text{coal}}/75.1)^{0.51}$$

where $\dot{m}_{\text{coal}}$ = mass flow rate of as-received coal, kg/s.

The auxiliary power requirement for the coal processing system is 0.13 MW_e per kg/s of as-received coal flow.

Air Separation Plant

The air separation plant used in Case III is based on the air separation plant design of the PSPEC studies.⁽⁴⁾⁽⁵⁾ This plant produces an ambient pressure flow stream of 70% O_2 and 30% N_2 which is then mixed with an air flow stream to produce a combined oxidizer flow stream which is 35% O_2 by volume. The power requirement of the air separation plant is 197.8 kWhr per ton of equivalent pure O_2 . The cost of the air

separation plant is calculated on the basis of $\$16.64 \times 10^6$ per 2000 ton per day facility. This is a turnkey cost. The values for power requirement and cost are the same as those used in the PSPEC studies.

Other Components and Systems

The following components and systems have only a cost model which is based on the costs in the EPRI/W study. The functional variation of the cost formula, if given, was taken from the Argonne study. These remaining components and systems, together with those previously presented in this chapter, account for all of the appropriate cost items which were used in the EPRI/W study.

$$\text{Site Preparation Cost} = 39.3 \times 10^6 \left(\frac{\text{PNET}}{858.6} \right)^{0.9}$$

where PNET = net electrical power generation, MW_e .

$$\text{Seed Storage and Handling Cost} = 4.4 \times 10^6 (\dot{m}_{\text{seed}}/15.4)^{0.4}$$

where $\dot{m}_{\text{seed}}$ = seed mass flow rate, kg/s.

Bottoming Plant Electrical Equipment Cost =

$$43.5 \times 10^6 [(\text{PNET}/2 + \text{PWRBOT})/908]^{0.53}$$

where PWRBOT = bottoming plant (gas turbine + vapor cycle) electrical power generation, MW_e .

$$\text{Coal Receiving and Storage Cost} = 24.0 \times 10^6 (\dot{m}_{\text{coal}}/75.1)^{0.16}$$

where $\dot{m}_{\text{coal}}$ = mass flow rate of as-received coal, kg/s.

$$\text{Waste Ash Handling Cost} = 2.8 \times 10^6 (\dot{m}_{\text{coal}}/75.1)^{0.36}$$

$$\text{Generator and Combustor Cooling Cost} = 1.0 \times 10^6$$

$$\text{Instrumentation and Controls Cost} = 31.8 \times 10^6$$

$$\text{Miscellaneous Power Plant Equipment Cost} = 12.3 \times 10^6 (\text{PNET}/858.)^{0.5}$$

$$\text{Miscellaneous MHD Support Systems Cost} = 1.0 \times 10^6$$

CHAPTER 5

SUMMARY AND CONCLUSIONS

In this work, three cases of MHD/gas turbine systems were investigated for their thermal and economic performance and their cooling water requirement. The Case I systems were designed to be very similar to an MHD/steam system so that a direct comparison of the performance of the two systems could be made. The Case II systems had some second generation MHD plant components, used seed regeneration to control the sulfur emissions, and used a low sulfur Western coal which is appropriate for the water-limited regions of the country. The Case III systems eliminated the directly-fired high temperature air heater which was used in Cases I and II by the use of oxygen enrichment so that the effect of this change, which is currently envisioned for early commercial MHD systems, could be determined. Each case consisted of a base case system in which the MHD plant was bottomed by a gas turbine plant whose only working fluid was air, and one or more vapor cycle variations of the base case in which a small ammonia vapor cycle bottoming plant was added to the system. The MHD/gas turbine systems for each of the cases were modeled by using realistic, and sometimes rather conservative, component specifications and performances. The component and subsystem costs of the MHD/gas turbine systems were based on previous work and current manufacturers' data.

The Base Case I MHD/gas turbine system has a thermal efficiency of 39.7% and a non-levelized cost of electricity of 46.9 mills/kWhr. This system requires no cooling water. The vapor cycle variations of Case I have thermal efficiencies which range from 42.8% to 46.3% and costs of electricity which range from 44.7 mills/kWhr to 42.5 mills/kWhr as the cooling tower heat loads of the systems range from 210 MW to 560 MW. The MHD/steam system after which the Case I systems were patterned has a thermal efficiency of 44.7% and a cost of electricity of 43.9 mills/kWhr along with a cooling tower heat load of 720 MW.

The Base Case II MHD/gas turbine system has a station thermal efficiency of 41.7% and a levelized cost of electricity of 58.3 mills/kWhr.

This base case system has no cooling tower heat rejection. The vapor cycle variations for Case II have thermal efficiencies which range from 44.6% to 48.0% and costs of electricity which range from 55.8 mills/kWhr to 52.9 mills/kWhr as the cooling tower heat loads vary from 200 MW to 550 MW. The cost of electricity for Base Case II, when calculated with the same economic rules of Case I, has a non-levelized value of 42.0 mills/kWhr.

The Base Case III MHD/gas turbine system has a thermal efficiency of 36.9% and a levelized cost of electricity of 60.8 mills/kWhr, and has no cooling water requirement. The one vapor cycle variation of Case III has a thermal efficiency of 41.0% and a cost of electricity of 56.6 mills/kWhr with a cooling tower heat load of 350 MW.

Conclusions which have been drawn from these results, and from the investigation of the system sensitivity to key variables, the irreversibility analysis, and the nitric oxide analysis, are presented below.

- The Base Case I MHD/gas turbine system requires no cooling water and has a system performance which is not much less than that of an MHD/steam system - a thermal efficiency which is 11% lower and a cost of electricity which is 7% higher (a difference which is only 1 mill/kWhr greater than the estimated relative uncertainty of the comparison). The ability of the MHD/gas turbine system to eliminate its need for cooling water, along with its good performance, makes the system an attractive alternative for electrical power generation in the water-limited regions of the country.
- The Base Case I MHD/gas turbine system thermal efficiency of 39.7% and 0 MW cooling tower heat rejection compare favorably with a comparably sized conventional coal-fired steam plant thermal efficiency of 36% and cooling tower heat load of 1100 MW.
- The addition of a small vapor cycle bottoming plant to the MHD/gas turbine system substantially improves both the thermal and the economic performance, but also imposes a limited cooling water requirement and some additional system complexity. For regions in

which the use of a small amount of cooling water is feasible, the addition of a vapor cycle plant may be a desirable option.

- The Case II systems show that an improvement in thermal performance on the order of 5% and an improvement in economic performance on the order of 10% can be expected for systems which have second generation MHD plants and which use low sulfur Western coal and seed regeneration.
- The Case III systems show that the use of oxygen enrichment, which offers greater reliability and a lower technological risk, substantially lowers the thermal performance of the MHD/gas turbine system relative to Case II, but has a relatively small negative impact on the economic performance of the system. The oxygen-enriched system may therefore be an attractive, early commercial MHD/system for siting in dry regions of the country.
- The MHD/gas turbine system performance displays a broad optimum with respect to the MHD plant variables of combustor pressure and electrical load parameter. The system performance, in general, shows greater sensitivity to the gas turbine plant variables.
- The irreversibility analysis has shown that the largest irreversibilities tend to occur in the MHD plant due to the high temperatures at which the processes in the plant take place. These irreversibilities are for the most part attributable to fundamental materials and design constraints rather than a trade-off of performance versus cost.
- The nitric oxide decomposition analysis has shown that the applicable 0.5 lb/MBtu nitric oxide standard can be met by the Case II and III MHD/gas turbine systems.
- A possible future standard of 0.4 lb/MBtu could be met by the Case II systems if the combustor stoichiometry were lowered from 0.935 to 0.92 (fuel rich). The possible, far future standard of 0.2 lb/MBtu could be met by a combination of an increased radiant heater residence time with a combustor stoichiometry of 0.88.
- The radiant heater wall temperature has a significant effect on the nitric oxide emissions level, with wall temperatures of

1700 K and lower causing the nitric oxide decomposition reactions to freeze in some regions of the radiant heater.

- The choice of potassium line radiation model has a moderate effect on the calculated gas cooling rate and level of nitric oxide emissions. The uncertainty in the behavior of the potassium line in the far wings is a limitation on the accuracy of the radiant heater model.

In summary, the MHD/gas turbine systems studied in this report show great flexibility in meeting the environmental constraints of power plant sites in the future, and offer very good thermal and economic performances. For power plant siting in regions of limited water supply, this study has shown the MHD/gas turbine system to be an especially attractive alternative to the MHD/steam system.

APPENDIX A

COST OF ELECTRICITY CALCULATION PROCEDURE

This appendix describes the procedure that was used to calculate the cost of electricity (COE) of the MHD/gas turbine systems in this study. The COE, which is representative of the economic performance of the system, is determined by summing the contributions to the COE of capital ($\text{COE}_{\text{capital}}$), fuel (COE_{fuel}), operating and maintenance ($\text{COE}_{\text{O \& M}}$), and for Cases II and III, seed reprocessing (COE_{seed}). The calculation of each of these components of the COE will be given in detail in the following paragraphs.

Capital Cost Portion of the COE

The value of $\text{COE}_{\text{capital}}$ is a function of the capital equipment cost (including direct and indirect labor), the power rating of the plant, the plant capacity factor, and other economic rules which are presented in Table A.1. With the exception of the fixed charge rate, and plant capacity factor, the other parameters which are presented in Table A.1 are used in the calculation of the total constructed plant cost. If the total constructed plant cost has been calculated, $\text{COE}_{\text{capital}}$ can be easily determined by

$$\text{COE}_{\text{capital}} (\text{mills/kWhr}) = \frac{f_{\text{fcr}} \text{ TCP} \cdot 1000}{8765.8 \text{ PNET } f_{\text{capf}}}$$

where TCP = total constructed cost of plant, \$

PNET = power plant rating, kW_e

f_{fcr} = fixed charge rate

f_{capf} = plant capacity factor.

As shown in Table A.1, the fixed charge rate for all three Cases is 0.18, based on a return (cost of capital) of 10%. This is the fraction of the total constructed plant cost which must be paid annually to cover the amortized cost of the capital expenditure, the insurance and taxes, and other capital related expenses. A detailed explanation of the fixed

Table A.1

Economic Rules Used in the Calculation of COE_{capital}

	<u>Case I</u>	<u>Cases II & III</u>
Contingency Factor for Conventional Components	0.10	0.10
Contingency Factor for Unconventional Components	0.10	0.20
A & E Service Factor	0.10	0.10
Indirect Labor Factor	0.90	0.90
Construction Time	6 1/2 yr	6 1/2 yr
Interest During Construction	10%/yr	10%/yr
Factor for Escalation and Interest During Construction	1.699	1.679
Escalation (Inflation) Rate	6.5%/yr	6.5%/yr
Fixed Charge Rate	0.18	0.18
Plant Capacity Factor	0.70	0.65

charge rate and the derivation of its value of 18% can be found in reference 12. The plant capacity factor is the total amount of electrical energy generated in a year divided by the maximum possible yearly generation (= 8765.8 hrs/yr x PNET). The value of the capacity factor is based on previous utility experience and on the anticipated reliability of the unit under consideration. The capacity factor for Case I is the same as that used in the EPRI/W study.⁽³⁾ The capacity factor used for Cases II and III is the same as that used in the PSPEC studies.⁽⁴⁾⁽⁵⁾

The calculation of the total constructed cost of the plant is slightly more involved since it includes not only the total capital cost, but also the cost of architecture and engineering (A & E) services, the cost of contingencies, and the cost of escalation and interest during construction. The total capital cost of the plant (CCAP) has been defined in this study to consist of the cost of equipment, materials, and direct and indirect labor. The factors used by the major architect and engineering firms to calculate the costs of indirect labor, A & E services, and contingencies vary among the firms due to their different construction procedures and past experiences. The factors which are used in this study follow those which were used in the EPRI/W study,⁽³⁾ the PSPEC studies,⁽⁴⁾⁽⁵⁾ and by Westinghouse/C. T. Main in the ECAS II study.⁽²⁾ The cost formulas for the plant components and subsystems, which were presented in Chapter 4, include the cost of direct and indirect labor. (The indirect labor cost was set to be 90% of the direct labor cost). Therefore, CCAP is calculated by summing the costs of the components and subsystems of Chapter 4. The cost of A & E services is calculated by

$$\text{Cost of A \& E services} = \text{A \& E Service Factor} \cdot \text{CCAP}.$$

The cost of contingencies is calculated by the use of a contingency factor. This is a percentage factor that is applied to allow for costs that are expected to be incurred in the construction of the power plant, but were not specifically accounted for in the total capital cost of the plant. It accounts for additional costs likely to be encountered due to incomplete designs, estimating errors and omissions, minor scope changes and unforeseen construction problems. The contingency factor is based on an A & E firm's previous construction experience. The contingency factor

is not applied to subsystems which have a turnkey cost, such as the air separation plant which is used in Case III. The cost of contingencies for Case I is given by

$$\text{Contingency Cost} = \text{Contingency Factor} \cdot (1 + \text{A \& E Service Factor}) \cdot \text{CCAP}.$$

For Cases II and III in which the contingency factor for conventional and unconventional components differs, CCAP must be separated into the cost of conventional components (CCAP_{con}) and the cost of unconventional components ($\text{CCAP}_{\text{uncon}}$), with the different contingency factors applied to the proper group. For Cases II and III, the unconventional components were considered to be: combustor and nozzle, MHD channel, magnet, diffuser, high temperature air heater (Case II), and the consolidation equipment for DC/AC power inversion. These are the same components that were considered to be unconventional in the Avco PSPEC study.⁽⁴⁾

The factor by which the total capital cost of the plant, the cost of A & E services, and the cost of contingencies are increased to account for escalation and interest during construction is based on an assumed "S" shaped cash flow curve of expenditure vs. time for the construction period. The value for the factor of escalation and interest during construction for Case I was taken from the EPRI/W study.⁽³⁾ For Cases II and III, this factor was based on the value corresponding to a 6 1/2 year construction time presented in the Avco PSPEC⁽⁴⁾ and ECAS II⁽²⁾ studies. In all of the Cases, the factor is based on an annual interest rate of 10% and an annual escalation rate of 6.5%. The factor for escalation and interest during construction, however, produces a total constructed cost which is in end-1984 dollars instead of mid-1978 dollars. This cost must be deflated by the factor $(1.065)^{6.5}$ to have a total constructed cost of the plant which is in mid-1978 dollars. The formula, then, for the total constructed plant cost can be written as,

$$\begin{aligned} \text{TCP} = & [\text{CCAP}_{\text{con}}(1+f_{\text{cntg,con}}) \\ & + \text{CCAP}_{\text{uncon}}(1+f_{\text{cntg,uncon}})](1+f_{\text{A\&E}})^f \text{escint} / (1+f_{\text{esc}})^t \text{const} \end{aligned}$$

where CCAP_{con} = capital cost of conventional components
 $\text{CCAP}_{\text{uncon}}$ = capital cost of unconventional components

$f_{\text{cntg,con}}$ = contingency factor for conventional components
 $f_{\text{cntg,uncon}}$ = contingency factor for unconventional components
 $f_{\text{A\&E}}$ = A & E service factor
 f_{escint} = factor for escalation and interest during construction
 f_{esc} = escalation (inflation) rate during construction
 t_{const} = construction time

This is the same procedure for obtaining TCP and $\text{COE}_{\text{capital}}$ as that used in the EPRI/W study.⁽³⁾ The economic rules presented in Table A.1 for Case I have the same value as in the EPRI/W study, while some of the values have been changed for Cases II and III to match those used in the PSPEC studies.⁽⁴⁾⁽⁵⁾

Fuel Cost Portion of the COE

The COE_{fuel} which was used in Case I was a non-levelized or "first year commercial operation", COE. This method gives the contribution of the fuel to the COE in mid-1978 dollars for the first year of operation. The COE_{fuel} reported in the EPRI/W study⁽³⁾ is a first year commercial operation COE. The COE_{fuel} which was used in Cases II and III is a levelized COE. A levelized COE_{fuel} accounts for the inflation of the fuel costs over the life of the plant by multiplying the non-levelized value of COE_{fuel} by a levelization factor. The value of the levelization factor, which is the same as that used in the GE PSPEC study,⁽⁵⁾ is 1.882 and is based on a 30 year plant lifetime, a 6.5% inflation rate and a 10% discount rate (cost of capital). The procedure used in the calculation of the levelization factor can be found in reference 31 or reference 12. Relative to the levelized COE_{fuel} , the first year commercial operation COE_{fuel} tends to make the capital cost portion of the COE more significant than the fuel and O & M portions. The levelized method considers the fact that, in present year dollars, the fuel cost escalates over the plant life while the capital portion stays constant and is therefore less significant than in the first year commercial operation method. A more detailed presentation of the concept and procedure of levelized costs can be found in Reference 12.

The expressions for COE_{fuel} can be simply written as:

$$\text{First Year Commercial Operation } COE_{fuel} \text{ (mills/kWhr)} = \frac{C_{coal}/292.9}{\eta} (1000) \text{ (non-levelized)}$$

$$\text{Levelized } COE_{fuel} \text{ (mills/kWhr)} = \frac{C_{coal}/292.9}{\eta} (1000) (1.882)$$

where C_{coal} = price of coal in mid-1978 \$, \$/MBtu
 η = station thermal efficiency of power plant

In the above expressions, the factor of 292.9 converts the price of coal from \$/MBtu to \$/kWhr. The factor of 1.882 in the levelized COE_{fuel} expression is the levelization factor. The price of coal in mid-1978 \$ used in Case I was \$1.166/MBtu and the price of coal used in Cases II and III was \$1.35/MBtu.

Operating and Maintenance Portion of COE

The value of $COE_{O \& M}$ for Case I was set to be the same as that for the EPRI/W Plant Design 1 system, ⁽³⁾ which was 4.89 mills/kWhr. This value includes the O & M costs associated with the flue gas scrubbers and the seed recycle system. The value of $COE_{O \& M}$ for Cases II and III does not include the O & M costs associated with seed reprocessing, since these costs are included in the separate portion of COE for seed reprocessing. For all three Cases, the cost of seed makeup is included in $COE_{O \& M}$. Like the fuel portion for COE, the O & M portion of COE is levelized for Cases II and III. The base, non-levelized value for $COE_{O \& M}$ for these cases was 2.7 mills/kWhr. This value was determined by taking the average of the EPRI/W value, excluding the O & M costs for scrubbers and seed recycle, and the ECAS II value ⁽²⁾ of 2.1 mills/kWhr in mid-1978 \$. This value of 2.7 mills/kWhr is within 0.1 mill/kWhr of the average of the GE PSPEC and Avco PSPEC values for $COE_{O \& M}$. When levelized by the levelization factor of 1.882, the same as for COE_{fuel} , the value of $COE_{O \& M}$ for Cases II and III is 5.08 mills/kWhr.

Seed Reprocessing Portion of the COE

As described in the seed recovery system model in Chapter 4, Cases II and III have separate contribution to the COE for the seed reprocessing system, which consists of seed recovery and seed regeneration. This contribution included both the amortized capital and the O & M cost of the system (excluding the seed makeup cost, which is included in $COE_{O \& M}$). A levelization factor of 1.882 is applied to the O & M portion of the seed reprocessing system cost. The value of COE_{seed} is determined from the annual costs of the seed recovery and seed regeneration systems which were given in Chapter 4. The formula for COE_{seed} is:

$$COE_{seed} \text{ (mills/kWhr)} = \frac{(ACSRC + ACSRG) \cdot 10^9}{8765.8 \text{ PNET } f_{capf}}$$

where ACSRC = annual cost of seed recycle system, \$

ACSRG = annual cost of seed regeneration system, \$

PNET = power plant rating, kW_e

f_{capf} = plant capacity factor.

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APPENDIX B

PROCEDURE FOR ADJUSTMENT OF MHD/STEAM SYSTEM PERFORMANCE

The MHD channels for the EPRI/W Plant Design 1 MHD/steam system and the Base Case I MHD/gas turbine system have the following performance characteristics:

	<u>MHD/Steam</u>	<u>MHD/Gas Turbine</u>
Net Electrical Power	612.0 MW _{AC}	632.8 MW _{AC}
Gross Electrical Power	621.3 MW _{DC}	642.4 MW _{DC}
Pressure Ratio	5.64	4.93

Substituting the performance of the channel of the MHD/gas turbine system for that of the channel of the MHD/steam system results in an increase in net channel electrical power of 20.8 MW_{AC} for the MHD/steam system, and in a reduction of energy to the steam plant of 21.1 MW. Using the 39.3% thermal efficiency of the steam plant in the MHD/steam system, the 21.1 MW reduction in energy to the steam plant produces a reduction in steam plant electrical power output of 8.3 MW_{AC}. Also, the pressure ratio for the MHD channel is less for the MHD/gas turbine system than for the MHD/steam system. By comparing the compressor specific work requirement of the EPRI/W Plant Design 10 with that of the EPRI/W Plant Design 1, the approximate correlation can be obtained that the specific work requirement is proportional to the 0.310 power of the pressure ratio. Thus, the compressor power requirement for the MHD/steam plant is adjusted downward to a value of $169.7 \cdot (4.93/5.64)^{0.310} = 162.7$ MW. For the improved channel performance in the MHD/steam plant, 7.0 MW less shaft power is needed to be produced by the compressor drive steam turbine, and this power can be shifted to drive the generator. The net MHD/steam system power output is then revised from a value of 858.6 MW_e to a value of $858.6 + 20.8 - 8.3 + 7.0 = 878.1$ MW.

The adjustments in the capital cost of the MHD/steam system were made by replacing the magnet and channel cost of the MHD/steam system with that of the MHD/gas turbine system and by reevaluating the costs

of the inverters, site preparation, and bottoming plant electrical equipment for the revised channel power and total plant power, based on the cost formula in Chapter 4. The adjustments for the costs, which include direct and indirect labor, are presented below.

<u>Item</u>	<u>Old Cost(10⁶\$)</u>	<u>Revised Cost(10⁶\$)</u>	<u>Change(10⁶\$)</u>
Magnet	60.93	58.49	- 2.44
MHD Channel	1.84	2.14	+ 0.30
DC/AC Electrical Conversion	52.27	53.21	+ 0.94
Site Preparation	39.27	40.16	+ 0.89
Bottoming Plant Electrical Equipment	43.46	43.86	<u>+ 0.40</u>
			+ 0.09

Thus, the cost of the MHD/steam system changes by an insignificant amount as a result of the improved MHD channel performance. Due to the increased power output of 878.1 MW, the revised COE for the MHD/steam plant is reduced by 0.9 mills/kWhr, with the revised breakdown of COE given below.

	<u>EPRI/W Value</u>	<u>Revised Value</u>
COE _{capital}	30.74	30.06
COE _{fuel}	9.12	8.92
COE _{O & M}	4.89	4.87
COE _{total}	44.75	43.85

APPENDIX C

COMPARISON OF MHD/STEAM AND MHD/GAS TURBINE SYSTEM COSTS

MHD/Steam, EPRI/W Plant Design 1

MHD/Gas Turbine, Base Case I

Account No.	Description	Cost*	Cost	Description
310,311	Land, Land Rights, Structures, and Improvements	40.2	36.2	Site Preparation
312.11	Coal Receiving and Storage	24.0	24.2	Coal Receiving and Storage
312.12	Coal Pulverizers and Feeders	23.4	24.0	Coal Processing
312.2	Slag and Ash Handling	2.6	2.9	Waste Ash Handling
312.3	Radiant Boiler and Superheater	11.3	10.0	Radiant Heater
312.4	Steam Generator Sections	17.2	1.0	Generator and Combustor Cooling
312.51,.52	Slag Taps & Quench	0.2	----	(included in Waste Ash Handling)
312.53	ESP and Seed Collection	12.0		
312.54	ID Fans	1.9		
312.55	Stack	2.7	19.5	Stack and Exhaust Gas System
312.56	Hot Gas Ducting	2.7		
312.57,.58	Flue Gas Desulfurization & Limestone Handling	29.4	29.4	Flue Gas Desulfurization & Limestone Handling
312.6	Auxiliary Boiler Systems	1.5	----	N/A
312.7	Other Boiler Plant Systems	4.6	----	N/A
	N/A	----	100.1	Gas Turbine Plant Heat Exchanger
312.8	Instrumentation and Controls	31.8	31.8	Instrumentation and Controls
314.	Turbogenerator Units	52.8	19.6	Gas Turbine/Generator/Compressor Sets
315.	Accessory Electrical Equipment	32.6	28.7	Part of Bottoming Plant Electrical Equipment
316.	Miscellaneous Power Plant Equipment	12.3	11.7	Miscellaneous Power Plant Equipment
317.11,.12	Combustor & Nozzle	1.3	1.3	Combustor and Nozzle
317.22	MHD Channel	2.1	2.1	MHD Channel
317.23	Diffuser	2.3	2.6	Diffuser
317.3	Magnet Subsystem	58.5	58.5	Magnet
317.4	Inverters and Electrode Control	53.2	53.2	DC/AC Inversion Units
317.51	MHD Air Compressor	13.0	9.8	MHD Air Compressor
317.52	Air Compressor Steam Turbine	12.7	15.6	Gas Turbine/Compressor Units for Air Compressor
317.53,.54	Low Temperature Air Heater & Ducting	54.2	----	(replaced by Radiant Heater)
317.55	High Temperature Air	109.4	94.9	Heater High Temperature Air Heater
317.61	Seed Storage and Injection	4.4	4.0	Seed Storage and Handling
317.62	Seed Recycle	31.6	31.6	Seed Recycle
317.8	Miscellaneous MHD Support Systems	1.0	1.0	Miscellaneous MHD Support Systems
350.	Transmission Plant	11.2	10.0	Part of Bottoming Plant Electrical Equipment

*Cost include direct and indirect labor, and adjustments for improved MHD channel performance.

APPENDIX D
COMPONENT AND SUBSYSTEM COSTS

Cost* (includes direct and indirect labor), 10⁶ \$

Item	Base Case I	Case I Var. 1	Case I Var. 2	Case I Var. 3	Case I Var. 4	Base Case II	Case II Var. 1	Case II Var. 2	Case II Var. 3	Case II Var. 4	Base Case III	Case III Var. 1
Site Preparation	36.2	38.7	39.3	40.6	41.5	40.7	43.3	44.4	45.0	46.2	36.5	40.1
Combustor and Nozzle	1.3	1.3	1.3	1.3	1.3	1.5	1.5	1.5	1.5	1.5	1.1	1.1
MHD Channel	2.1	2.0	2.0	2.0	2.0	2.5	2.5	2.5	2.5	2.5	2.0	2.0
Diffuser	2.6	3.2	3.2	3.2	3.2	4.3	4.5	4.5	4.5	5.3	9.8	8.7
Magnet	58.5	53.7	53.7	53.7	53.7	65.2	63.6	63.6	63.6	63.6	63.8	63.8
Seed Storage and Handling	4.0	4.0	4.0	4.0	4.0	3.7	3.7	3.7	3.7	3.7	3.1	3.1
Seed Reprocessing	61.0	61.0	61.0	61.0	61.0	39.1	39.1	39.1	39.1	39.1	33.3	33.3
Stack and Exhaust Gas System	19.5	19.7	19.7	19.7	19.7	20.9	21.0	21.0	21.0	21.0	15.9	15.9
High Temperature Air Heater	94.9	94.5	94.5	94.5	94.5	125.8	125.8	125.8	125.8	125.8	-	-
MHD Air Compressor	9.8	10.3	10.3	10.3	10.3	10.6	11.5	11.5	11.5	11.5	6.2	6.8
Radiant Heater	10.0	9.7	9.7	9.7	9.7	10.4	10.3	10.3	10.3	10.4	15.6	17.8
Gas Turbine Plant Heat Exchangers												
HXA1	8.6	10.6	7.7	7.5	7.1	5.2	9.0	8.4	5.9	5.7	9.2	9.9
HXA2	26.9	25.5	25.6	25.6	26.6	28.3	24.5	24.3	27.5	21.8	32.0	30.0
HXA3	22.9	23.2	22.8	23.1	23.0	22.8	22.0	20.6	22.1	21.9	18.0	17.7
HXB1	4.3	6.6	-	6.1	-	5.0	7.8	-	8.0	-	9.3	10.3
HXB2	16.9	14.2	16.6	15.2	17.1	19.4	19.0	20.7	14.9	19.2	23.2	19.9
HXM1	5.8	-	-	-	-	6.4	-	-	-	-	3.1	-
Intercoolers	14.7	12.7	7.3	2.4	-	15.7	13.1	6.1	6.0	0.0	20.0	8.9
Total	100.1	92.9	80.0	79.9	72.8	102.8	95.4	80.1	84.4	68.6	114.8	96.7
Turbine/Generator/Compressor Sets	35.2	36.4	34.5	32.9	31.4	36.7	37.6	34.9	34.0	31.8	50.7	47.7
DC/AC Inversion Units	53.2	52.5	52.5	52.5	52.5	58.0	57.8	57.8	57.8	57.8	47.2	47.2
Bottoming Plant Electrical Equipment	38.8	42.1	42.7	43.9	44.8	40.7	43.7	44.8	45.3	46.4	42.9	46.5
Coal Receiving and Storage	24.2	24.2	24.2	24.2	24.2	25.2	25.2	25.2	25.2	25.2	25.2	25.2
Coal Processing	24.0	24.0	24.0	24.0	24.0	27.5	27.5	27.5	27.5	27.5	27.5	27.5
Waste Ash Handling	2.9	2.9	2.9	2.9	2.9	3.1	3.1	3.1	3.1	3.1	3.1	3.1
Generator and Combustor Cooling	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0
Instrumentation and Controls	31.8	31.8	31.8	31.8	31.8	31.8	31.8	31.8	31.3	31.3	31.3	31.8
Miscellaneous Power Plant Equipment	11.7	12.2	12.3	12.5	12.7	12.6	13.0	13.2	13.3	13.5	11.8	12.4
Miscellaneous MHD Support Systems	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0
Ammonia Vapor Cycle Bottoming Plant	-	23.7	34.6	48.6	60.1	-	22.4	39.3	46.4	61.7	-	39.3
Air Separation Plant	-	-	-	-	-	-	-	-	-	-	68.0†	68.0†
Total	623.6	642.6	640.1	655.1	659.9	665.2	686.3	687.8	699.3	700.1	602.4	630.3

* All costs are in mid-1978 \$

† Cost of Air Separation Plant is turnkey cost which includes A&E Services and Contingency Costs

APPENDIX E

MHD CHANNEL MODEL

The model for the MHD channel consisted of a one-dimensional model for the core coupled with an integral model for the electrode wall boundary layer and correlations obtained from a two-dimensional analysis for the insulator wall boundary layer. The model assumes that the core region is characterized by uniform properties and a uniform velocity. Because of the segmented-Faraday design of the channel, the approximation is made that

$$J_x = 0.$$

For the magnetic field in the z direction and the gas velocity in the x direction, the core equations can be written as

$$\text{Core Momentum Equation} \quad \frac{dP}{dx} = J_{yc} B - G_c \frac{du_c}{dx}$$

$$\text{Core Energy Equation} \quad G_c \frac{dh_{oc}}{dx} = J_{yc} E_{yc} - q_{rad}'''.$$

The enthalpy, h_o , in the energy equation is the total enthalpy, and the pressure, P , in the momentum equation is the static pressure. The radiation heat transfer term, q_{rad}''' , accounted for molecular band radiation by CO_2 and H_2O , and also line radiation by potassium. The band radiation was calculated by the use of charts for the emissivity and absorptivity of the gas presented by McAdams.⁽³²⁾ The line radiation was set to be 50% greater than the value obtained from a classical Lorentzian model of the potassium line. This gives a value which is closely equal to the average of the Lorentzian value for the line radiation and the Russian/slope 1 (see Section 3.8.2) value for the line radiation.

The integral models for the electrode wall boundary layer are based on those used by James.⁽³³⁾ The momentum integral equation for the electrode wall, following James,⁽³⁴⁾ can be written as

$$\frac{d\delta_2}{dx} = \frac{\tau_w}{G_c u_c} - \delta_2 \left[\frac{1}{w} \frac{dw}{dx} + \frac{1}{\rho_c} \frac{d\rho_c}{dx} + \left(2 + \frac{\delta_1}{\delta_2} \right) \frac{1}{u_c} \frac{du_c}{dx} \right]$$

where

$$\delta_1 = \frac{1}{\rho_c u_c} \int_0^l (\rho_c u_c - \rho u) dy$$

$$\delta_2 = \frac{1}{\rho_c u_c^2} \int_0^l \rho u (u_c - u) dy$$

l = distance from electrode wall to the center of the channel

w = distance from the insulator wall to the center of the channel.

For this equation J_y is assumed constant in the y direction, resulting in the pressure gradient compensating exactly for the MHD force. Thus neither pressure gradient terms nor MHD body force terms appear in the above integral equation.

The energy integral equation is similar to the simplified boundary layer model of James, ⁽³⁵⁾ which includes the effect of Joulean heating, with the exception that the equation includes consideration of radiant heat transfer, magnetic field effects, and a changing core enthalpy. Constant area flow is assumed in the model. The model allows the enthalpy boundary layer thickness, Δ , to differ from the momentum boundary layer thickness, δ , and assumes that the enthalpy of the gas beyond a finite enthalpy boundary layer thickness is equal to the enthalpy of the core of the gas. (This assumption allows the limits of integration to be changed from l to Δ in the definition of Δ_2 and r below). The energy integral equation can then be written as

$$\frac{d\Delta_2}{dx} = \frac{q_w''}{G_c H_c} + \frac{J_{yc} u_c B_o \delta_o}{G_c H_c} - \frac{J_{yc}^2 r}{G_c H_c} - (\delta_1 + \Delta_2) \frac{J_{yc} E_{yc}}{G_c H_c}$$

where

$$\Delta_2 \equiv \frac{\int_0^{\Delta} \rho u (h_{oc} - h_o) dy}{G_c H_c}$$

$$\delta_o \equiv \int_0^l \left(1 - \frac{u}{u_c}\right) dy$$

$$r \equiv (1 + \beta^2) \int_0^{\Delta} \left(\frac{1}{\sigma} - \frac{1}{\sigma_c} \right) dy$$

$$H_c = h_{oc} - h_{ow}$$

h_{oc} , h_{ow} = gas total enthalpy of the core and at the wall.

The term in this equation which accounts for the effect of Joulean heating is $J_{yc}^2 r / G_c H_c$. The additional boundary layer resistance over that of the core is represented by r , which is a function of Δ . The effect of Joulean heating on the boundary layer is to reduce Δ_2 and therefore Δ (related to Δ_2 by the assumption of a 1/5th power law profile), and in turn reducing r since the upper limit of integration is lowered. James shows that an ordinary differential equation can be written for the ratio of the enthalpy boundary layer thickness to the momentum boundary layer thickness, $\omega (\equiv \Delta/\delta)$, from the above energy integral equation. Assumptions which are made in the derivation of this ODE for ω are: 1) a 1/5th power law profile for both enthalpy and velocity for rough wall flow behavior⁽³⁶⁾; 2) a variation of the Stanton number with ω which is similar to that for the flat-plate, thermal entry length correlation given by Kays⁽³⁷⁾:

$$St = St_o (\Delta/\delta)^{-1/5}$$

where St_o is the Stanton number for flat-plate flow in which $\Delta = \delta$, and the $-1/5$ power replaces the $-1/7$ power used by Kays because of the different profile being assumed. For these assumptions, then, the ODE for ω which can be obtained from the energy integral equation is

$$\frac{d\omega}{dx} = \frac{5}{6} \frac{1}{\delta_2} \left[\omega^{-2/5} St_o - \omega \frac{d\delta_2}{dx} \right] - 7 \left[\omega^{-1/5} U^* + \omega^{1/5} \theta^* + \left(\frac{1}{6} \omega^{-1/5} + \frac{5}{42} \omega \right) JE^* \right]$$

where

$$U^* = - \frac{\delta_1}{\delta} \frac{J_{yc} u_c B}{G_c H_c}$$

$$\theta^* = \frac{J_{yc}^2 r^*}{G_c H_c}, \quad r^* = r/\Delta$$

$$JE^* = \frac{J_{yc} E_{yc}}{G_c H_c}$$

In the MHD channel model, the above equation for ω was solved to obtain $\Delta (= \omega \delta)$ which was then used to obtain the boundary layer resistance, r . The electrode voltage drop was then determined by the following relation:

$$V_{drop} = u_c B \delta_1 / \delta + |J_{yc}| r + V_{arc}$$

The first term accounts for the "fluid dynamic" voltage drop, which is the decrease in voltage due to the velocity defect of the boundary layer; the second term accounts for the increased resistance of the boundary layer over that of the core; and the third term accounts for the voltage across the arc which is assumed to transport the current through the slag layer. V_{arc} was set to be 50 V. This expression gives the voltage drop over each electrode, so that the total voltage drop between the electrodes is twice the value of V_{drop} .

The insulating wall was modeled by using correlations for the momentum displacement thickness, δ , the enthalpy boundary layer thickness, Δ , and the Stanton number, St , which were obtained from runs for typical MHD channel designs of a two-dimensional boundary layer code written by Rankin.⁽³⁸⁾ The average current density in the insulating wall boundary layer was assumed to be 75% of J_{yc} . This approximation was used by Hokenson⁽³⁹⁾ and agrees well with the results of Rankin's code.

The electrical parameters for the MHD channel, $\overline{E}_y$, E_{yc} , J_{yc} , and E_{xc} , are determined by the specification of the electrical load parameter and the use of Ohm's Law:

$$\begin{aligned} \text{Definition of load parameter} \quad K &\equiv \overline{E}_y / (u_c B) \\ \overline{E}_y &= K u_c B \\ E_{yc} &= \overline{E}_y + 2 V_{drop} \end{aligned}$$

from Ohm's Law,

$$J_{yc} = \sigma_c (E_{yc} - u_c B) \text{ for } J_x = 0.$$

From the expressions for V_{drop} and E_{yc} , the following equation can be written for J_{yc} when $J_{yc} < 0$,

$$J_{yc} = \frac{\sigma_c u_c B(K - 1 + (V_{\text{arc}} + V_{\text{fd}})/u_c B\ell)}{1 + r\sigma_c/\ell}$$

where

$$V_{\text{fd}} = u_c B \delta_1 / \delta \text{ (fluid dynamic voltage drop)}$$

ℓ = distance from electrode wall to center of channel.

from Ohm's Law, $J_x = \sigma E_x - \beta J_y$

which gives, for $J_x = 0$, $E_{xc} = \beta J_{yc} / \sigma_c$.

The energy extracted from the gas in the form of electrical power is calculated by summing the contribution to the power output from three regions: the core region, the electrode wall boundary layer, and the insulating wall boundary layer. In the core region, the core values of J_y and E_y are used to determine the power output. In the electrode boundary layer region, the core value of J_y is used with the spatially averaged value of E_y for the region. In the insulating wall boundary layer, the spatially averaged value of the product of $J_y E_y$ is used.

An Euler predictor-corrector routine with a stepsize of 0.1 m was found to give good accuracy for the solution of the core and boundary layer ordinary differential equations. The variation of some channel parameters with axial distance along the channel is shown in Table E.1 for Base Case I.

Table E.1 Selected MHD Channel Parameters for Base Case I

X (m)	T (K)	P (atm)	σ ($\frac{\text{mhos}}{\text{m}}$)	β	u (m/s)	K	E _{yc} (kV/m)	J _{yc} (A/cm ²)	E _{xc} (kV/m)	B (T)	V _{drop} (V)	St _{elect}	St _{ins}	Δ_{elect} (m)	Δ_{ins} (m)	$\delta_{l,\text{elect}}$ (m)	$\delta_{l,\text{ins}}$ (m)
0	2636	4.12	7.71	1.05	835	0.80	3.4	-0.58	-0.1	5.0	60	.00168	.00171	.0069	.0030	.0012	.0010
2	2582	3.33	6.84	1.53	818	0.77	4.0	-0.73	-1.6	6.0	140	.00155	.00140	.0422	.0270	.0070	.0020
4	2517	2.62	5.78	1.90	796	0.78	4.0	-0.59	-1.9	6.0	210	.00155	.00131	.0826	.0342	.0138	.0006
6	2451	2.07	4.77	2.35	775	0.79	4.0	-0.43	-2.1	6.0	260	.00163	.00134	.0419	.0657	.0218	-.0028
8	2390	1.67	3.93	2.85	757	0.80	3.9	-0.32	-2.3	6.0	290	.00176	.00136	.0353	.0896	.0311	-.0038
10	2334	1.37	3.25	3.40	739	0.80	3.8	-0.24	-2.5	6.0	300	.00202	.00137	.0219	.1164	.0413	-.0059
12	2284	1.15	2.72	3.98	724	0.80	3.7	-0.18	-2.6	6.0	320	.00235	.00138	.0122	.1481	.0525	-.0072
14	2238	0.98	2.29	4.59	710	0.80	3.7	-0.14	-2.8	6.0	350	.00266	.00140	.0077	.1872	.0646	-.0100
16	2198	0.85	1.95	4.56	697	0.80	3.2	-0.10	-2.3	5.3	350	.00272	.00141	.0079	.2336	.0774	-.0131

APPENDIX F

COAL AND ASH ANALYSIS OF ILLINOIS #6 AND MONTANA ROSEBUD COALS

<u>Proximate Analysis, Coal As Received, Percent</u>	<u>Montana Rosebud</u>	<u>Illinois #6</u>
Moisture	22.7	13.0
Volatile Matter	29.4	36.7
Fixed Carbon	39.2	40.7
Ash	8.7	9.6
<u>Ultimate Analysis, Percent</u>		
Hydrogen	6.0	5.9
Carbon	52.1	59.6
Nitrogen	0.79	1.0
Oxygen	31.5	20.0
Sulfur	0.85	3.9
Ash	8.76	9.6
Heating Value, As Rec'd (Btu/lb)	8936	10788
Coal Rank	Subbituminous B	HVCB
<u>Ash Analysis, Percent</u>		
SiO ₂	47.3	47.4
Al ₂ O ₃	21.7	19.7
Fe ₂ O ₃	6.4	21.2
TiO ₂	0.8	0.8
P ₂ O ₅	0.5	0.25
CaO	13.8	7.9
MgO	5.0	0.9
Na ₂ O	3.9	0.2
K ₂ O	0.6	1.7

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APPENDIX G

CHEMICAL EQUILIBRIUM AND TRANSPORT PROPERTIES PROGRAM

The chemical equilibrium subroutine (EQUIL) which was used in this study is a modified version of the NASA-Lewis thermochemical program (40) which is capable of being used on minicomputer systems. This program computes the equilibrium composition and thermodynamic properties of a system by using Newton-Raphson iteration to solve a set of linearized equilibrium and atom balance equations. EQUIL considers gaseous species, both neutral and electrically charged, and condensed phase species. The condensed phase is modeled by a quasi non-ideal solution model in which $\text{CaO}_{(l)}$ and $\text{SiO}_{2(l)}$ are allowed to deviate from ideal behavior while all of the other condensed species are assumed to behave ideally. Non-ideal behavior of $\text{CaO}_{(l)}$ and $\text{SiO}_{2(l)}$ was experimentally observed by Byker, et al. (28), and it was included in the model only to allow an approximate accounting for this behavior, which has an effect on the amount of slag vaporized in the combustor and on the electrical conductivity by influencing the amount of AlO_2^- present. The equilibrium program requires the activity coefficients (defined as the actual partial pressure of a species divided by the partial pressure which the species would have if it behaved ideally in solution) for $\text{CaO}_{(l)}$ and $\text{SiO}_{2(l)}$ to be specified as inputs and does not attempt to model the functional dependence of the activity coefficients on composition or temperature. The values for the activity coefficients of $\text{CaO}_{(l)}$ and $\text{SiO}_{2(l)}$ were set to be 0.005 and 1.0 respectively in the first combustor stage, and 0.01 and 0.5 respectively in the second combustor stage and all points downstream. The species which have been considered by EQUIL in this study are: AlO , AlO_2 , $\text{Al}(\text{OH})_2$, CO , CO_2 , Ca , CaO , CaOH , $\text{Ca}(\text{OH})_2$, Fe , FeO , H , H_2 , H_2O , K , KO , KOH , Mg , MgO , N_2 , NO , NO_2 , O , O_2 , OH , PO , PO_2 , SO_2 , SiO , SiO_2 , AlO_2^- , K^+ , O^- , OH^- , PO_2^- , e^- , $\text{Al}_2\text{O}_3(l)$, $\text{CaO}_{(l)}$, $\text{CaSiO}_3(l)$, $\text{FeO}_{(l)}$, $\text{K}_2\text{SiO}_3(l)$, $\text{K}_2\text{SO}_4(l)$, $\text{K}_2\text{CO}_3(l)$, $\text{MgO}_{(l)}$, $\text{MgSiO}_3(l)$, and $\text{SiO}_{2(l)}$. The thermodynamic

data used in the chemical equilibrium code is in curvefit form. The thermodynamic quantities which have been curvefit are G/RT , H/RT , S/R , C_p/P where G , H , S , C_p , R , and T are respectively the Gibbs free energy, enthalpy, entropy, specific heat, gas constant, and temperature. Except for the species of $Al(OH)_2$, $CaSiO_3(l)$, PO_2 , and PO_2 , all of the curvefits are based on JANAF table data.⁽¹⁰⁾ Data for $Al(OH)_2$ were taken from reference 41. Data for $CaSiO_3(l)$ were taken from a chemical equilibrium code developed at Argonne National Laboratory by Blackburn.⁽⁴²⁾ The heat of formation of PO_2 was taken from measurements by Drowart, et al.⁽⁴³⁾ and the electron affinity for PO_2 of 2.8 eV is based on an analysis of experimental results by Annen, et al.⁽⁹⁾

The determination of the electrical conductivity and electron mobility is based on the Frost mixture rule,⁽⁴⁴⁾ using the calculational procedure of Mitchner.⁽⁴⁵⁾ Energy dependent momentum-transfer cross-section data used to calculate the electron-neutral collision frequencies are taken from Spencer and Phelps⁽⁴⁶⁾ with revised data for potassium atoms taken from reference 47 and listed in Table G.1 for convenience. The cross-section data for KOH was also revised by reducing the values of Spencer and Phelps by a factor of two. This revision was based on recent calculations by Collins, et al.⁽⁴⁸⁾ Besides the electron conductivity and mobility, the other transport properties which were calculated were viscosity and thermal conductivity. The viscosity is calculated using the semiempirical mixture rule of Wilke⁽⁴⁹⁾ which also appears in reference 50. Data for the calculation of the viscosity was taken from reference 51. The thermal conductivity was determined from the viscosity by using the semiempirical Eucken relation:⁽⁵²⁾⁽⁵³⁾

$$k = (C_p + \frac{5}{4} R)\mu$$

where

k = thermal conductivity

C_p = specific heat

R = specific gas constant

μ = viscosity

A more detailed presentation of the transport property calculations is given by Pepper.⁽⁶⁾

Table G.1

Revised Cross Sections (from ref. 47)

Electron Energy (eV)	Electron Momentum-Transfer Collision Cross section for potassium (10^{-20} m^2)
0.0	708
0.001	465
0.002	341
0.003	267
0.005	194
0.007	202
0.0085	288
0.010	373
0.015	1318
0.02	3149
0.03	3653
0.04	2345
0.05	1827
0.07	1169
0.10	751
0.12	683
0.15	582
0.17	513
0.20	412
0.25	351
0.30	290
0.35	256
0.40	223
0.50	180
0.70	147
1.0	95
1.2	82.3
1.3	77.1
1.5	66.8
1.7	51.0
1.9	67.0
2.1	66
2.2	63
2.5	54
2.8	47
3.0	43
3.3	39.5
3.6	36.5
4.0	33.5
4.5	32
5.0	31
6.0	30
7.0	30
8.0	30
10.0	30
12.0	30
15.0	30
17.0	30
20.0	30
25.0	30
30.0	30
50.0	30
75.0	30
100.0	30

APPENDIX H

DIFFUSER DESIGN AND COST MODEL DETAILS

The MHD gas flow in the diffuser was modeled as plug flow with the velocity being determined by the flow area and gas density from the continuity equation,

$$u = \dot{m}_{\text{gas}} / (\rho A)$$

where the expansion angle of the diffuser of $2\theta = 6.2^\circ$ specifies the area variation. As stated in Chapter 4, the variation of pressure with axial distance is specified, being based on the results of an experimental study by Dolan and Runstadler.⁽²⁹⁾ For the plug flow assumption, the energy equation which was used to calculate the gas cooling rate in an axial marching routine was,

$$h_2 = h_1 + \frac{1}{2}(u_1^2 - u_2^2) - (q_{\text{rad}}''' \Delta V + q_{\text{conv}}'' \Delta A) / \dot{m}_{\text{gas}}$$

where

- u_1, h_1 = velocity and static enthalpy at start of step
- u_2, h_2 = velocity and static enthalpy at end of step
- ΔV = diffuser volume covered in step
- ΔA = diffuser wall area covered in step
- $\dot{m}_{\text{gas}}$ = MHD gas flow
- q_{rad}''' = gas cooling rate by radiation, power/volume
- q_{conv}'' = gas cooling rate by convection, power/area

The gas cooling rate by convection was calculated by

$$q_{\text{conv}}'' = St \rho u (h - h_{\text{aw}})$$

where

- h = mean enthalpy of the gas
- h_{aw} = adiabatic wall enthalpy
- ρ = gas density

u = mean velocity

St = Stanton number given by

$$St = (T_w/T_{gas})^{-0.25} 0.0295 Re^{-0.2} Pr^{-0.4}$$

The gas cooling rate by radiation is given by

$$q_{rad}''' = 4/d_h \cdot (\epsilon_w + 1)/2 \cdot \sigma(\epsilon_g T_g^4 - \alpha_g T_w^4) + q_{rK}'''$$

where

d_h = mean hydraulic diameter of diffuser

ϵ_w = emissivity and absorption of the wall

ϵ_g, α_g = emissivity and absorptivity of the gas

σ = Stephan/Boltzman constant $(=5.67 \times 10^{-8} \frac{W}{m^2 K^4})$

T_w = wall temperature

q_{rK}''' = gas cooling rate due to radiation from the potassium line.

In the above expression, the first term accounts for gas cooling by molecular radiation, particularly from CO_2 and H_2O . The emissivities and absorptivities can be obtained from charts, such as those presented by McAdams.⁽³²⁾ Some uncertainty is associated with the value of the second term, q_{rK}''' , due to uncertainty in the absorption coefficient of potassium. This term has been calculated by taking the average of two values, one being the value of q_{rK}''' obtained from a classical Lorentzian model of the potassium line radiation, and the other being the value of q_{rK}''' obtained from a Russian/slope 1 model of the potassium line (see section 3.8 for further details). These two values differ by roughly a factor of 2. The value of q_{rK}''' ranges from 30% of q_{rad}''' at the entrance of the diffuser to 40% of q_{rad}''' at the exit of the diffuser. The wall temperature, T_w , of the diffuser was specified to be 2000 K and the wall emissivity was taken to be $\epsilon_w = 0.6$.

The air cooling design of the conical diffuser consists of compressed air flowing through an annular passage formed by the inner and outer shells of the diffuser. A cross sectional view of diffuser wall is shown in Figure H.1. The diffuser was designed to maintain a constant

value of $G_{air} (\equiv \rho u)$ for the air flowing through the annular passage. Using the continuity equation,

$$\dot{m}_{air} = G_{air} A = \text{constant}$$

the flow passage spacing, r , can then be approximately specified by,

$$r \approx \frac{\dot{m}_{air}}{G_{air} \pi (R + t_{cer} + t_{si})}$$

The thickness of the outer shell and the insulation were set at $t_{so} = 0.5$ in, $t_{ins} = 6$ in. These thicknesses are constant for the total length of the diffuser and were selected on the basis of their maximum service requirements. The thickness of the inner shell, t_{si} , and the thickness of the ceramic liner, t_{cer} , were calculated as a function of the axial location in the diffuser based on the local strength and heat transfer requirements. The thickness of the ceramic liner is specified by the requirement that the heat flux from the MHD gas to the 2000 K wall is matched by the heat flux from the wall to the air coolant. This is determined in the axial marching routine by first calculating the average temperature of the air coolant, T_{av} , for the axial step in the diffuser:

$$h_x + \delta x = h_x + q'' \Delta A / \dot{m}_{air} + \frac{1}{2} (u_x^2 - u_{x+\delta x}^2)$$

$$T_{x+\delta x} = f(h_{x+\delta x}) \quad (\text{equation of state})$$

$$T_{av} = \frac{1}{2} (T_x + T_{x+\delta x})$$

The average Nusselt number, based on the analytical solution for fully developed turbulent flow between parallel planes with one side heated and one side insulated, ⁽⁵⁴⁾ is then calculated for the axial step. From the Nusselt number, the convective heat transfer coefficient, h_{conv} , is calculated, based on the hydraulic diameter of the annular passage. The temperature of the outer surface of the ceramic liner, T_c , which is adjacent to the inner metal shell, is given by,

$$T_c = T_{av} + q'' / U_{av}$$

From the thermal circuit diagram shown in Figure H.2, U_{av} can be seen to be:

$$U_{av} = (1/h_{conv} + R_{contact})^{-1}$$

where $R_{contact}$ is the contact resistance between the ceramic liner and the inner shell. The thermal resistance of the shell was assumed to be negligible and a value for the contact resistance of $R_{contact} = 0.002 \frac{m^2K}{W}$ was used. Knowing T_c , the ceramic liner thickness, t_{cer} , can be found from,

$$t_{cer} = (T_w - T_c) k_{cer} / q''$$

where the thermal conductivity of the ceramic, k_{cer} , is evaluated at the mean of T_c and T_w .

The thickness of the inner shell is a function of the temperature of the shell since the design strength of the metal is temperature dependent. The temperature of the inner shell, T_{si} , is found from the thermal circuit to be,

$$T_{si} = T_{av} + q'' / h_{conv}.$$

The minimum allowable thickness of the shell is determined from the ASME Power Piping Code, Paragraph 104.1.2:

$$t_{min} = \frac{PD}{2(S_D + yP)}$$

where

P = pressure difference between air and MHD gas, psi

D = equivalent diameter of support brace spacing; in

S_D = design strength of shell material, psi

y = 0.7 for $T_{si} > 1150^\circ F$, 0.4 for $T_{si} < 900^\circ F$

$y = 0.4 + 0.3(T_{si} - 900)/250$ for $900^\circ F < T_{si} < 1150^\circ F$

t_{min} = minimum shell thickness, in .

The actual thickness of the shell, t_{si} , is sized up to the next standard plate thickness. In the above equation, the value of D was set to 10 in. This is the diameter of a pipe which has the same stress in the pipe wall as the maximum stress in the inner shell.

The pressure drop of the air flow in the radiant heater is calculated by using the friction factor correlation for turbulent flow in tubes, which also works well for non-circular geometries when the Reynolds number is based on the hydraulic diameter,

$$f = 0.046 \text{ Re}^{-0.2} (T_{si}/T_{av})^{-0.10} .$$

The last term in this expression is a temperature correction factor to account for temperature dependent properties.

In the cost model for the diffuser, the cost was calculated by summing the contribution from seven items:

$$\begin{aligned} \text{Diffuser Cost} = & C_{\text{shell},i} + C_{\text{shell},o} + C_{\text{labor}} + C_{\text{ind labor}} + \\ & C_{\text{ceramic}} + C_{\text{insulation}} + C_{\text{struct}} \end{aligned}$$

where

$$\begin{aligned} C_{\text{shell},i}, C_{\text{shell},o} &= \text{Cost of inner and outer shells} \\ C_{\text{labor}} &= \text{Cost of direct labor} \\ C_{\text{ind labor}} &= \text{Cost of indirect labor } (=0.9 C_{\text{labor}}) \\ C_{\text{ceramic}} &= \text{Cost of ceramic liner} \\ C_{\text{insulation}} &= \text{Cost of insulation} \\ C_{\text{struct}} &= \text{Cost of structural steel} \end{aligned}$$

All of the above costs were calculated on a cost per unit weight basis using the cost figures in Table 4.1. The labor cost was specified to be equal to 100% of the cost of materials. The weight of the structural steel was assumed to be 30% of the weight of the diffuser.

APPENDIX I

GAS TURBINE PLANT HEAT EXCHANGER DESIGN AND COST MODEL DETAILS

The design procedure used in the model for the gas turbine plant heat exchangers is given below. This design procedure is followed by a description of the cost model.

Step 1: The following quantities are specified for the heat exchanger design:

$T_{h,o}, T_{h,i}, T_{c,o}, T_{c,i}, d_o, d_h, a_{cor}, \sigma_h, \alpha_h, \alpha_c, \gamma, R_{foul,h},$

$R_{foul,c}, C_{min}, C_{min}/C_{max}, \dot{m}_c, \dot{m}_h, \epsilon, \Delta P/P_c, \Delta P/P_h, N$

where subscripts c,h denote cold and hot side
subscripts i,o denote inlet and outlet

d = tube diameter

d_h = hydraulic diameter for tubes in crossflow

a_{cor} = corrosion allowance for tube wall thickness

σ = ratio of free-flow area to frontal area

α = ratio of total transfer area of one side of exchanger to total exchanger volume

γ = ratio of fin area to total heat transfer area for one side (for air/air heat exchangers, hot side only)

R_{foul} = resistance due to surface fouling

C_{min} = minimum capacity rate ($C = \dot{c} \dot{m}$)

C_{mix}/C_{max} = capacity rate ratio

$\dot{m}$ = mass flow

ϵ = heat exchanger effectiveness

$\Delta P/P$ = relative pressure drop

N = number of heat exchanger passes

- The average hot and cold side temperatures are calculated, and average values for the density (ρ), viscosity (μ), Prandtl number (Pr), and specific heat (c_p) are evaluated for each side.
- Initial guesses for the flow-stream mass velocities, G , for the hot and cold sides are made.
- The effectiveness of the heat exchanger passes, ϵ_p , is determined iteratively from the following relation: (55)

$$\epsilon = \frac{\left(\frac{1 - \epsilon_p C_{\min}/C_{\max}}{1 - \epsilon_p} \right)^N - 1}{\left(\frac{1 - \epsilon_p C_{\min}/C_{\max}}{1 - \epsilon_p} \right)^N - \frac{C_{\min}}{C_{\max}}}$$

for the case where $C_{\min}/C_{\max} = 1$,

$$\epsilon = \frac{N \epsilon_p}{1 + (N-1) \epsilon_p}$$

- The number of heat transfer units for the heat exchanger is determined as a function of $C_{\min}/C_{\max}$ and ϵ from the results of an analytical solution for a crossflow heat exchanger with both fluids unmixed. (56)
- The maximum tube wall temperature is estimated by the approximation,

$$T_{w,\max} = T_{c,o} + 0.4(T_{h,i} - T_{c,o}).$$

- For the maximum outlet temperature, the minimum allowable tube wall thickness is determined, a corrosion allowance added (for MHD gas/air heat exchangers only), and the result is rounded up to the next standard wall thickness. An inner tube diameter, d_i , is then calculated.

Step 2: - The following quantities are calculated:

$$Re_c, Re_h, St_c, St_h, f_c, f_h, h_c, h_h$$

where $Re =$ Reynolds number based on d_i or d_h .
 $St =$ Stanton number
 $f =$ friction factor
 $h =$ convective heat transfer coefficient.

- The average overall conductance for heat transfer, based on the cold side surface area, for the bare tube MHD gas/air heat exchangers is,

$$U_{av} = [(1/\bar{h}_h + R_{foul,h}) d_i/d_o + 1/\bar{h}_c + R_{foul,c}]^{-1}.$$

The average overall conductance for the air/air heat exchangers is a function of the fin efficiency. For the fin geometry chosen, the fin efficiency is approximately given by:

$$\eta_f = 0.94 \frac{\tanh(m\ell)}{m\ell}$$

where

$$m = \sqrt{\frac{2h}{k\delta}}$$

$\ell =$ width of fin

$h =$ convective heat transfer coefficient

$k =$ thermal conductivity

$\delta =$ fin thickness

The overall surface efficiency is given by $\eta_o = 1 - \gamma(1 - \eta_f)$.

$$U_{av} = [(\frac{1}{\eta_o \bar{h}_h} + R_{foul,h}) \frac{\alpha_c}{\alpha_h} + \frac{1}{\bar{h}_c} + R_{foul,c}]^{-1}$$

- The maximum tube wall temperature occurs at the cold side outlet and is given by:

$$T_{w,max} = T_{c,o} + (T_{h,i} - T_{c,o}) U_{av} (1/\bar{h}_c + R_{foul,c})$$

- The minimum allowable tube wall thickness is calculated for $T_{w,max}$, a corrosion allowance added (for MHD gas/air heat exchangers only), and the result is rounded up to the next standard wall thickness. The inner tube diameter is

calculated and then compared to the value which was obtained in Step 1 for the estimated maximum wall temperature. If the two values differ, Step 2 is repeated with the new value for d_i .

Step 3: - The total cold side and hot side heat transfer surface areas

A_c and A_h , are calculated:

$$\begin{aligned} \text{for MHD gas/air heat exchangers, } A_c &= N_{tu} C_{min}/U_{av} \\ A_h &= (d_o/d_i) A_c \end{aligned}$$

$$\begin{aligned} \text{for air/air heat exchangers, } A_c &= N_{tu} C_{min}/U_{av} \\ A_h &= (\alpha_h/\alpha_c) A_c \end{aligned}$$

- The cold and hot side free-flow areas, $A_{ff,c}$ and $A_{ff,h}$, are calculated, and σ_c is calculated.
- The pressure drops for the cold and hot sides are calculated as functions of G , P , σ , A , A_{ff} , ρ , and f . If the calculated $\Delta P/P$ for either of the sides is not within 0.001 of the specified $\Delta P/P$, a new value of G for the side is estimated and the design procedure from Step 2 is repeated until the difference between the calculated and specified $\Delta P/P$ is less than 0.001.
- For the MHD gas/air heat exchangers, the following quantities are calculated for the hot and cold sides of each pass: μ , Pr , c_p , T_i , T_o . The the average values of Re , St and h are determined and average conductances, U_{av} , calculated for each pass. The maximum wall temperature and the resulting tube wall thickness for each pass are determined.
- The heat exchanger dimensions are calculated as follows:

$$A_{fr,h} = A_{ff,h}/\sigma_h$$

$$A_{fr,c} = A_{ff,c}/\sigma_c$$

$$V_p = (A_c/N)/\alpha_c$$

$$L_{comm} = A_{fr,c} A_{fr,a}/V_p$$

$$L_c = A_{fr,c}/L_{comm}$$

$$L_h = A_{fr,h}/L_{comm}$$

where

- A_{fr} = frontal area of the hot or cold inlet side
- V_p = volume of one pass of the heat exchanger
- L_{comm} = length of the side of one heat exchanger pass which the hot side inlet and cold side inlet share in common.
- L_c = length of the side of one heat exchanger pass which, with L_{comm} , circumscribes the cold side inlet.
- L_h = length of the side of one heat exchanger pass which, with L_{comm} , circumscribes the hot side inlet.

It should be noted that L_{comm} , L_c , and L_h are lengths specifying the dimensions of one heat exchanger pass, and there are N passes in the heat exchanger.

The heat exchanger surface geometry used for the MHD gas/air heat exchangers was that of 1 1/2 in. OD tubes with a longitudinal-tube pitch ratio of 1.25 and a transverse-tube pitch ratio of 2.00. The cold side (flow inside tubes) correlations for St and f were taken from Kays and London,⁽⁵⁷⁾ as were the hot side (tubes in crossflow) correlations.⁽⁵⁸⁾ The heat exchanger surface geometry for the air/air heat exchangers was that of finned circular tubes with a tube OD of 1.024 in. and a transverse tube spacing of 3.08 in. and a longitudinal tube spacing of 2.06 in. The cold side correlations for St and f were the same as those for the MHD gas/air heat exchangers, and the hot side correlations were taken from Kays and London.⁽⁵⁹⁾

The cost model considers nine cost categories for the MHD gas/air heat exchangers and seven for the air/air heat exchangers. The MHD gas/air heat exchangers are considered to be 100% site assembled because of the special design and materials necessitated by the severe service requirements. The air/air heat exchangers, on the other hand, are assumed to be constructed of prefabricated modules. The common cost categories between these two types of heat exchangers are: structural steel (C_{struct}), direct labor (C_{labor}), indirect labor ($C_{ind labor}$), insulation ($C_{insulation}$), inlet and outlet leaders (C_{header}), and site

material (C_{sitml}). In addition, the air/air heat exchangers have the module cost category (C_{module}), while the MHD gas/air heat exchangers have categories for the tube cost (C_{tube}), tube sheet cost (C_{tbsht}), and the steel enclosure cost (C_{shell}).

The cost of the structural support steel is based on the assumption that the weight of the structural steel is equal to 30% of the total weight of the heat exchanger. This is the same estimation rule used in the EPRI/W study.⁽³⁾

The direct labor cost for the MHD gas/air heat exchangers is the sum of the labor cost for site erection of the heat exchanger and the labor cost for tube-and-sheet welding and nozzle welding. Since these units are completely site assembled, the site erection labor charge is set to be 100% of the direct material costs ($C_{struct} + C_{insulation} + C_{header} + C_{tube} + C_{tbsht} + C_{shell}$). For this cost, however, the value of C_{tube} is based on the cost of Croloy 304 H material and not on the actual value of C_{tube} if a more expensive material was used. The cost of welding is based on a unit cost of \$12 per linear inch for tube-and-sheet welding and a cost of \$110 per nozzle for welding. The heat exchangers were assumed to be constructed with multiple gas side passes, with no tube-and-sheet welding between passes unless the tube material or wall thickness is changed between passes. Tube-and-sheet welding and nozzle welding is required for each tube at the air inlet and outlet. The tube material or wall thickness was changed between passes only if the reduction in material cost exceeded the total increased labor (direct + indirect) cost due to additional tube and sheet welding.

The direct labor cost for the air/air heat exchangers is only for site erection of the heat exchanger. This is set to be the sum of 40% of the materials cost of the prefabricated module and 100% of $C_{insulation} + C_{header} + C_{struct}$.

The indirect labor cost for all heat exchangers is set to be 90% of the direct labor cost. This estimation rule for the indirect labor cost, as well as the rules used in the direct labor cost calculation, are the same as those used in the EPRI/W study.⁽³⁾

The cost of insulation for the heat exchanger is based on the surface area of the enclosure, with the cost per unit area being the same as that

for gas manifold insulation for the MHD gas/air heat exchangers, and the same as that for air manifold insulation for the air/air heat exchangers.

The cost of the headers is based on a maximum design velocity of 25 m/s and a length equal to L_{comm} . The steel used for the headers is Croloy 2 1/4, with exterior insulation if the fluid temperature is less than 900°F and interior insulation used if the temperature is greater than 900°F.

The cost of site material is set to be 5% of the total heat exchanger material cost, the same as in the EPRI/W study.⁽³⁾

The cost of modules for the air/air heat exchangers is based on cost data presented in Reference 60. The cost figure used in the present study is \$18.15/ft² (mid-1978 \$) based on the bare tube area.

The cost of tubes, tube sheets, and the steel enclosure for the MHD gas/air heat exchangers is calculated on a cost per unit weight basis using the data in Table 4.1 and the results of the heat exchanger design routine. The tube sheet material is specified to be the same as the tube material. The steel enclosure is constructed from Croloy 1 1/4.

The total heat exchanger cost is then given by:

for MHD gas/air heat exchangers

$$\text{Cost of Heat Exchanger} = C_{\text{struct}} + C_{\text{labor}} + C_{\text{ind lab}} + C_{\text{insulation}} \\ + C_{\text{header}} + C_{\text{sitml}} + C_{\text{tube}} + C_{\text{tbsht}} + C_{\text{shell}}$$

and for air/air heat exchangers

$$\text{Cost of Heat Exchanger} = C_{\text{struct}} + C_{\text{labor}} + C_{\text{ind lab}} + C_{\text{insulation}} \\ + C_{\text{header}} + C_{\text{module}} + C_{\text{sitml}}$$

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REFERENCES

1. "Economic Studies of Coal Gasification Combined Cycle Systems for Electric Power Generation," Final Report EPRI AF-642, Fluor Engineers and Constructors, Inc., January 1978.
2. "Evaluation of Phase 2 Conceptual Design and Implementation Assessment Resulting from the Energy Conversion Alternatives Study (ECAS)," NASA TM X-73515, April 1977.
3. "Open-Cycle MHD Systems Analysis," Final Report EPRI AP-1316, Westinghouse Electric Corporation, February 1980.
4. "Parametric Study of Potential Early Commercial MHD Power Plants (PSPEC)," Avco Everett Research Laboratory, NASA CR-159633, December 1979.
5. "Parametric Study of Potential Early Commercial MHD Power Plants (PSPEC)," General Electric Space Division, NASA CR-159634, February 1980.
6. Pepper, J. W., "Effect of Nitric Oxide Control on MHD-Steam Power Plant Economics and Performance," SU-IPR Report No. 614, Stanford University, December 1974.
7. "MHD Air Heater Development Technology," Progress Report for the Period Nov. 26, 1979 March 31, 1980, Fluidyne Engineering Corp., May 1980.
8. Lahoda, E. J. and Lippert, T. E., "Engineering Design for the Westinghouse MHD Seed Regeneration Process," Seventh International Conference on MHD Electrical Power Generation, Cambridge, MA, June 1980.
9. Annen, K. D., Kuzmenko, P. J., Keating, R., and Self, S. A., "Electron and Positive Ion Measurements in MHD Combustion Plasmas with Phosphorous Addition," Journal of Energy, Vol. 5, No. 1, 1981.
10. JANAF Thermochemical Tables, NSRDS-NBS 37, June 1971 and later revisions.
11. "Proceedings of the Chemical Equilibrium and Transport Properties Workshop," Division of Magnetohydrodynamics, U.S. Department of Energy, Germantown, MD, September 13-14, 1979, pp. 33-35.
12. "Technical Assessment Guide," Electric Power Research Institute, EPRI PS 1201-SR, July 1979.
13. London, A. L., Class notes from course on Engineering Thermodynamics, Chapter V.

14. Ludwig, C. B., et. al., "Handbook of Infrared Radiation from Combustion Gases," NASA SP-3080, 1973.
15. Vasil'eva, I. A., Deputatova, L. V. Nefedov, A. P., "Investigation of the Far Wings of Resonance Lines of Alkaline Metals in Combustion-Products Plasma," Second U.S.-U.S.S.R. Colloquium on Magnetohydrodynamic Electrical Power Generation, Washington, D. C., June 1975.
16. Im, K. H., and Chung, P.M., "Nucleation and Evolution of Slag Droplets in Coal Combustion," 17th Symposium on the Engineering Aspects of Magnetohydrodynamics, Stanford University, March 1978.
17. McGregor, W. K., "On the Radiation from Small Particles," J. Quant. Spectrosc. Radiat. Transfer, Vol. 19, No. 6, 1978.
18. Konopka, W., Reed, R. A., Calia, V. S., and Oman, R. A., "Controlled Hot Gas/Particle Experiments for Validation of a Standardized Infrared Radiation Model," Final Report AFRPL-TR-81-44, Grumman Aerospace Corporation, July 1981.
19. Ahluwalia, R. K., and Im, K. H., "Heat Transfer Including Particles and Gas Radiation in Subsonic MHD Diffusers-II," AIAA Paper 80-0252, Pasadena, California, January 1980.
20. Hals, F. A., et. al., "Progress in Development of Auxiliary MHD Power Plant Components at Avco-Everett Research Laboratory, Inc.," ASME Paper No. 74-WA/Ener 6, 1974.
21. Monat, J. P., "Experimental Study of Nitric Oxide Formation Kinetics," Ph.D. Thesis, Dept. of Mechanical Engineering, Stanford University, 1977.
22. Hanson, R. K., Flower, W. L., and Kruger, C. H., "Determination of the Rate Constant for the Reaction $O + NO \rightarrow N + O_2$," Combustion Science and Technology, Vol. 9, p. 79-86, 1974.
23. McCullough, R. W., Kruger, C. H., and Hanson, R. K., "A Flow Tube Reactor Study of Thermal Decomposition Rates of Nitric Oxide," Combustion Science and Technology, Vol. 15, p. 213-223, 1977.
24. Berry, G. F., and Dennis, C. B., "Performance Analysis of the MHD-Steam Combined Cycle, Including the Influence of Cost," Argonne National Laboratory, ANL/MHD-80-3, August 1980.
25. Bickle, E., Stellite Division, Cabot Corporation, Private Communication, September 12, 1980.
26. Marksberry, C. L., Fluidyne Engineering Corporation, Private Communication, June 4, 1980.

27. "CDIF-MHD Generator System Conceptual Design," Final Report, TID-27151, MEPPSCO, Incorporated, March 1977, p. 3-4.
28. Byker H. J., et al., "Activity Coefficients in Seeded Coal Slag and Their Effects on Vaporization and Crystallization Equilibria," Seventh International Conference on MHD Electrical Power Generation, Cambridge, Massachusetts, June 1980.
29. Dolan, F. X., and Rustadler, P. W., Jr., "Pressure Recovery Performance of Conical Diffusers at High Subsonic Mach Numbers," Creare Incorporated, NASA CR-2299, August 1973.
30. Kays, W. M., and London, A. L., Compact Heat Exchangers, 2nd ed., McGraw-Hill, New York, 1964.
31. "Comparative Study and Evaluation of Advanced Cycle Systems," Final Report EPRI AF-664, Vol. I, General Electric Company, February 1978.
32. McAdams, W. H. (ed.), Heat Transmission, 3rd ed., Chapt. 4, McGraw-Hill, New York, 1954.
33. James, R. K., "Joule Heating Effects in the Electrode Wall Boundary Layer of MHD Generators," HTGL Report No. 115, Stanford University, January 1980.
34. Ibid, pp. 22-23.
35. Ibid, pp. 9-11.
36. Ahluwalia, R. K., and Doss, E. D., "Convective Heat Transfer in MHD Channels and its Influence on Channel Performance," Paper No. AIAA-80-0178, AIAA 18th Aerospace Sciences Meeting, January 1980, p. 3.
37. Kays, W. M., Convective Heat and Mass Transfer, 1st ed. McGraw-Hill, New York, 1966, p. 242.
38. Rankin, R. R., "Insulating Wall Boundary Layer in a Faraday MHD Generator," FE-2341-7, Stanford University, April 1978.
39. Hokenson, G. J., "On Wall Friction in MHD Channel Flows," AIAA Journal, Vol. 15, September 1977, pp. 1350-1353.
40. Gordon, S., Zelenzik, F. J., and Huff, "A General Method for Automatic Computation of Equilibrium Compositions and Theoretical Rocket Performance of Propellants," NASA TN D-132, October 1959.
41. "Research in Coal-Based Magnetohydrodynamics," Monthly Technical Progress Report, Pittsburgh Energy Research Center, September 1976, pp. 13-15.

42. Blackburn, P. E., "GASCON and MHDGAS: Fortran IV Computer Codes for Calculating Gas and Condensed Phase Compositions in the Coal-Fired Open-Cycle MHD System," ANL/MHD-77-4, Argonne National Laboratory, December 1977.
43. Drowart, J., Myers, C. E., Szwarc, R., Vander Auwera-Mahieu, A., and Vy, O.M., Journal Chemical Society Faraday Transactions, II, Vol. 68, 1972, p. 1749.
44. Frost, L. S., "Conductivity of Seeded Atmospheric Pressure Plasmas," Journal of Applied Physics, Vol. 32, October 1961, pp. 2029-2036.
45. Mitchner, M., "A Computer Program for the Calculation of the Electron Transport Coefficients," SU-IPR Report No. 280, Stanford University, August 1969.
46. Spencer, F. E., Jr., and Phelps, A. V., "Momentum Transfer Cross-Sections and Conductivity Integrals for Gases of MHD Interest," 15th Symposium on the Engineering Aspects of MHD, Philadelphia, May 1976.
47. "Research in Coal-Based Magnetohydrodynamics," Monthly Technical Progress Report, Pittsburgh Energy Research Center, July 1976, p. 10.
48. Collins, L. A., Norcross, D. W., and Schmid, G. B., "Electron Collisions With Highly Polar Molecules: Integrated and Momentum Transfer Cross Sections and Conductivity Integrals for KOH and CsOH," J. Phys. B: Atom. Molec. Phys., Vol. 12, No. 6, 1979, pp. 1019-1030.
49. Wilke, C. R., J. Chem. Phys., Vol. 18, 1950, pp. 517-519.
50. Bird, R. B., Stewart, W. E., and Lightfoot, E. N., Transport Phenomena, 1st ed., John Wiley and Sons, Inc., New York, 1960, p. 24.
51. Hirschfelder, J. O., Curtis, C. F., and Bird, R. B., Molecular Theory of Gases and Liquids, John Wiley and Sons, Inc., New York, 1967.
52. Eucken, A., Physik. Z., Vol. 14, 1913, pp. 324-332.
53. Bird, R. B., Stewart, W. E., and Lightfoot, E. N., Transport Phenomena, p. 257.
54. Kays, W. M., Convective Heat and Mass Transfer, p. 180.
55. Kays, W. M., and London, A. L., Compact Heat Exchangers, p. 20.
56. Ibid, p. 39.

57. Ibid, p. 123.
58. Ibid, p. 190.
59. Ibid, p. 223.
60. Milora, S. L., and Tester, J. W., Geothermal Energy as a Source of Electric Power, The MIT Press, Cambridge, Massachusetts, 1976, p. 135.