FOSSIL
ENERGY

ENGINEERING TEST FACILITY CONCEPTUAL DESIGN

Final Report, Part 1

June 1978

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Avco Everett Research Laboratory, Inc.
Everett, Massachusetts

MASTER

U. S. DEPARTMENT OF ENERGY

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Part 1

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1.0 INTRODUCTION AND SUMMARY

1.1 SCOPE

This document is the final document reporting material and information developed in the ETF Conceptual Design Activity of Department of Energy (DOE) Contract No. EF-77-C-01-2614.

Reference is made to two previous documents submitted reporting results from work developed under this contract. The first document was our preliminary ETF Design Criteria Document dated April 1977. The second document was the ETF Conceptual Design Document, Preliminary Draft, Volumes I and II, dated December 1977. This document is based upon information presented in the two previous documents submitted which represent additional sources of information of the total design effort performed under this contract.

This document reports on the specific ETF reference system design recommended in our previous preliminary ETF conceptual Design Document of December 1977, and referred to in DOE's letters of December 13, 1977 and March 7, 1978.

1.2 OVERALL OBJECTIVE

The major design goals of the ETF are:

- To provide for demonstration of long duration operation of a complete combined MHD/steam power system of pilot scale with coal as the primary fuel source.
- To provide for testing and evaluation of the overall system, subsystems and components performance, controls and interactions, and to identify and resolve critical problem areas.
- To provide design information and the necessary engineering base and confidence for subsequent commercialization of open cycle MHD power generation.

1.3 BASIS FOR DESIGN

1.3.1 Design Criteria and Requirements

The ETF conceptual designs are based upon the initial design guidelines contained in Attachment 1 -- Design Guidelines of Statement of Work of Contract EF-77-C-01-2614, design criteria and requirements contained in our Design Criteria Document, dated April 14, 1977, and subsequent additional ETF Project Groundrules and Directions contained in ERDA's letters of June 14 and Dec. 13 of 1977 and Jan. 25 and March 7 of 1978. The above mentioned documents and letters are the reference sources for detailed descriptions of the criteria, requirements and groundrules for the ETF conceptual designs. Basic criteria, groundrules and requirements which are considered of overall significance and which have major impact on the ETF conceptual designs are summarized below. These include important major groundrule additions specified in ERDA's above mentioned letter of June 14, 1977, and design criteria modifications incorporated to comply with these additional groundrules.

1. Project schedule is to initiate ETF operation by the 1985-1987 time period. A successful ETF project is assumed to be followed by commercial MHD power generation without an intermediate step involving a demonstration plant as in coal gasification, etc.
2. The ETF is a complete experimental MHD/steam power plant facility with a separately fired high temperature air preheater system and a seed recovery/ regeneration system, capable of supplying power to an electric utility grid from the prime MHD cycle and the steam bottoming cycle.
3. Montana subbituminous coal is the primary fuel and base load is considered the reference commercial duty operation and application. Basic load capacity factors, availabilities, forced outage rates, etc. shall be considered.
4. Off-the-shelf equipment shall be utilized as much as possible, and all equipment shall be designed to meet expected operating lifetime requirements of larger scale commercial power plant equipment. Definition of acceptable operating lifetimes and maintenance of equipment are important design criteria items.

5. The ETF shall be capable of addressing, demonstrating and resolving all technological issues of a complete MHD/steam power system involving:
 - a. Satisfactory long duration MHD generator channel performance and life with electrical characteristics, fluid dynamic and slagging conditions appropriate to projected coal burning central station MHD power plant operation.
 - b. Superconducting magnet system operation at the high magnetic field strengths projected for commercial MHD power generation.
 - c. Satisfactory coal combustor performance including seed injection, slag handling and removal for the operating conditions required by the MHD generator.
 - d. Satisfactory high temperature preheater performance and life for the operating conditions required by MHD combustion systems.
 - e. Seed recovery and processing with recycle and feeding of seed practical for economic commercial use.
 - f. DC/AC inversion of MHD generator electrical output and supply of power both from the MHD generator and the bottoming plant steam turbine generator to electric utility networks.
 - g. Practical bottoming plant operation with satisfactory life of boiler and air heater tube materials.
 - h. Satisfactory emission control of all pollutants to comply with all environmental regulations and standards.
 - i. Practical coal processing, drying and feeding.
 - j. Satisfactory control characteristics for full and part load operation of the overall power system and its system components including load change, startup, shutdown and safety aspects. ("Black Start" capability is not a requirement.)
6. The design shall be based on the state-of-the-art in technology and at the same time fulfill all requirements for subsequent commercialization of MHD.

7. Provision for flexibility in testing and operation and for possible future facility modifications.
8. The overall design philosophy adopted is to reduce technical uncertainties at the expense of optimum performance and to minimize total risks and costs.

1.3.2 Potential MHD Power Cycles and Preliminary Design Parameters for Early Commercial Power Plant(s)

The basic ETF power cycle and its major design parameters are selected on the basis of present technical feasibility together with consideration of the technological issues which must be addressed and resolved before commercialization of MHD can be initiated.

Numerous technical-economic analyses and design studies of conceptual commercial MHD power systems have been conducted in the past. Such studies in this country have concentrated on coal burning central station MHD power generation. The more recent and well known ECAS study considered a mature coal burning MHD power system for base load application. All of these studies have provided valuable information for assessment and evaluation of the future potential of open cycle MHD power generation for electric utility utilization as well as for direction of MHD development efforts. However, there is a need to define a particular reference design of a commercial coal burning combined MHD/steam power system which can be considered representative of the particular technology projected for the first projected commercial MHD power plant(s). This is important because the ETF shall provide the necessary basis for such first plant(s) technology. The MHD Baseline Power Plant Design Study performed on ERDA contracts E(49-18)-2228 and 2243 is stated to intend to reflect present or reasonable projected technology as of the date of issue in 1976. There are alternate design parameters and technological solutions that can be considered other than those used in the above mentioned Baseline Reference Design. Such alternate designs could be closer to the present state-of-the-art and experience in MHD technology development than those considered in that and other earlier conceptual designs. Examples of such alternates include separately fired high temperature regenerative heaters, combustion with oxygen enriched air, MHD channel cooling by low temperature and low pressure feed water, as well as selection of other MHD generator design parameters including those for the superconducting magnet and diffuser.

Thus, to complement previous conceptual design studies of commercial MHD power plants, system engineering analyses have been conducted, specifically aimed at developing information for definition of reference design for "first" commercial MHD power plant(s). It is mentioned that these system analyses represent only a smaller effort of the ongoing ETF conceptual design activity. Because of this, design optimization has not been attempted in these initial analyses and the results are of preliminary nature. Rather the system analyses conducted here have concentrated on defining a reference design concept which can be considered practical and viable for first commercial MHD power plant application based on present experience and state-of-the-art in technology. Thus the analyses are valuable for projecting "first" commercial MHD power plant systems and components major reference design parameters as well as their specific operational and performance requirements and characteristics. MHD like any other new technology is expected to improve significantly and mature after its commercial introduction. Although the development of a "first" MHD power plant reference baseline design now is important to define ETF basic design requirements, it should be continuously updated to reflect developments occurring in MHD technology.

1. 3. 2. 1 First Commercial MHD power Plant Design Analysis

Several conceptual commercial MHD power cycles have been proposed. They can broadly be distinguished by either the use of preheated air in combustion of the fuel which requires very high preheat temperatures (about 2000^oF and higher) or by the use of oxygen for enrichment of the combustion air. The latter circumvents the stringent requirement for a very high preheat temperature. High temperature preheaters can again be divided into two main categories, directly fired (preheating by MHD generator exhaust gas) and indirectly fired (preheating by burning of a separate fuel).

The high temperature air preheater represents a critical component of an MHD power system. It determines the performance of the MHD generator and establishes the thermal efficiency which can be attained by the overall power system. The operating conditions and design parameters for the bottoming plant are also substantially different for a power system with indirectly fired preheaters using a separate fuel as compared to a

power system with directly fired preheaters. Most design studies in the past have concentrated on the use of directly fired air preheaters, which have the potential of reaching the highest thermal efficiency and best performance of any MHD power system considered.

However, the technical feasibility of high temperature directly fired air preheaters remains yet to be demonstrated and its application in "first" commercial power plants is presently considered very much uncertain and not likely. Instead we have in our system analysis explored the possible use of indirectly fired high temperature regenerative preheaters for which a relatively good technology base already exists. As an alternate, the use of oxygen enrichment of the combustion air along with more conventional metal tubular type recuperators is considered.

In case of the use of indirectly fired air preheaters, the clean fuel required for separate firing of the preheaters is assumed to be a low Btu gas produced from coal in a gasification process. This involves additional losses. Production of the low Btu fuel gas required, by partial gasification of the coal fuel through chemical regeneration of the MHD generator exhaust gas has the potential of reaching the same high cycle efficiencies with the use of separately (indirectly) fired high temperature preheaters as those attainable with directly fired preheaters. A power cycle based on the concept of chemical regeneration is shown on Figure 1-1. However, the technical feasibility of chemical regeneration has not yet been demonstrated. Therefore, this concept is presently not considered available for first commercial power plant, but its potential is important to recognize. The use of a low Btu gasifier for production of a relatively clean fuel gas for separate firing of the preheater system results in an all coal burning power system design concept. This has been selected for investigation as one reference design for first commercial MHD power plant. Investigation of an alternate power plant design concept with the use of oxygen enrichment of the combustion air is included as another design concept for establishing its potential application for base load service.

This latter plant design concept has also important potential use for intermediate load operation. Investigation of this is beyond the scope of this contract.

Figure 1-1 Alternate Combined MHD Steam Power Cycle with Separate Firing of the High Temperature Air Preheater by a Low Btu Fuel Gas Produced from Chemical Regeneration of Part of the MHD Generator Exhaust Gas

Previous analyses have shown that there are economic advantages in integrating the oxygen production with the power plant when base load (or intermediate load) operation are considered. In addition, past analyses have also shown that the use of oxygen enrichment has a limited potential compared to the use of high temperature preheated air, mainly because higher overall plant thermal efficiencies are attainable with the use of high temperature preheated air. Yet the use of oxygen enrichment has been indicated to be sufficiently attractive to merit its further investigation for first MHD power plant application. (Its important potential use for intermediate load operation has also been indicated in previous studies and should be investigated further.)

The investigations of the use of combustion air oxygen enrichment are included both for comparison and for establishing its potential as an alternate design concept for first commercial MHD power plant. It might represent an attractive option and if so it could enable the commercial introduction of MHD at an earlier date than would otherwise be possible if the development of high temperature air heater systems becomes a pacing item in commercial MHD development.

The approach in our MHD generator design analysis for first commercial power plant application is to utilize relatively "conservative" design parameters consistent with present state-of-the-art of MHD technology or as close to this as possible. This pertains to critical MHD generator design parameters such as magnetic field, channel power density and electrical stresses (E_x , J_y), channel cooling and electrical isolation (use of low temperature and low pressure boiler feedwater) diffuser efficiency, etc., all of which effect MHD generator technical feasibility, performance and reliability.

To summarize, our system engineering analysis of the first commercial plant is aimed at defining a reference design that reduces technical risks, but yet is commercially viable. Significant improvements and advancements are expected to occur after the commercial introduction of MHD technology as the technology matures.

The basic power cycles under investigation are shown in simplified schematics in Figures 1-2(a) and (b) and Figure 1-3. Two variations are shown for the cycle with the use of separately fired high temperature air

Figure 1-2(b) Combined MHD Steam Power Cycle with Separate Firing of the High Temperature Air Preheater by Low Btu Fuel Gas Produced from a Gasifier Integrated with the Power Plant. The cycle air to be preheated is in this preheater system arrangement delivered directly to the high temperature air preheater from the cycle air compressor.

Figure 1-3 MHD Steam Power Cycle with Direct Combustion of Coal with Oxygen Enriched Air. The oxygen plant is integrated with the power plant.

preheaters. They differ in the arrangement of the low and intermediate temperature (recuperative) and high temperature (regenerative) sections of the air preheater system. Overall plant thermal efficiencies will essentially be the same for both these preheater system arrangements so that the results of cycle performance analyses presented below for the power system with the use of higher temperature air preheaters pertains to both arrangements.

Basic design parameters for the power cycle with separately fired high temperature air preheaters are shown in Table 1-1.

The major design parameters for the MHD prime cycle (MHD burner and MHD generator/diffuser) are shown in Table 1-2. Note that the MHD burner heat losses have been assumed 4% and 5% for the case with 2500°F and 3000°F air preheat temperature respectively. This is considered in line with selection of "conservative" first plant design parameters. All MHD generator design and performance parameters presented in this chapter are based on the MHD channel calculations presented and discussed in Section 2.2.2. The calculated MHD generator performance data are applicable both to an ideal Faraday and multiterminal diagonal channel configurations. Also note that a diffuser recovery factor of 0.6 has been utilized in the MHD generator performance calculations.

Preliminary analyses were performed on the steam cycle to establish the attainable bottoming cycle steam plant efficiency with the use of low temperature and low pressure feedwater heating of the MHD channel. The steam cycle and its assumed feedwater heating arrangement is schematically shown in Figure 1-4. The preliminary analysis indicated that a steam cycle efficiency ($\eta = \frac{\text{gross steam power}}{\text{total heat absorbed for steam production}}$) of close to 42% is attainable for the steam plant conditions indicated. Consequently, this steam plant efficiency has been used in all preliminary cycle calculations (both for preheated air and oxygen enrichment of the air) in arriving at total gross steam power.

A preliminary energy balance for the MHD power system considered, with air preheat temperatures of 2500°F and 3000°F respectively is shown in Table 1-3.

TABLE 1-1

BASIC DESIGN PARAMETERS FOR MHD POWER CYCLE WITH
SEPARATELY FIRED HIGH TEMPERATURE AIR PREHEATERS
(FIRST COMMERCIAL PLANT)

Application	Baseload
<u>Coal Fuel</u>	
Type	Montana Sub bituminous
Moisture Content Raw	22.7%
Moisture Content Dry (as fired)	5.0%
Thermal Input to MHD Burner (HHV)	1000 MW
<u>Oxidizer</u>	
Type	Air
Preheat Temperature	2500°F/3000°F
<u>Prime MHD Cycle</u>	
MHD Burner Air/Fuel Equivalent Ratio (ϕ)	0.95
Potassium Seeding	1% K (by weight)
MHD Generator Magnetic Field	6 T (peak)/5 T (average)
Seed Regeneration	Modified Formate Process
<u>Steam Bottoming Cycle</u>	
*Steam Conditions	3500 psi/1000°F/1000°F
Second Combustion Stage Air/Fuel Eq. Ratio (ϕ)	1.05
Condenser Pressure	2.3" Hga

*Supercritical assumed as in ECAS and previous baseline studies. Subcritical 2400 psi/1000°F/1000°F might be more appropriate.

TABLE 1-2

PRELIMINARY PERFORMANCE DATA FOR MHD/STEAM
POWER PLANT WITH SEPARATELY FIRED H. T. AIR HEATER

		<u>Case I</u> <u>2500°F Preheat</u>	<u>Case II</u> <u>3000°F Preheat</u>
<u>MHD Coal Combustor</u>			
Combustion Product Flow Rate	kg/s	353	353
Exit Pressure	atm	5.5	6.5
Exit Temperature	°K	2744	2845
Heat Loss	%	4	5
Ash Carry-Over	%	15	15
<u>MHD Generator</u>			
Mach # (inlet)		0.85	0.85
Electrical Load Parameter		0.775	0.79
Max. Transverse Current Density	amps/cm ²	0.7	0.9
Max. Axial Field	kV/m	1.9	1.7
Max. Hall Parameter	$\omega\tau$	3.6	3.9
Power Output (gross)*	MW	288	341
Enthalpy Extraction*	%	20.5	22.6
Isentropic Efficiency*	%	68.5	71.5
Heat Loss	MW	48	49
<u>Channel Dimensions (Gas Side)</u>			
Inlet	m x m	0.95 x 0.95	0.88 x 0.88
Outlet	m x m	2.1 x 2.1	2.12 x 2.12
Length	m	19.5	16.5
Power Density (average)	MW/m ³	6.4	8.7

*Diffuser recovery factor $\frac{\Delta p \text{ (recovered)}}{\Delta p \text{ (actual)}} = 0.6$

STEAM CYCLE
STEAM PLANT EFFICIENCY ~ 42 %

Figure 1-4 Schematic of the Bottoming Steam Cycle with Feed Water Heating Arrangement

TABLE 1-3

PRELIMINARY ENERGY BALANCE FOR MHD/STEAM POWER
PLANT WITH SEPARATELY FIRED H. T. AIR HEATER

		<u>Case I</u>	<u>Case II</u>
		<u>2500°F Preheat</u>	<u>3000°F Preheat</u>
MHD Power	MW	288	341
Steam Power		<u>402</u>	<u>431</u>
Total Gross Power	MW	<u>690</u>	<u>772</u>
Compressor Power*	MW	74	82
Auxiliary Losses**		<u>27</u>	<u>28</u>
Total	MW	<u>101</u>	<u>110</u>
Net AC/Power Output	MW	<u>589</u>	<u>662</u>
*Based on 88% Efficiency and 15% Pressure Loss ($\frac{P_{comp.} - P_{burner}}{P_{burner}}$)			
** Inverter Loss (1%)	MW	2.9	3.4
Transformer Loss (0.75%)	MW	4.6	5.2
Magnet Aux.	MW	0.8	0.8
Coal and Ash Handling	MW	3.5	3.5
Pumps (condensate & circ. water)	MW	3.5	3.5
Seed Feeding	MW	1.0	1.0
Fans (I. D. & Sec. air and H. T. Air Preheater System)	MW	6.5	6.5
Seed Regeneration	MW	2.5	2.5
Misc.	MW	<u>1.5</u>	<u>1.5</u>
		<u>26.8</u>	<u>27.9</u>

TABLE 1-4
LOW Btu GAS PRODUCERS

Type	O ₂ /Air	Prod. HHV	Prod. Temp.	Pressure	Efficiency		Status	
					Cold	Hot		
			°F	Atm	%	%		
Fixed Bed								
Lurgi	x/x	285/196	700	23-28	63	>90	Comm.	* 12' ϕ ~ 800 TPD [13 plants]
Wellman-Galusha	-/x	-/168	1200	1	75	93	Comm.	* 10' ϕ ~ 84 TPD [over 150 plants]
WD/GI	x/x	280/175	1000	1	77		Comm.	* 12' ϕ ~ 100 TPD [over 100 plants]
Fluidized Bed								
Battelle/Carbide	-/x	-	1700	6	-		Bench	---
BCR	-/x	-/160	2000	15	88		PDU	---
CO ₂ Acct.	-/x	-/380	1500	10	77		Pilot	30 TPD ~ 4' ϕ x 70'
U-G ₂₈	-/x	-/154	1700	23	79		Pilot	6 TPD** 22' ϕ x 50' ~ 3000 TPD
Westinghouse	-/x	-/135	1200	14	72	(94)	PDU	PDU 1200 lbs/hr (14 TPD)
Winkler	x/x	288/126	1500	1	72		Comm.	* 18' ϕ x 75' ~ 1100 TPD [16 plants - 3 still in operation]
Entrained Bed								
B&W	x/x	301/102	1800	1-20	74	(95)	Semi Comm.	** 7.5' ϕ x 200' ~ 3000 TPD at 175 psig [duPont - ran 1 400 TPD plant for 1 year]
C-E	-/x	-/127	1600	1	69		Pilot	9' ϕ x 78' ~ 120 TPD
Foster-Wheeler	-/x	-/177	2000	23	-		Pilot Design	480 TPD
K-T	x/-	286/-	2700	1	75		Comm.	* 4 head gasifier $\approx$ 860 TPD [13 plants]
Texaco	x/-	253/-	500	23	-		Comm.*	* Comm. on Liquid Fuel [~70 plants] **9' ϕ x 15' $\approx$ 1900 TPD (650 psig)(coal)

* Commercial Size
** Projected Commercial Size

The amount of thermal heat input with the fuel for separate firing of the high temperature air preheater system is calculated to be 335 MW_{th} for the case with 2500°F preheat temperature and 465 MW_{th} for the case with 3000°F preheat temperature. This corresponds to 25% and 32% respectively of the combined total fuel thermal input to the MHD burner (1000 MW) and to the preheater burners.

The fuel conversion efficiency of the gasification plant which is required to produce the low Btu fuel gas for firing of the high temperature air preheater system must be considered. Several gasification processes can be contemplated. Pertinent data for various low Btu gasifiers commercially available or under development are listed in Table 1-4. With the use of hot low Btu fuel gas (500°F and above) conversion efficiencies in the range of 90 - 95% can be expected. The use of hot fuel gas can here be considered because the necessary sulfur removal can be accomplished by efficient cleaning of the MHD generator exhaust gas. This involves seed processing. The auxiliary power requirement for gas cleaning and seed processing is included in the energy balance in Table 1-3. The additional fuel thermal requirement by the assumed use of a modified formate process described in Section 2.5.1 would be relatively quite small and it is estimated to be 10 MW_{th} or less. The calculated total overall net plant thermal efficiency $\left[\frac{\text{net electrical output}}{\text{coal heat input (HHV)}} \right]$ is shown in Figure 1-5 as a function of the conversion efficiency of the fuel required for separate firing of the high temperature air preheater system. Two efficiency curves are shown pertaining to 2500°F and 3000°F air preheat temperature respectively. The fuel and power requirements for gas cleaning and seed processing are included in the calculated efficiencies.

The basic design parameters assumed for first commercial plant with alternate use of oxygen enrichment of the combustion air are shown in Table 1-5.

Figure 1-5 Net Plant Thermal Efficiency of MHD Steam Power Cycles With Separately Fired High Temperature Air Preheaters As a Function of the Conversion Efficiency of the Fuel Required for Separate Preheater Firing

TABLE 1-5

BASIC DESIGN PARAMETERS FOR MHD POWER CYCLE
WITH COMBUSTION AIR OXYGEN ENRICHMENT
(FIRST COMMERCIAL PLANT)

Application	Base Load
<u>Coal Fuel</u>	
Type	Montana sub bituminous
Moisture Content Raw	22.7%
Moisture Content Dry (as fired)	5.0%
<u>Oxidizer</u>	
Type	Oxygen enriched air
Oxygen Content	30%/34% (by volume)
Preheat Temperature	1100°F/1200°F/1400°F
<u>Prime MHD Cycle</u>	
MHD Burner Air/Fuel Equivalent Ratio (ϕ)	0.95
Potassium Seeding	1% K (by weight)
MHD Generator Magnetic Field	6 T (peak)/5 T (average)
Seed Regeneration	Modified Formate Process
<u>Steam Bottoming Cycle</u>	
* Steam Conditions	3500 psi/1000°F/1000°F
Second Combustion Stage Air/Fuel Eq. Ratio (ϕ)	1.05
Condenser Pressure	2.3" Hga

* Supercritical assumed as in ECAS and previous baseline studies.
Subcritical 2400 psi/1000°F/1000°F might be more appropriate.

Design parameters listed in this table for the coal fuel, prime MHD cycle and steam bottoming cycle are the same as those listed in Table 1-1. for the power cycle with separately fired high temperature air preheaters.

The thermal input to the MHD burner was first considered to be 1000 MW in the design analysis with oxygen enrichment of the combustion air. This was the same as that for the air cycle discussed earlier (MHD burner only). The fuel thermal input was subsequently raised to 1330 MW_{th} and 1450 MW_{th} which corresponds to the combined fuel heat inputs to the MHD burner and high temperature air preheater burners for the air cycle at 2500°F and 3000°F preheat temperature respectively. (The exact corresponding figures for the air cycle is 1335 MW_{th} and 1465 MW_{th}.)

The preheat temperature was first assumed to be 1100°F. Additional analyses were also conducted at 1200°F and 1400°F preheat temperature to establish the effect of preheat temperature on plant performance. This would provide valuable information for later cycle optimization, because previous analyses have shown that the cost of a metal tubular recuperator increases significantly by raising the preheat temperature from 1100°F to 1400°F because this requires increased use of more costly high temperature alloys. Beyond a preheat temperature of 1400°F, metal alloy preheater materials costs are expected to rise relatively sharply. Therefore, this was not given consideration in these preliminary analyses. Two degrees of oxygen enrichment were considered, corresponding to an oxygen enrichment of the combustion air of 30% and 34% by volume respectively. The flame temperatures reached with these two different degrees of oxygen enrichment and a preheat temperature of 1100°F corresponds roughly to those reached with combustion of air and preheat temperatures of 2500°F and 3000°F respectively.

The oxygen plant is assumed integrated with the power plant. Oxygen is considered produced at a purity of 80% (by volume) to minimize the power requirements and costs involved in the manufacturing of oxygen.

Prime MHD cycle performance data with combustion air oxygen enrichment of 30% and 34% are listed in Table 1-6. These data are based on a coal thermal input of 1000 MW_{th} (HHV) and an oxidizer preheat temperature of 1100°F. Note that combustion heat losses of 4.0% and

TABLE 1-6

PRIME MHD CYCLE PERFORMANCE DATA FOR POWER CYCLE
WITH COMBUSTION AIR OXYGEN ENRICHMENT

Basis: Coal Thermal Input (HHV) 1000 MW Oxidizer Preheat Temp. 1100°F

	Case 1 <u>30% O₂</u>	Case 2 <u>34% O₂</u>
<u>MHD Coal Combustor</u>		
Combustion Product Flow Rate (kg/s)	267.5	241.4
Exit Pressure (atm)	5	6
Exit Temperature (°K)	2703	2791
Heat Loss (%)	4.0	4.2
<u>MHD Generator</u>		
Inlet Mach #	0.85	0.85
Electrical Load Parameter	0.75	0.76
Max. Transverse Current Density (amps/cm ²)	0.6	0.73
Max. Axial Field (kV/m)	2.09	1.65
Max. Hall Parameter ($\omega\tau$)	3.79	3.52
Power Output (gross)* (MW)	190	200
Enthalpy Extraction* (%)	17.3	18.6
Isentropic Efficiency* (%)	65	66.0
<u>Channel Dimensions (Gas Side)</u>		
Inlet (m x m)	0.86 x 0.86	0.75 x 0.75
Outlet (m x m)	1.8 x 1.8	1.73 x 1.73
Length m	14	15
Power Density (average) (MW/m ³)	7.38	8.24

*Diffuser recovery factor $\frac{\Delta p \text{ (recovered)}}{\Delta p \text{ (actual)}} = 0.5$

4.2% have been assumed and that the diffuser recover factor is 0.5. Again this is consistent with the consideration of relatively "conservative" cycle design parameters.

A preliminary energy balance for the same two power cycles are listed in Table 1-7. In computation of net plant efficiencies, an additional fuel input of 10 MW has been conservatively estimated for the seed regeneration formate process.

Prime MHD cycle performance data for the higher assumed coal thermal inputs of 1450 MW are shown in Tables 1-8 and 1-9 for preheat temperatures of 1100°F, 1200°F and 1400°F.

Corresponding energy balances for the cases listed in these two tables are shown in Tables 1-10 and 1-11. Tables 1-9 and 1-10 pertain to a combustion air oxygen enrichment of 30% and Tables 1-8 and 1-11 to 34% combustion air oxygen enrichment.

The computed preliminary net plant efficiencies for the power cycles considered with the use of separately fired air preheaters, or alternatively, oxygen enrichment of the combustion air are all significantly higher than the comparable reference conventional steam plant efficiency of 31.8% used by GE in ECAS. (Steam conditions 3500 psi/1000°F/1000°F and Ill. #6 coal fuel were here assumed.) Thus, the computed cycle performance data might very well be attractive for first commercial power plant. It is again pointed out that the basic design parameters utilized in our preliminary "first" commercial plant analysis purposely have been "conservative." Examples of this are assumed combustion heat losses of 4.0 - 5.0%, diffuser recovery factor of 0.5 - 0.6, channel performance with cooling by low pressure feedwater at less than 250°F, etc. Significant improvements can be expected from further development and additional system engineering analysis is proposed and should also, in our view, be continuously conducted to reflect developments occurring in MHD technology.

TABLE 1-7

PRELIMINARY ENERGY BALANCE FOR MHD POWER CYCLE
WITH COMBUSTION AIR OXYGEN ENRICHMENT

Basis: Coal Thermal Input (HHV) 1000 MW
Oxidizer Preheat Temperature 1100°F

		Case 1 <u>30% O₂</u>	Case 2 <u>34% O₂</u>
MHD Power	MW	190	200
Steam Power		<u>317</u>	<u>314</u>
		507	514
Cycle Compressor Power*	MW	49.8	49.7
O ₂ - Plant Compressor Power		24.8	32.3
Auxiliary Losses**		<u>19.5</u>	<u>19.5</u>
	MW	94.1	101.5
Net Plant Output	MW	<u>412.9</u>	<u>412.5</u>
Net Plant Eff.	%	<u>40.9</u>	<u>40.9</u>

*Based on 88% compr. eff. and 15% pressure loss $(\frac{P_{comp.} - P_{burner}}{P_{burner}})$

** Inverter Loss (1%)	1.9	2.0
Transformer Loss (0.75%)	3.4	3.5
Magnet Aux.	0.8	0.8
Coal and Ash Handling	3.5	3.5
Pumps (condensate & circ. water)	2.7	2.7
Seed Feeding	0.7	0.7
Fans (I. D. & Sec. air)	3.5	3.5
Seed Regeneration	2.0	2.0
Misc.	<u>1.0</u>	<u>1.0</u>
	<u>19.5</u>	<u>19.5</u>

TABLE 1-8

PRIME MHD CYCLE PERFORMANCE DATA FOR POWER CYCLE WITH
COMBUSTION AIR OXYGEN ENRICHMENT OF 30%

Basis: Coal Thermal Input 1450 MW

	<u>Case 1</u> <u>1100°F</u> <u>Preheat</u>	<u>Case 2</u> <u>1200°F</u> <u>Preheat</u>	<u>Case 3</u> <u>1400°F</u> <u>Preheat</u>
<u>MHD Coal Combustor</u>			
Combustion Product Flow Rate (kg/s)	388.9	388.9	388.9
Exit Pressure (atm)	5	5	6
Exit Temperature (°K)	2703	2718	2759
Heat Loss (%)	4.00	4.00	4.00
<u>MHD Generator</u>			
Inlet Mach #	0.85	0.85	0.85
Electrical Load Parameter	0.77	0.78	0.75
Max. Transverse Current Density (amps/cm ²)	0.56	0.58	0.71
Max. Axial Field (kV/m)	1.88	1.83	1.95
Max. Hall Parameter ($\omega\tau$)	3.53	3.7	3.6
Power Output (gross)* (MW)	284	289	317
Enthalpy Extraction* (%)	17.8	17.9	19.2
Isentropic Efficiency* (%)	66.8	67.6	66.1
<u>Channel Dimensions (Gas Side)</u>			
Inlet (m x m)	1.04 x 1.04	1.04 x 1.04	0.95 x 0.95
Outlet (m x m)	2.15 x 2.15	2.14 x 2.14	2.17 x 2.17
Length m	16	15	15.5
Power Density (average) (MW/m ³)	6.71	7.33	8.0

* Diffuser recover factor $\frac{\Delta p \text{ (recovered)}}{\Delta p \text{ (actual)}} = 0.5$

TABLE 1-9

PRIME MHD CYCLE PERFORMANCE DATA FOR POWER CYCLE WITH
COMBUSTION AIR OXYGEN ENRICHMENT OF 34%

Basis: Coal Thermal Input 1450 MW

	Case 1 1100°F <u>Preheat</u>	Case 2 1200°F <u>Preheat</u>	Case 3 1400°F <u>Preheat</u>
<u>MHD Coal Combustor</u>			
Combustion Product Flow Rate (kg/s)	351	351	351
Exit Pressure (atm)	6	6	6.75
Exit Temperature (°K)	2791	2802	2834
Heat Loss (%)	4.05	4.05	4.05
<u>MHD Generator</u>			
Inlet Mach #	0.85	0.85	0.85
Electrical Load Parameter	0.77	0.78	0.77
Max. Transverse Current Density (amps/cm ²)	0.72	0.72	0.81
Max. Axial Field (kV/m)	1.7	1.6	1.54
Max. Hall Parameter ($\omega\tau$)	3.65	3.6	3.42
Power Output (gross)* (MW)	296	301.3	321.3
Enthalpy Extraction* (%)	19	19.1	20
Isentropic Efficiency* (%)	67.4	68.2	67.8
<u>Channel Dimensions (Gas Side)</u>			
Inlet (m x m)	0.91 x 0.91	0.91 x 0.91	0.86 x 0.86
Outlet (m x m)	2.05 x 2.05	2.06 x 2.06	2.07 x 2.07
Length m	14.5	14.7	16
Power Density (average) (MW/m ³)	8.88	8.85	8.85

* Diffuser recovery factor $\frac{\Delta p \text{ (recovered)}}{\Delta p \text{ (actual)}} = 0.5$

TABLE 1-10

PRELIMINARY ENERGY BALANCE FOR POWER CYCLE WITH
COMBUSTION AIR OXYGEN ENRICHMENT OF 30%

Basis: Coal Thermal Input 1450 MW

		Case 1 1100°F <u>Preheat</u>	Case 2 1200°F <u>Preheat</u>	Case 3 1400°F <u>Preheat</u>
MHD Power	MW	284	289	317
Steam Power	MW	<u>455</u>	<u>454</u>	<u>448</u>
		739	743	765
Cycle Compressor Power	MW	72.4	72.4	82.1
O ₂ - Plant Power	MW	36.1	36.1	36.1
Auxiliary Losses	MW	<u>27.6</u>	<u>27.7</u>	<u>28.0</u>
		136.1	136.2	146.2
Net Plant Output	MW	<u>602.9</u>	<u>606.8</u>	<u>618.8</u>
Net Plant Efficiency	%	<u>41.3</u>	<u>41.6</u>	<u>42.1</u>

TABLE 1-11

PRELIMINARY ENERGY BALANCE FOR POWER CYCLE WITH
COMBUSTION AIR OXYGEN ENRICHMENT OF 34%

Basis: Coal Thermal Input of 1450 MW

		Case I 1100°F <u>Preheat</u>	Case II 1200°F <u>Preheat</u>	Case III 1400°F <u>Preheat</u>
MHD Power	MW	296	301	321
Steam Power	MW	<u>454</u>	<u>453</u>	<u>448</u>
	MW	750	754	769
Cycle Compressor Power	MW	72.3	72.3	72.3
O ₂ - Plant Comp. Power	MW	47.0	47.0	47.0
Auxiliary Losses	MW	<u>27.8</u>	<u>27.9</u>	<u>28.1</u>
	MW	147.1	147.2	153.4
Net Output	MW	<u>602.9</u>	<u>606.8</u>	<u>615.6</u>
Net Plant Eff.	%	<u>41.3</u>	<u>41.6</u>	<u>42.2</u>

1. 4 OVERALL ETF PLANT DESIGN DESCRIPTION

1. 4. 1 Selected Plant Design Concept and Power System Configuration

The basic power system configuration for the ETF is shown by the simplified schematic shown in Figure 1-6. This schematic serves the purpose for overall description only.

The power system configuration is essentially a combined MHD steam power system with an indirect type high temperature air preheater separately fired initially with clean fuel oil. Final operation of the high temperature air preheater will be with low Btu fuel gas as fuel produced from gasification of coal.

Seeded coal combustion gases are generated in an MHD combustor by burning of coal with preheated air and by the addition of a potassium seed material to the high temperature combustion gases produced. DC power is extracted from these gases in the MHD generator. It is inverted to ac power for supply to the electrical network together with additional ac power generated from the bottoming steam plant recovering waste heat from the hot MHD generator exhaust gases. Steam power produced in the bottoming plant is furthermore, utilized for the turbine drive of the cycle compressor. The heat content of the MHD generator exhaust gas is utilized both for steam generation and for preheating of combustion air and recycle gas for the high temperature air preheater to an intermediate temperature in recuperative metallic heat exchangers. Seed is recovered and processed as necessary with removal of sulfur before recycling.

The particular arrangement of the air preheater system with the air from the compressor delivered directly to the high temperature air preheater was selected because of special design considerations in the high temperature air preheater. This results in a more reliable preheater design than for the alternate preheater arrangement shown in Figure 1-7. This is because a lower temperature in the bottom part of the regenerator is achieved which is also in line with industrial high temperature regenerator experience. Reference is made to Section 2.11 for further discussion of this important preheater design consideration.

The plant design is based on a progressive plant design concept. This provides for progressive testing of components and operation of the

Figure 1-6 Schematic Drawing of Basic ETF System Configuration

Figure 1-7 Schematic Drawing of Basic ETF System Configuration with Alternate Preheater Arrangement

facility. The main purpose of the progressive design concept is to enable testing of MHD prime cycle components separately before integrated operation of these with the remaining portion of the plant as a complete power system. Such a design concept also offers the possibility of performing MHD components testing before the construction of the plant has been completed. Furthermore, it provides during the total program the opportunity for separate initial testing of MHD channels and other prime cycle components at different operating conditions and of different and improved designs as such become available before combined operation of these with the bottoming plant.

The separate testing and operation of the MHD prime cycle components independent of the steam bottoming plant is made possible by the addition of an exhaust gas bypass system as indicated on the schematic in Figure 1-6. This bypass system when operative routes the MHD generator exhaust gas directly to the stack. It quenches the hot MHD generator exhaust gas by water-spraying and cleans the gas of seed and particulate matter by scrubbing before emission of the gas to the atmosphere. During separate testing of the MHD components, the required cooling is accomplished by isolated water cooling. This is provided, in addition to cooling of MHD components, by feed water from the boiler circuit during normal combined MHD steam plant operation.

The high temperature air preheater is initially separately fired with oil. This is in concurrence with DOE's initial Design Guidelines. However, the progressive design concept includes the use of a low Btu fuel gas produced from a coal gasifier for separate firing of the high temperature air preheater system subsequent to the initial use of oil. The detailed design and construction of the gasifier is delayed to match the time for which it is required in the test program schedule. This delay of the gasifier will enable taking benefits from advances which may accrue from parallel developments occurring in coal gasification.

Provision for oxygen enrichment of the combustion air during initial testing has been incorporated in the design. The oxygen storage facilities have been designed for a capacity that will enable 8 hours of continuous testing at full design flow rate and 33 1/3% oxygen content in the oxidizer

($N_2/O_2 = 2$). Also, the MHD generator power output can first be dissipated in a water cooled resistive load bank. This permits initial characterization of the MHD generator operating parameters and testing of the load inverters before connection to the electric utility network system. A detailed description of the progression in the planned testing and operation of the ETF with accompanying test program time schedules is presented in the following Section 1. 5.

Modification of the ETF to future possible operation with high temperature directly fired air preheaters has been considered. However, the design parameters for a practicable directly fired high temperature air preheater system remains to be developed. Thus, it is not possible now to define with a high degree of certainty the modifications of the ETF necessary for future use with directly fired high temperature air preheaters, even though space requirements can be satisfied.

Nevertheless, the division of the steam generator in four major furnaces and sections (primary radiant slag furnace, seed recovery furnace, third chamber with convective sections, and economizer) provides for opportunities to utilize one or more sections (primary radiant slag furnace in particular) if the ETF facility in the future is modified for operation with directly fired air preheaters.

Chemical regeneration represents another potential future MHD power plant design concept. If this proves technically feasible, the possibility exists for modifying the ETF plant design for future testing of this novel design concept also. In this case the MHD generator exhaust bypass arrangement could be utilized for the supply of hot exhaust gas to a coal devolatilizer/gasifier system which then must be added to the facility.

1. 4. 2 Plant Design Parameters and Performance

1. 4. 2. 1 General

The overall cycle efficiency of a combined MHD/steam power cycle is in simple terms

$$\eta_o = \frac{P_{MHD} - P_{compr} + P_{steam} - P_{aux}}{Q_{fuel}}$$

where

$$\begin{aligned}P_{\text{MHD}} &= \text{MHD power output (gross)} \\P_{\text{steam}} &= \text{Steam power output (gross)} \\P_{\text{compr}} &= \text{Cycle compressor power} \\P_{\text{aux}} &= \text{Auxiliary power losses} \\Q_{\text{fuel}} &= \text{Fuel energy input to plant}\end{aligned}$$

Neglecting auxiliary losses (P_{aux}) this is expressed as

$$\eta_o = \eta_{\text{MHD}} + \eta_s (1 - \eta_{\text{MHD}}) \cdot f$$

where

$$\eta_{\text{MHD}} = \frac{P_{\text{MHD}} - P_{\text{compr}}}{Q_{\text{fuel}}} \quad (\text{MHD eff.})$$

$$\eta_s = \text{steam plant efficiency} \left(\frac{\text{Steam Power}}{\text{Heat Absorbed in Steam Cycle}} \right)$$

f = waste heat recovery factor

Thus it is seen that the overall efficiency depends upon the net energy output ($P_{\text{MHD}} - P_{\text{compr}}$) or efficiency (η_{MHD}) of the prime MHD cycle and the efficiency of the bottoming steam plant along with the fraction of waste heat that can be recovered. The basic way to optimize the overall cycle efficiency is to increase η_{MHD} , which means to raise the relative fraction of net power ($P_{\text{MHD}} - P_{\text{compr}}$) from the prime MHD cycle. The amount of power which can be extracted from the MHD generator is governed by thermodynamic and electrical properties of the working fluid along with design parameters and constraints selected for the channel. The MHD generator working fluid is again specified by the fuel, oxidizer and seed compositions and their preheat temperatures, together with the heating value of the fuel and assumed MHD combustion conditions.

Efficient utilization of the waste heat contained in the higher temperature MHD generator exhaust gas and lower temperature exhaust gas from the high temperature air preheater system becomes important for maximizing, $f \cdot \eta_s$ and hence overall cycle efficiency, η_o . Maximum heat recovery is obtained by cooling the combustion gases to the lowest temperature level so as to minimize stack losses, and to recover all heat losses from the prime cycle components [MHD combustor cooling, nozzle cooling, channel cooling, diffuser cooling, high temperature air heater system cooling (if applied), gasifier system cooling (if applied), compressor intercooling (if applied)]. The actual heat energy which can be utilized is limited by physical and economical constraints in heat exchanger design.

1. 4. 2. 2 Plant Size

Mass flow rates, thermal input and power output for the ETF have been established from the following three main considerations:

1. State-of-the-art in MHD technology
2. Scale-up to commercial MHD power generation
3. Cost effectiveness.

The attractiveness of the MHD concept for base load central station power generation is primarily derived from its high thermal efficiency potential. This still provides the principal motivation for development of open cycle MHD power generation. As discussed in the previous section the overall MHD power cycle efficiency depends upon the relative fraction of power extracted from the MHD generator, which is determined by the enthalpy extraction and isentropic efficiency of the MHD generator channel. The enthalpy extraction and isentropic efficiency which can be attained are again determined by the size of the MHD generator. The MHD generator performance follows scaling laws where the enthalpy extraction and isentropic efficiency increase with size. Therefore, the ETF MHD channel cannot be expected to reach the same high performance as that attainable in a larger commercial plant. Consequently, the ETF plant efficiency can also not be as high as efficiency values projected for large commercial plants. However, the ETF MHD channel must be capable of demonstrating values of enthalpy extraction and isentropic efficiency which are considered sufficiently high for projection to those required for efficient operation in

larger commercial plants. This then became a prime requirement for establishing the ETF plant capacity level and its size. Our MHD channel performance and tradeoff analysis discussed in Section 2.1.1 resulted in the selection of 65 kg/s as optimum for the channel. This selection was primarily based on

- Performance goals of enthalpy extraction (η_e) and isentropic efficiency (η_{is}) are achieved even for the lowest preheat temperature of 2500°F considered.

$$\eta_e > 12.5\%$$

$$\eta_{is} > 50\%$$

- The incremental gain in the MHD channel performance beyond 65 kg/s is relatively small.

A mass flow rate of 65 kg/s corresponds to a power level of 35.5 MW at 2500°F preheat temperature and 45.6 MW at 3000°F preheat temperature.

Plant performance analysis also revealed that the total heat loss from the high temperature MHD components were too large to be completely absorbed and utilized within the bottoming steam cycle at a total mass flow rate of less than about 65 kg/s. Thus, a total mass flow rate of 65 kg/s was also in concord with the objective of seeking a relatively efficient ETF plant design.

The main purpose of all tradeoff analyses conducted of the total system and the various system components and subsystems (discussed in their respective sections of this document) was to define the optimum ETF design that fulfills the ETF program objectives and all performance and operational requirements. Such tradeoff analyses involved considerations of technical risks and costs with assessments of the present state-of-the-art of MHD technology, technological issues that must be resolved and of the requirements for subsequent scale-up to commercial MHD power generation. The conclusion from our size analysis is that an ETF capacity level corresponding to a total mass flow rate of 65 kg/s or 250 MW thermal input to the MHD generator at 2500°F preheat temperature or 270 MW_{th} at 3000°F preheat temperature represents an optimum.

In summary, the bases for this ETF size selection are:

1. It is scalable to commercial size and satisfies all critical channel operating parameters.

$$14.2 \leq \eta_e \leq 16.9\%$$

$$57.0 \leq \eta_{is} \leq 57.8\%$$

$$8.4 \text{ MW/m}^3 \leq P_{\text{average}} \leq 11.6 \text{ MW/m}^3$$

$$2.4 \text{ kv/m} \leq E_{x_{\text{max}}} \leq 2.5 \text{ kv/m}$$

$$0.7 \text{ amps/m}^2 \leq J_{y_{\text{max}}} \leq 1.1 \text{ amps/cm}^2$$

2. It is cost effective because the ETF plant cost increases with increased mass flow rate and the selected mass flow rate of 65 kg/s is the minimum flow rate that satisfies the channel performance goals and yet provides a certain margin for achieving these goals.
3. It provides for pilot scale demonstration and testing of all technological issues identified for a complete MHD/steam power system.

The selection of the ETF optimum capacity level was strongly governed by the MHD generator channel tradeoff analysis. This prime consideration given to the MHD channel is natural, because the MHD channel is the central item of any MHD power system in which the unique feature of the MHD concept is concentrated and from which the important MHD power system design requirements are derived.

Table 1-12 summarizes comparative important MHD channel performance data for the Avco Mk VI MHD generator channel, the #1 CDIF MHD flow train, the selected ETF MHD channel design for 2500°F and 3000°F preheat temperature and for various projected commercial baseload power plants. Baseline 1 design data corresponds to the design data presented in Section 1.3.2 for first commercial plants with a separately fired high temperature air preheater and preheat temperatures of 2500°F and 3000°F respectively. Baseline 2 corresponds to design data developed in ERDA's baseline reference plant study on contracts E(49-18)-2228 and 2243. The last column pertains to the mature MHD power plant design considered in ECAS. These two latter design studies are both based on the use of directly fired high temperature air preheater along with a preheat temperature of 2500°F.

TABLE 1-12
COMPARATIVE MHD CHANNEL PERFORMANCE DATA

						COMMERCIAL BASELOAD			
		AVCO MK-VI	CDIF # 1	ETF		BASELINE 1		BASELINE 2	ECAS
				2500 °F PREHEAT	3000 °F PREHEAT	2500 °F PREHEAT	3000 °F PREHEAT	2500 °F PREHEAT	2500 °F PREHEAT
THERMAL INPUT	MW _{th}	15	40 - 50	250	270	1397	1511	2850	5300
PRESSURE RATIO		2.9 - 3.5	5.5	4.0	5.4	5.5	6.5	7.0	8.0
INLET TEMP. (STAG.)	°K	2900	2875	2675	2794	2744	2845	2780	2830
MAX. MAGNETIC FIELD	TESLA	2.8	3	6	6	6	6	6	6
MASS FLOW RATE	kg/s	2.5	8 - 10	65	65	353	353	700	1270
POWER OUTPUT	MW	0.2-0.6	1.5-2.5	35.5	45.6	288	342	634	1420
MAX. TRANSV. CURRENT	amps/cm ²	0.8-1.5	0.9-1.5	0.65	1.1	0.7	0.9		1.0
MAX. AXIAL FIELD	kV/m	1.0-3.0	2.0-3.0	2.5	2.4	1.9	1.7		2.7
HALL PARAMETER	$\omega \tau$	1.5-2.5	1.6-2.3	4.4	4.4	3.6	3.9		4.1
POWER DENSITY (AVG)	MW/m ³	5-15	7-12	8.4	11.6	6.4	8.7	10.5	10.0
ENTHALPY EXTRACTION	%	1.5-4.0	4 - 6	14.2	16.9	20.6	22.6	22.3	26.5
ISENTROPIC EFFICIENCY	%	≈ 40		57.0	57.8	68.5	71.5		76.0
DIFFUSER EFFICIENCY	%	≈ 45		46	46	61	61		70

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The enthalpy extraction of the various channels listed in Table 1-12 are plotted in Figure 1-8 against the thermal inputs to the channels which are a measure of the channel capacity or its size. It has already been mentioned that the MHD generator follows scaling laws where enthalpy extraction and isentropic efficiency will increase with size for certain assumed design conditions. The reason for this performance improvement with size is that the power extraction occurs throughout the gas volume while the losses due to heat transfer, friction and electrode drop occur at the walls. The expected improvement in performance with size, caused by the decrease in surface to volume ratio, pertains also to the MHD combustor and nozzle as well. Thus, for the same input conditions of fuel, combustion air and seed to the combustor (fuel heating value, preheat temperature, seed concentration, reactants compositions, etc.) the gas channel inlet temperature and hence its conductivity and available temperature regime for energy extraction in the channel will increase as the combustor size increases. All of this means that the size plays an important role for establishing the attainable thermodynamic performance parameters of the channel (η_e , η_{is}) and of the MHD prime cycle (η_{MHD}). The important improvement in overall cycle efficiency resulting from improved MHD cycle efficiency (η_{MHD}) was reviewed earlier.

Taking all size effects into consideration it was judged that the ETF channel should, as a minimum, be capable of 12.5% enthalpy extraction at 50% isentropic efficiency at the lowest considered preheat temperature of 2500°F. These lower limiting values of performance for the ETF channel were considered minimum requirements for extrapolation to the higher performance values projected for efficient operation in large scale commercial MHD power plants. The curve shown in Figure 1-8 is representative of our scaling in performance from the ETF channel size based on the lower limiting values of the ETF channel performance requirements ($\eta_e = 12.5\%$, $\eta_{is} = 50\%$) to the calculated performance values projected for larger baseload channels. This scaling has taken into account improvements both in burner and channel performances resulting from increases in size. (Ref. Table 1-12). The calculated performance values of the ETF channel at selected design conditions of 250 MW and 270 MW thermal inputs for 2500°F and 3000°F preheat temperatures are somewhat higher than those

Figure 1-8 Enthalpy Extraction as a Function of Channel Thermal Input for Various MHD Channels

judged as the minimum performance requirements. This provides for a certain margin which we consider highly desirable for the ETF.

The critical ETF channel design parameters such as power density, axial field, current density and Hall parameter are of the same order (or exceed) those projected for commercial operation. On the other hand, the experimental work with the Mk VI generator has already provided a good basis for the ETF channel design. Thus, in conclusion, the ETF channel design fulfills all design criteria considering both technical risks in scale-up from present state-of-the-art and requirements for subsequent scaling to commercial MHD power generation. Reference is made to the following sections of this report for further discussion and description of the channel and other system components and subsystem designs.

1. 4. 2. 3 System Design Parameters

Overall system design parameters are listed in Table 1-13. These are only briefly reviewed below in this section. Reference is made to the subsequent subsystem/components design section for detailed description of their respective design and operational characteristics. The total enthalpy of the gas delivered to the MHD channel (after deduction of upstream heat losses) corresponds to a thermal input of 250 MW at 2500°F and 270 MW at 3000°F preheat temperature.

Montana subbituminous coal is the primary fuel for firing of the MHD combustor. It is pulverized to 70% through 200 mesh and assumed dried to 5% moisture content before firing. The coal is dried with exhaust gas from the high temperature air heater. This has the advantage of accomplishing coal drying in all modes of operation with relatively warm combustion gases (~650°F) free of seed.

The MHD combustor is a single stage type combustor. The main reason for selection of a single stage combustor as opposed to a multistage combustor type is that it appears to offer less heat loss and less complexity. It is emphasized that very limited information is available of actual coal combustor operating conditions and its important interphasing with the MHD channel and other power system components. Therefore, the coal combustor design is considered preliminary and should be verified or revised as necessary when further information becomes available from future experimental work with coal combustion.

TABLE 1-13
OVERALL SYSTEM DESIGN PARAMETERS

Thermal Input to MHD Channel

At 2500°F preheat temperature	250 MW
At 3000°F preheat temperature	270 MW

Main Fuel

Type	Montana sub bit. , 8920 Btu/lb HHV, 22.7% moist. (Secondary coal type Ill. #6, 11265 Btu/lb HHV, 8.9% moist.)
Size	Pulverized 70% through 200 mesh
Moist. content, dried	5% (2% for Ill. #6)
Oxidizer	Air

MHD Combustor

Type	Single Stage	
Air Preheat temp. (°F)	2500	3000
Comb. exit pressure (atm)	4.0	5.4
Comb. exit temp. (°F)	4360	4570
Air/fuel Equivalence ratio	0.93	0.93
Comb. gas seed conc.	1% K	1% K
Slag rejection	80 to 90%	80 to 90%
Coolant	High pressure B. F. water	

MHD Generator

Channel Type	Ideal Faraday and Multiterminal Diagonal
Channel Flow	Subsonic
Channel Coolant	Low Pressure B. F. water, temp. 220°F max.
Magnet	Sup. cond., 6 Tesla peak field intensity
Diffuser exit pressure (atm)	1.0
Diffuser pressure recovery factor	0.46
Diffuser Coolant	High Pressure B. F. water

High Temperature Air Preheater

Type	4 regenerator units, separately fired
Fuel	Initially oil, eventually LBtu gas

Bottoming Heat Recovery Plant

<u>Primary Furnace</u>		
Type	Radiant, wet bottom	
Gas cooling time	2 seconds to 2900°F	
H. E. surfaces	evaporator	
Slag removal	46%	
<u>Secondary Furnace</u>		
Type	Radiant, seed condenser	
H. E. surfaces	evaporator, finishing superheater	
<u>Afterburner</u>		
Final air/fuel equiv. ratio	1.05	
<u>Third Furnace Chamber</u>		
Type	Convective	
H. E. surfaces	Evaporator, front superheater, air and recycle gas tubular heaters	
<u>Economizer</u>		
Type	Convective	

Bottoming Steam Cycle

Steam conditions	1250 psi/950°F
Steam cycle eff. for power turbine	33.2%
Steam cycle eff. for compressor turbine	31.8%
Condensing pressure (in Hg)	2.5

Seed Regeneration

Process	Formate
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Two MHD generator channel design configurations are presented, the ideal Faraday and multiterminal diagonal types. The two terminal diagonal channel type has not been pursued further because it does not offer potential for baseload application due to its expected lesser efficiency. In these conceptual design analyses the performance of the ideal Faraday and and multiterminal diagonal channels are considered to be the same, but with different electrical load circuitries. The channel flow is subsonic which results in efficient operation and favorable part load operating characteristics. For cooling of the MHD channel, low pressure water of a maximum temperature of 220°F has been employed in accordance with present state-of-the-art in MHD technology.

The MHD generator magnet is superconducting with a maximum field intensity of 6 Tesla. The magnet design is novel. Special attention has been given to making the design modular to allow critical components such as the windings and major structural items to be factory-fabricated in subassemblies for easy transport to the site and to facilitate field construction, assembly and disassembly.

The diffuser pressure recovery factor is estimated to be 0.46. This is in line with expected efficiencies reported for MHD subsonic diffusers of this type and blockage factor. The gas exits from the diffuser transition section into the radiant primary boiler furnace operating at 1 atm pressure.

The high temperature regenerative air preheater system consists of four units with individual, external reheat burners. It offers a high degree of flexibility with possible full capacity operation with either three of four units. Reliability is emphasized with the selection of a relatively low temperature air inlet, and by minimizing the number of valves required for operation at the highest temperatures of the preheater system. The preheater system operates with combustion gases essentially under atmospheric pressure. The high temperature air preheater system is separately fired initially with oil and eventually with a low Btu gas delivered from a coal gasifier. The amount of fuel necessary for firing of the high temperature air preheater is minimized by preheating of the combustion air required for burning of this fuel from the MHD generator exhaust gas. Recycling of combustion gas is utilized for tempering purposes and to match the flow

rates of the fluids. The preheater recycle gas is also preheated by the MHD generator exhaust gas.

The bottoming heat recovery plant has been designed with careful consideration to the peculiar problems associated with its application to MHD. The primary radiant boiler slag furnace is refractory lined and provides for a gas residence time of two seconds before cooling to 2900°F. This gas cooling rate along with the initial substoichiometric fuel/air equivalence ratio of 0.93 in the MHD combustor is expected to reduce the NO_x-level in the gas to below EPA standards. As discussed in Section 2.13.2 a fuel/air equivalence ratio of 0.94 or less is required to satisfy present EPA standards. Additional reduction in the NO_x level to roughly half of present EPA standards can be accomplished by further reduction of the fuel/air equivalence ratio to 0.90 in the MHD combustor. Preliminary investigations indicate that the MHD generator performance would remain the same for this slight additional fuel enrichment. The primary radiant furnace has a bottom slag-tap and close to half of the ash contained in the MHD generator exhaust gases is assumed to be removed and tapped from the furnace bottom.

The gas exits from the primary furnace at a gas temperature of 2900°F into a second seed recovery furnace where the gas is cooled to about 1900°F. This serves to condense and solidify seed in the gas, and some of the seed together with some of the remaining ash is collected in a dry hopper located at the bottom of the second furnace.

Afterburning with final oxidation of the gas is carried out at a gas temperature of 1900°F. Analytical investigations show that this gas temperature is sufficiently high for relatively rapid and complete oxidation of unburned carbon monoxide in the gas. The final total air/fuel mixture ratio corresponds to 5% in excess of the stoichiometric ratio.

The last gas cooling is accomplished by convective heat exchange including surfaces needed for heating of combustion air and recycle gas for the high temperature air heater. The gas is cooled to its final temperature of 275°F in an economizer fed with high pressure boiler feed water.

The stack gas is cleaned of seed and fly-ash impurities in an electrostatic precipitator. Seed (with fly-ash) recovered from the secondary seed recovery furnace, convective sections of the boiler and the electrostatic precipitator is processed as necessary for sulfur removal and recycled.

The basic design conditions selected for the bottoming steam plant were steam pressure and temperature of 1250 psig/950°F, and 2.5 in. Hg vacuum. The steam conditions were based on initial information obtained from two U.S. steam turbine manufacturers indicating that 1250 psig/950°F was in compliance with their standard off-the-shelf equipment. Recently, one U.S. manufacture has indicated that main steam throttle conditions of 1450 psig/1000°F and 1800 psig/1000°F are not considered as special design conditions for ETF size. This information was obtained too late to change the basic steam conditions for the bottoming plant design. However, preliminary estimates of changes in performance and costs resulting from selection of these other indicated standard (or not special) steam conditions have been made and are presented in the next section where possible improvements in plant performance are discussed.

The selected seed regeneration process for the ETF is a modification of the formate process. The design of the seed regeneration process is based upon the assumption that K_2SO_4 is formed in the gas. Consequently, the seed regeneration process must consider processing of K_2SO_4 for sulfur removal and recycling of seed. The possibility exists that K_2S is formed in the gas under prevailing reducing gas conditions. This would have distinct advantages for seed processing in that processing to reduce K_2SO_4 to K_2S would not be required or significantly lessened. However, thermochemical data related to K_2S are presently uncertain, so that complete formation and subsequent processing of K_2SO_4 was adopted.

The main reason for the selection of the formate process for seed regeneration at the present time is that this process has been used in industrial practice before and is thermodynamically attractive. It is mentioned that the formate process is subject to the resolution of certain process uncertainties. In spite of this, it is considered a viable process whereas the conversion of sulfate to carbonate by the PERC process remains uncertain. The use of carbon or coal as the main agent for reducing K_2SO_4 to K_2S appears potentially attractive. Therefore, the carbon reduction process should in our opinion be considered as a potential candidate in future ETF design work even though important problems remain to be solved before a carbon reduction process can be considered technically and economically

feasible for potassium seed regeneration. It is recognized that the formate process produces CaSO_4 as a solid waste disposal product, and that the PERC and carbon reduction process yield H_2S which is considered converted to elemental sulfur as the final waste product. In the ETF design analysis CaSO_4 and sulfur were considered equally acceptable as waste products, whereas technical uncertainties were emphasized. The modified formate process for seed regeneration requires a small amount of coke or coal to provide CO for the formation of KCO_2H .

1.4.2.3 Plant Performance

1.4.2.4.1 Full Load

Mass flow rates and heat inputs and outputs for the MHD/steam plant and for the high temperature air preheater system at design conditions are listed in Table 1-14 and 1-15. Two conditions are specified, one for a combustion air preheat temperature of 2500°F and the other for a combustion air preheat temperature of 3000°F . All ETF design requirements including the performance of the MHD channel are met at 2500°F preheat temperature as stated earlier. This preheat temperature is also considered to be the operating preheat temperature for testing of the ETF at first. However, all equipment has been designed for operation at the higher preheat temperature of 3000°F so that the ETF is capable of demonstrating the important improved performance resulting from the possible operation with this higher preheat temperature.

Schematic heat balance cycle diagrams of the ETF plant for operation with 2500°F and 3000°F preheat temperatures at full load are shown in Figures 1-9 and 1-10. State point conditions are indicated on these cycle diagrams as well as heat losses from the high temperature MHD components and the amounts of heat absorbed from the combustion gases in the bottoming plant. The indicated conditions represent the basic ETF plant parameters for the design of all plant equipment to be described in the following sections of this report.

ETF overall heat balance with heat inputs and heat outputs is shown in Table 1-16 and the plant energy balance with power outputs is shown in Table 1-17. A breakdown of the ETF plant auxiliary normal power requirements are contained in Table 1-18.

TABLE 1-14

MASS AND HEAT BALANCES--MHD/STEAM PLANT

Mass Flow Rates

Raw Coal (22.7% H ₂ O)	pph	71,400
Dried Coal (5% H ₂ O)	pph	58,100
Primary Air (93% of Theor.)	pph	454,000
Secondary Air (5% Excess)	pph	54,000
Seed (All as K ₂ SO ₄)	pph	11,500

2500°F
Preheat

3000°F
Preheat

Heat Inputs

Coal (HHV)	MW	186.5	186.5
Coal Drying	MW	4.5	4.5
Preheat Primary Air	MW	87.2	107.3
Preheat Secondary Air	MW	1.4	1.4
Seed Chemistry	MW	<u>0.7</u>	<u>0.7</u>
Total		<u>280.3</u>	<u>300.4</u>

Heat Outputs

MHD Power	MW	35.5	45.6
MHD Heat Losses			
Burner (Incl. Nozzle)	MW	13.6	16.8
Channel	MW	11.5	15.5
Diffuser (Incl. Trans. Section)	MW	13.7	14.5
Steam Gen. and Economizer	MW	162.8	166.6
I. T. Heaters	MW	19.3	17.5
Stack Loss:			
Sensible at 275°F	MW	8.1	8.1
Latent	MW	11.6	11.6
Slag Heat Loss	MW	1.4	1.4
Other Heat Losses	MW	<u>2.8</u>	<u>2.8</u>
Total		<u>280.3</u>	<u>300.4</u>

TABLE 1-15

MASS AND HEAT BALANCES--H. T. AIR HEATER

		<u>2500°F Preheat</u>	<u>3000°F Preheat</u>
<u>Mass Flow Rates:</u>			
Fuel	$10^3 \times \text{pph}$	13.57	17.70
Combustion Air	$10^3 \times \text{pph}$	201.20	267.51
Recycle Gas	$10^3 \times \text{pph}$	<u>166.60</u>	<u>96.05</u>
Total	$10^3 \times \text{pph}$	381.37	381.26
<u>Heat Inputs</u>			
Fuel (HHV)	MW	77.00	100.40
Air (1100°F)	MW	15.10	20.00
Recycle Gas (1100°F)	MW	<u>13.00</u>	<u>7.50</u>
Total		<u>105.10</u>	<u>127.90</u>
<u>Heat Outputs</u>			
MHD Cycle Air	MW	76.20	93.80
L. T. Air Heater	MW	6.30	8.20
Coal Drying	MW	4.50	4.50
Recycle Gas Heat	MW	2.10	1.20
<u>Stack Loss:</u>			
Sensible Heat (250°F)	MW	2.30	3.10
Latent Heat	MW	4.50	6.00
Total Heat Losses from Air Heater System	MW	<u>9.20</u>	<u>11.10</u>
Total		<u>105.10</u>	<u>127.90</u>

Figure 1-9 System Heat Balance with 2500°F Preheat Temperature

Figure 1-10 System Heat Balance with 3000°F Preheat Temperature

TABLE 1-16
ETF OVERALL ENERGY BALANCE

Energy Inputs:

A. Combustion Energy:

Coal (HHV)	MW	186.5	186.5
Oil (HHV)	MW	<u>77.0</u>	<u>100.4</u>
Subtotal	MW	263.5	286.9

B. Other Energy Inputs:

Air + Gas Blower Power	MW	2.0	1.9
Air Compressor Power	MW	12.0	13.6
Seed/Sulfur Chemistry	MW	<u>0.7</u>	<u>0.7</u>
Subtotal	MW	14.7	16.2
Total Input (A + B)	MW	<u>278.2</u>	<u>303.1</u>

Energy Outputs:

MHD Power	MW	35.5	45.6
Heat to Steam	MW	201.6	213.4
Leaving Losses:			
Stack Gas (Sensible)	MW	10.4	11.2
Stack Gas (Latent)	MW	16.1	17.6
Slag	MW	1.4	1.4
Other Plus Unaccounted Losses	MW	<u>13.2</u>	<u>13.9</u>
Total Output	MW	<u>278.2</u>	<u>303.1</u>

TABLE 1-17
ETF PLANT ENERGY BALANCE

Gross Power Outputs:

MHD Power	MW	35.5	45.6
Steam Power:			
Air Compressor [*]	MW	12.0	13.6
A/C Generator ^{**}	MW	<u>54.4</u>	<u>56.7</u>
		101.9	115.9

Auxiliaries and Losses:

Inverter and Transformer Losses ^{***}	MW	1.2	1.4
Auxiliaries	MW	10.1	10.1
Air Compressor	MW	<u>12.0</u>	<u>13.6</u>
		23.3	25.1
Net Electrical Output	MW	<u>78.6</u>	<u>90.8</u>
Efficiency ^{****}	%	<u>29.8</u>	<u>31.7</u>

^{*}Based on turbo-compressor steam cycle efficiency of 31.8%.

^{**}Based on turbine-generator steam cycle efficiency of 33.2%.

^{***}Inverter losses estimated at 2%, transformer losses at 1/2%.

^{****}This is based on total fuel heat inputs with coal and oil as listed in Table 1-16.

TABLE 1-18
ETF PLANT AUXILIARIES

		<u>Normal Running Load-kW</u>
SC Magnet	Compressor	240
Air Preheaters	Htr. Press. Air Compr.	375
	FD Comb. Air Blower	2240
	Recycle Gas Blower	610
	480 V Loads	60
Coal Combustor	480 V Loads	10
Coal Handling	Feeders, Conveyers and Others	100
	Coal Pulverizer	400
	Coal Fan	150
	Flue Gas Compressor	650
Petrocarb. System	Coal and Seed Feed Motive Air Compr.	34
Steam Generator	Boiler Circ. Pumps	200
	Boiler Feed Pump	1010
	Condensate Pumps	80
	I. D. Fan	550
	Soot Blowers (Cont. + Int.)	708
	480 V Auxiliaries	50
	El. Precipitator	160
Balance of Plant	Circ. Water Pumps	540
	Cooling Water Pumps	76
	Instr. Air Compr.	35
	Service Air Compr.	260
	Steam Turb. Gen. Aux.	50
	Cooling Tower Fans	270
	Fire Pump	10
Seed Regeneration	Compressor and Other	300

TABLE 1-18
ETF PLANT AUXILIARIES (Continued)

		Normal Running Load-kW
Station Services	Bldg. and Yard Ltg.	230
	Control Pwr., Batt. Chg. Vital A/C	100
	Control Bldg. HVAC	400
	SWYD, Inverter and X FMR Aux.	150
	HVAC	50
Total		10090

The calculated plant efficiencies in Table 1-17 are based on the total heat input with coal to the MHD burner and with clean fuel (oil or clean low Btu fuel gas) to the high temperature preheater burners. For comparison a conventional plant with the same steam cycle efficiency (steam turbine power/total heat absorbed in steam cycle) of 33.2% as used for the ETF steam turbine generator (1250 psi/950°F), would correspond to a net thermal plant efficiency of about 28.4% assuming 90% steam generator efficiency and 5% of the gross output power used for auxiliaries. This indicated conventional steam plant efficiency does not consider scrubber losses or other losses related to necessary sulfur removal.

1.4.2.4.2 Part Load

Commercial baseload power plants are designed to operate at their best efficiency at nominal load, but the maintenance of a relatively high plant efficiency at part load is also an important consideration. Plant controls and load following characteristics are additional important design considerations for part load operation. These latter aspects are reviewed later in Section 1.4.4. This section deals directly with the part load performance on efficiency aspects.

A general review of the overall efficiency of a combined MHD/steam power plant was presented in Section 1.4.2.1. In general, in order to maintain a relatively high plant efficiency at part load, one continues to seek to optimize the net energy output (MHD generator power output minus compressor power) of the prime MHD cycle, and to utilize the waste heat effectively in the bottoming steam cycle as in full load operation. Thus, the objective of our part load performance analysis has been to establish the plant operating parameters at off design with optimum efficiency always as a goal.

The MHD generator is a central item in these analyses. Little information has so far been developed regarding part load operation of MHD channels. Therefore, our analyses presented here are preliminary and much further design work is required to establish the optimum channel and other plant operating conditions at part load.

Results from our part load channel calculations are presented in Section 2.3.3, and Tables 2-19 and 2-20 in that section summarizes the results of these calculations.

Part load of the MHD generator was accomplished by reducing the mass flow rate, and hence the thermal input to the channel, while maintaining gas stoichiometry, seed fraction and the channel electrical load parameter. The reduction in mass flow rate is accompanied by a reduction in burner pressure. It is important to note that for subsonic channel operation as selected for the ETF, the reduction in burner pressure from nominal load is less than the corresponding reduction in mass flow rate. Our analyses show that the electrical efficiency can be maintained or perhaps could even be increased from that of normal load operation. This leads to relatively efficient part load channel performance. It indicates the favorable part load operating characteristics of the subsonic channel operation as compared to supersonic channel operation in which case the burner pressure must be reduced in direct proportion to the reduction in mass flow rate.

The variations in gross MHD generator power output as a function of mass flow rate are plotted in Figure 1-11 for preheat temperatures of 2500°F and 3000°F. The corresponding variations of the net power output from the prime MHD power cycle (MHD generator power minus compressor power) is also plotted on this figure.

Figure 1-12 shows the variations in channel enthalpy extraction as a function of mass flow rate for the two preheat temperatures of 2500°F and 3000°F. The net enthalpy extraction which is the important measure of the net power output from the prime MHD cycle (MHD generator power minus compressor power) remains practically constant down to 90% of nominal mass flow rate.

From this point on down to 70% of nominal mass flow rate the reduction in net enthalpy extraction is quite small and 0.6 percentage points for both preheat temperatures. (From 11.9% to 11.3% for 3000°F preheat and from 9.4% to 8.8% for 2500°F preheat.) A further reduction to 50% of nominal mass flow rate leads to a more pronounced reduction of the enthalpy extraction for the higher preheat temperature of 3000°F as compared to that for 2500°F preheat. This behavior results mainly from the operating characteristics of the particular compressor selected, because a substantial reduction in compressor efficiency is associated with mass flow rates of less than 70% of nominal at 3000°F preheat whereas the compressor efficiency

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Figure 1-11 Power Output as Function of MHD Channel Mass Flow Rate - % of Nominal

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Figure 1-12 Enthalpy Extraction as Function of MHD Channel Mass Flow Rate - % of Nominal

remains relatively high for the same reduced flow rates at 2500°F preheat. It points out the importance of establishing part load operating characteristics for the compressor before its final design and selection. The channel operating parameters during part load operation remains reasonable, and the values of axial field (E_x), transverse current (J_y) and Hall parameter ($\omega\tau$) decrease with decreasing mass flow rate (Ref. Tables 2-19 and 2-20, Section 2.3.3).

In conclusion, it appears that the MHD generator can be operated at part load without any significant reduction in its thermal performance down to mass flow rates of at least 70%. Different preliminary MHD system performance data at various loads are listed in Table 1-19.

Limited part load performance investigations were also carried out for the bottoming steam plant. These revealed that a steam superheat temperature of 950°F could not be maintained below a total gas mass flow rate of 90% of nominal, because of resulting reduced gas temperatures in the various sections of the steam generator. Also, for the same reason the preheat temperatures of the combustion air and the recycle gas for the high temperature air preheater would decrease with decreasing gas mass flow rate through the steam generator with its air and recycle gas heaters. According to our preliminary investigations, the steam superheat temperature and steam production would decrease to about 885°F and 395,000 pph respectively at 70% gas mass flow rate, and to about 840°F and 305,000 pph respectively at 50% gas mass flow rate. The preheat temperature of the combustion air and recycle gas are expected to decrease to about 755°F and 680°F respectively at 50% gas mass flow rate. These significant decreases in air and recycle gas preheat temperatures at 50% mass flow rate must be compensated for by additional fuel consumption in the high temperature air preheater system which has the effect of lowering the overall net plant efficiency.

Preliminary data of steam power generated in the plant at 70% and 50% of nominal mass flow rates are listed in the Table 1-20 together with the corresponding MHD power produced, auxiliary power required and estimated losses. Further analysis is necessary to verify the part load performance data and to establish overall heat and energy balances at part load operating conditions.

TABLE 1-19.

DIFFERENT PRELIMINARY MHD SYSTEM PERFORMANCE DATA AT VARIOUS LOADS

Basis: Montana sub-bituminous coal (5% moisture as fired)

93% Theoretical MHD Combustion Air, 3000°F Air Preheat Temperature

Mass Flow Rate	%	100	90	70	50
<u>MHD Burner:</u>					
Pressure (exit)	ata	5.40	4.99	4.19	3.40
Temperature (exit)	°F	4570	4560	4540	4515
Heat Loss (incl. nozzle)	MW	16.8	15.8	13.7	11.2
<u>MHD Generator:</u>					
Power Output	MW	45.6	40.7	29.8	19.0
Enthalpy extraction	%	16.9	16.7	15.8	14.1
Heat Loss	MW	15.5	14.3	11.6	9.1
<u>Diffuser (incl. transition section):</u>					
Pressure Recovery Factor		0.46	0.46	0.46	0.46
Heat Loss	MW	14.5	13.3	11.1	8.7
<u>MHD - Exhaust to Boiler:</u>					
Pressure	ata	1.0	1.0	1.0	1.0
Temperature	°F	3700	3690	3670	3650

TABLE 1-20
PRELIMINARY PLANT PART LOAD PERFORMANCE DATA

Total Gas Mass Flow Rate - % of Nominal		70	50
<u>Gross Power Output</u>			
MHD Power	MW	29.8	19.0
Steam Power:			
A/C Generator	MW	40.4	28.7
Air Compressor	MW	<u>8.4</u>	<u>5.7</u>
	MW	78.6	53.4
<u>Auxiliary and Losses</u>			
Air Compressor	MW	8.4	5.7
Aux. and Other Losses	MW	<u>9.0</u>	<u>7.5</u>
	MW	17.4	13.2
Net Electrical Output	MW	<u>61.2</u>	<u>40.2</u>

1.4.2.4.3 Plant Performance Improvements

As stated earlier, the steam bottoming plant was designed for 1250 psig/950°F steam conditions in accordance with initial information obtained from two U.S. turbine manufacturers as standard conditions for turbine generator equipment of ETF size. Recent information obtained indicate that throttle conditions of 1450 psig/1000°F and 1800 psig/1000°F are not considered special design conditions for the ETF application. Therefore, preliminary estimates have been made of the incremental gains in steam plant efficiency and costs which would result from selection of higher steam pressure and temperature. Reference is made to Section 2.13.4.1 for a discussion of these estimates. The table below taken from this section summarizes the results:

<u>Throttle Conditions psig/°F</u>	<u>Increase In Steam Plant Efficiency</u>	<u>Increase In Steam Plant Cost</u>
1250/950	base	base
1400/1000	0.6%	0.1%
1800/1000	0.9%	14%

Considering the incremental gain in efficiency relative to the incremental increase in cost it appears that steam throttle conditions of 1400 psi/1000°F represent the optimum for ETF application. The resulting increase of 0.6 percentage points in steam plant efficiency for these throttle conditions raises the power output 1.2 MW and the overall plant efficiency 0.45 percentage points. Improvement of the steam plant efficiency would also result from rearrangement and redesign of the feed water heating system. This possibility will now be discussed.

First, the possibility exists to incorporate diffuser cooling as part of the boiler circuit because the heat transfer rate in the diffuser is compatible with this. However, high pressure feed water cooling of the combustor will be retained because it operates at peak cycle temperature and pressure with relatively high heat transfer rates. As previously stated it is presently considered prudent to employ feed water of relatively low pressure and low temperature for cooling of the channel. This is consistent with present state-of-the-art.

Secondly, the possibility exists to reduce the economizer duty. This can be accomplished by different arrangements of the convective heat exchanger sections of the boiler including heaters for combustion air and recycle gas for the high temperature air preheater. Such alternate rearrangements of the convective heat exchanger sections are schematically illustrated in Figure 1-13(a). The rearrangements are based on minimum temperature differentials or temperature approaches between gas to gas of close to 100°F and between gas to water evaporation of about 50°F . With these surface rearrangements, the exit temperature of the gas from the convective boiler sections are reduced from 805°F to 590°F which then represents the inlet gas temperature to the economizer. Correspondingly, the economizer duty is reduced from originally 22.3 MW to about 13.5 MW.

The inclusion of diffuser cooling in the boiler circuit together with rearrangement of boiler, air and gas heater surfaces as discussed above lead to the fuel water and steam cycle configuration as shown on the schematic in Figure 1-13(b). The preliminary data points listed on the figure pertain to the original steam pressure and temperature conditions of 1250 psi and 950°F . Two high pressure feed water heaters operating with extraction steam pressures of 244 psia and 640 psia have been selected. The estimated incremental gain in steam turbine generator cycle efficiency with this feed water arrangement is estimated to be 1.1 percentage points. This corresponds to a gain in overall cycle efficiency of 0.7 percentage points. Thus, for selection of steam throttle conditions of 1400 psig/ 1000°F and the use of two high pressure feed water heaters the total gains in the overall plant net efficiency from base design conditions is estimated to be 1.1 percentage points. A summary of the attainable net plant efficiencies possible with the performance gains discussed above is shown in the table below.

<u>Steam Conditions</u>	<u>Net Plant Efficiency</u>	
	<u>2500° Preheat</u>	<u>3000° Preheat</u>
1. Base 1250 psig/ 950°F	29.8	31.7
2. a. 1450 psig/ 1000°F	30.2	32.1
b. 1450 psig/ 1000°F (modified w/two HPFW Heaters)	30.9	32.8

Figure 1-13(a) Alternate Arrangement of Boiler Convective Heat Exchanger Section and Economizer

		①	②	③	④	⑤	⑥	⑦	⑧	⑨
FLOW	(LB/HR)	521,945	105,047	516,594	516,594	516,594	523,044	523,044	523,044	523,044
PRESSURE	(PSIA)	1,285	1,285	1.23			20			
TEMPERATURE	(° F)	950	950	109					312.5	398
ENTHALPY	(BTU/LB)	1,468.5	1,468.5	76.7		182.51	196.3		292.4	398.8

		⑩	⑪	⑫	⑬	⑭	⑮	⑯	⑰	⑱
FLOW	(LB/HR)	567,577	567,577	626,992	626,992	6,450	44,523	59,415	44,533	59,415
PRESSURE	(PSIA)					190	244	640		
TEMPERATURE	(° F)	398		478	553	550			322	398
ENTHALPY	(BTU/LB)	383.7	472	426.6	554	1,298	1,316	1,408	292.4	372.8

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Figure 1-13(b) Modified Steam Cycle with Two High Pressure Feedwater Heaters and Diffuser Coding as Part of Boiler Circuit

A further slight increase in steam plant efficiency might be accomplished by splitting of the economizer duty in two. One part of the economizer duty would then be utilized for production of saturated steam for supply to the dearator and to the seed regeneration process instead of extraction steam from the turbine. The steam would be produced at a relatively low pressure such as 250 psi. The remaining portion of the economizer would serve as before to heat high pressure boiler feed water. The economizer duty for direct feed water heating with this latter arrangement would be reduced to about 9.5 MW. Further design work is necessary to establish the optimum design of the steam plant and feed water cycle arrangements. Since this is a tradeoff essentially between capital costs and operating costs (efficiency) rather than demonstration of technical feasibility, this tradeoff must be based on values of capital charge rate and capacity factor applicable to the ETF to be meaningful.

1. 4. 3 Overall Plant Lay-out Considerations

The plant layout and general arrangement of equipment for the selected ETF design concept is shown on the plot plan and plant island general arrangement drawing in Figures 4-1 and 4-2 of Section 4. 0. Plant Island sections and details with elevations of equipment arrangements are shown in Figures 4-3 through 4-7 of the same section. This section also contains a description of the equipment arrangement with tradeoffs considered.

A prime consideration in the plant layout has been to provide for ease of testing, operation, maintenance and constructability. A horizontal configuration of the MHD generator and its superconducting magnet was selected which gives a direct and straight flow path of combustion gases from the burner to the steam generator. The selection of a horizontal flow path facilitates installation and maintenance and the removal and replacement of channels. The channel centerline elevation was chosen to minimize excavation and also to keep the equipment close enough to grade elevation to facilitate maintenance. The air outlet from the high temperature air heater is at the same elevation as the burner. This provides for horizontal high temperature air ducting of minimum length which minimizes heat loss and duct bends. The high temperature air heater and combustor slag collection equipment is located in a pit.

Component accessibility and adequate lay-down space are provided. The maintenance building is located such as to ease channel transportation in case of required maintenance. The control room is centrally located for convenience and ready access to the plant.

The exit end of the channel is at ground potential so that the front end and the burner will be operating at the axial Hall voltage developed across the channel. The equipment arrangement and plant layout have taken into account safety considerations for exposure to magnetic and electrical fields. Thus, personnel will be located in areas where the field is less than 200 gauss for continuous or relatively long term exposure. The equipment is arranged such that there is one common chimney for operation both with and without the bottoming steam plant. Space requirements for possible future plant modifications was also considered in lay-out of the plant.

1. 4. 4 Plant Operation and Controls

The operational characteristics and controls of the various subsystems and components are described individually in their respective sections of this report, and Section 3.0 contains a description of the Facility Operating Control System and its functions. The necessary hardware and instrumentation of the Facility Operating Control System is described in Section 3.0 and those of the various individual subsystems in their respective sections.

This section defines and discusses the operational requirements of the plant from the overall plant viewpoint. This comprises all different modes and phases of plant testing and operation including normal operation, transients, start-up, shut-down, emergency and safety situations.

Little information has so far been developed related to the electrical and aerodynamic response of the MHD generator to load variations and to the dynamic behavior and response of a combined MHD steam power system as a whole. Such knowledge is of utmost importance to the operation and controls of a commercial MHD steam power system and thus also of the ETF MHD steam power system. Because of this, a brief discussion of the unique operational control characteristics of the combined MHD steam power system is first presented below followed by a more detailed description of start-up, shutdown, emergency and safety procedures for the various modes of facility operation.

1. 4. 4. 1 Operational Control

The MHD generator is a central and novel item of a combined MHD steam power system. Its operational characteristics, dynamic response and interfacing with the rest of the plant plays an important role in defining overall plant operation, controls and system dynamic behavior.

The main objectives of the coordinated plant control system are:

1. To operate the plant at any load condition at its optimum efficiency.
2. To provide for rapid response and load following characteristics.
3. To secure stable and reliable operation during steady state and all transient conditions and to protect and safeguard equipment and personnel.

4. To provide for start-up and shutdown of the plant according to specified equipment and plant requirements.
5. To provide for controlled operation at all operating modes during the various phases of the facility test program.

The dynamics of the MHD power system (high temperature air pre-heater, MHD combustor, MHD generator) and steam power system (boiler, turbogenerator and turbocompressor) and the coupling between these are important factors to be considered for development of a coordinated control system that provides for optimum plant operational performance, stability and response.

A thorough knowledge of the dynamics of the whole system, based on theoretical investigations and actual operation is essential to develop optimal plant operational modes and required controls and to realize the benefits to be derived from this. This remains yet to be studied and developed.

The control system structure developed in this conceptual design phase is based on the best understanding and available information of the overall process as of now. Therefore, the facility operating control system has also been designed with a high degree of flexibility. This will enable various control modes of operation to be field tested and functional reconfigurations to be implemented.

The basic requirements and functions that the control system must satisfy for controlled coordinated operation of the plant are identified on the system model sketch in Figure 1-14. This is a simplified model block diagram of a combined MHD steam power system. Primary controlled process variables, process inputs and outputs are indicated. The combined power system operation and coordinated controls are divided into its major subsystems. The system model sketch in Figure 1-14 serves the purpose of a simplified overall description of the plant operational controls.

A detailed description of the facility operating control system with its unit generation controls and associated subsystem controls are presented in Section 3.0. A detailed block diagram of the Generation Controls and Subsystems Controls is contained in that section. Primary functions of the plant control system are as follows.

Figure 1-14 MHD Simplified Model Block Control Diagram

The load demand signals emanate from the Generation Controls through comparison between required and actual generated power. Inherent in the load demand signals are data such as frequency bias, load limits, rates of change of load demand, runback information, start-up and shutdown commands and contingency action.

The MHD combustor firing rate is adjusted in accordance with desired total generation demand. However, the fuel (and seed) input to the MHD combustor must be controlled in accordance with the air input from the turbocompressor and thus coordinated with the turbocompressor control system. The proper stoichiometric ratio of coal and air in the combustor is adjusted by measurement of the CO/CO₂ content in the MHD generator exhaust gas. The seed flow rate can be adjusted by measurement of the gas conductivity. The possible measurement of gas conductivity is discussed in Section 1.5.2 in connection with description of the MHD channel instrumentation.

The electrical loading (load parameter) of the channel can be adjusted in a predetermined manner through its inverter controls (firing angle) by measurements of channel voltages and currents. The desired value of the channel load parameter (E_{load}/E_{oc}) is then assumed to be established in advance through theoretical calculations and prior field testing for all load conditions (mass flow rates). Ideally, the optimum control system would be a control system that would be capable of self-optimization. In simplified terms this could be conceived of as always seeking to adjust the channel loading parameter for maximum power output at each load condition after deduction of its required compressor power. ($P_{net\ MHD} = P_{MHD} - P_{compressor}$).

The separate fuel, its necessary combustion air and recycle gas inputs to the high temperature air preheater are adjusted in accordance with the MHD combustion air input. The controlled variables for the high temperature air preheater combustion system are air preheat temperature (fuel input), O₂ - trim (combustion air) and dome temperature (recycle gas).

The boiler controls utilizes a standard three element feed water control system with adjustment of feed water flow by drum level. The

boiler has oil burners for separate oil firing during start-up with their separate combustion control system. This is not shown on the simplified systems model sketch in Figure 1-14. In combined operation with the MHD generator, the secondary air introduced for afterburning of the MHD generator exhaust gas in the boiler is trimmed by O_2 - measurement of the boiler flue gas.

The steam turbine admission valves for the turbocompressor and turbogenerator are both adjusted from load demand and controlled by steam pressure. The important role of the turbocompressor in determining the firing rate and main energy input to the MHD combustor is emphasized.

The MHD generator has very rapid response characteristics. Thus, it will respond momentarily to changes in its inputs. However, the change in energy input to the MHD generator is brought about by simultaneous changes of fuel and combustion air inputs to the MHD combustor. Since the MHD combustion air is delivered from a turbocompressor which is part of the steam bottoming plant, the transient utilization of the boiler energy storage for first satisfying the increased turbocompressor power requirement (increased MHD combustion air demand) will enhance the total responsiveness of the system to demands for more power output.

The important dynamic coupling between the MHD generator and its cycle compressor is duplicated by the selection of an axial flow type compressor with operational characteristics similar to those projected for commercial power plant applications.

1. 4. 4. 2 Plant Operation

Upon completion of preliminary operation and testing of equipment components, the plant will be ready for normal plant-type operations. This section defines the operational requirements of the plant as a whole. Detailed operational requirements of the plant components as well as the control and instrumentation requirements are included in each of the respective component sections elsewhere in this report.

The following modes and phases of operation are described in this section.

Unit Support Auxiliaries Start-up

Unit Cold Start-up

Unit Warm Start-up
Unit Hot Start-up
Unit Normal Shutdown
Unit Emergency Shutdown
MHD Components Independent Operation
MHD ETF Failure Analysis

Operation of the high temperature air heaters described in this Section is based on the use of No. 2 fuel oil as the fuel source. The high temperature air heaters are also planned to be operated eventually with low Btu gas as fuel produced from an on-site gasifier.

Unit Support Auxiliaries Start-up

Prior to anticipated plant start-up, the following unit support auxiliaries must be started up or verified as operating.

- a. Physically inspect plant and equipment for start-up readiness
- b. Instrument air system operating
- c. Service air system operating
- d. Fire protection system functional
- e. Service water system operating
- f. Water treatment system operating
- g. Wastewater treatment system operating
- h. Energize electrical power supply systems required for plant operation
- i. Energize and checkout electronic control systems and safety interlock systems
- j. Start-up fuel oil forwarding system

At this point, the plant is ready for the continued start-up procedure.

Unit Cold Start-up

A unit cold start-up is defined as the high temperature air heater at or near ambient temperature, the magnet de-energized and the bottoming plant at cold conditions. Following is a description of the start-up sequence required for the major equipment components.

START DAY

High temperature air heaters (HTAH). The HTAH requires a total of twenty-nine (29) days to complete its start-up from cold condition assuming the preheater has been completely dried out. For the initial start-up or after subsequent reline, an additional 6 day dryout period during which the refractory would be heated to 250°F would be required. The start-up sequence is as follows:

- a. Verify cooling water systems in service
- b. Position valves as required for HTAH heat soak
- c. Establish purge of HTAH
- d. Start small auxiliary burners maintaining a 100F per day temperature rise to a dome temperature of 900°F

DAY 9

- a. Start air blower and flue gas recirculation blower
- b. Start main oil burners and extinguish auxiliary burners when their capability has been exceeded
- c. Continue HTAH heat soak at a rate of 100F per day temperature rise

DAY 25

High Temperature Air Heaters

- a. Continue at required heat soak rate

Auxiliary Steam Turbine

- a. Verify lube oil coolers in service
- b. Start-up lube oil system
- c. Place turbine on turning gear

Main Steam Turbine

- a. Verify lube oil coolers in service
- b. Start-up lube oil system
- c. Place turbine on turning gear

AC Generator

- a. Start seal oil system if not in service
- b. Fill generator with hydrogen to operating pressure as required
- c. Verify hydrogen coolers ready for service

DAY 28

Condensate System

- a. Establish safe operating level in condenser hotwell
- b. Start a condensate pump

Low Pressure Auxiliary Boiler

- a. Start low pressure auxiliary boiler

Deaerating Heater

- a. Admit peaking steam from low pressure steam system
- b. Place control systems on automatic

Boiler Feed Pump

- a. Start boiler feed pump and allow to recirculate on low flow supplying feedwater to boilers as required

High Pressure Auxiliary Boiler

- a. Prepare High Pressure Auxiliary Boiler for Start-up
- b. Start and warm-up boiler

DAY 29

Condenser

- a. Verify condenser ready for service
- b. Start-up circulating water system
- c. Establish steam seal system for both main and auxiliary steam turbines
- d. Place steam jet air ejectors in service

Auxiliary Steam Turbine

- a. Start turbine heating
- b. Roll turbine and accelerate to loading speed

Air Compressor

- a. Place air compressor in service and accelerate to minimum load, blowing off until loading is required

High Temperature Air Heaters

- a. After twenty-nine (29) days the HTAH should have obtained a heat soak temperature of 3000°F
- b. The HTAH is now ready to be placed in service

DAY 30

MHD Channel and Magnet

- a. Insure that cooling water systems are in operation
- b. Verify magnet cyrogenic systems are at correct operating conditions
- c. Start energizing magnet. Twenty-four (24) hours are required to completely energize magnet

Steam Generator (Boiler)

- a. Establish proper drum water level
- b. Start boiler circulation
- c. Set vent and drain valves in start-up positions
- d. Prepare boiler for service
- e. Start induced draft fan and maintain furnace pressure

Integrated Unit Warm-up

- a. Start admitting high pressure air through the HTAH into the MHD combustor through the MHD component train and bottoming plant at a rate not to exceed an increase in refractory temperature of 100°F per hour. Limit the primary radiant furnace outlet temperature to 1000°F until steam flow is established through the superheaters. Curves showing the start-up heating requirements of the refractory for both the MHD component train ductwork and the steam generator are shown on Figure 1-15 and Figure 1-16.
- b. Increase the total air flow through the boiler to 30% of MCR by allowing secondary air flow through the burner windboxes. Hold the 30% air flow rate for 5 min to establish furnace purge.
- c. After purge has been established, begin oil firing in the primary radiant furnace controlling the firing rate to the required refractory warm-up rate.
- d. Prepare coal processing system for start-up. Charge the petro-carb injectors, keeping the system pressurized with inert flue gas.

Figure 1-15 Warm-Up Schedule for HTAH Ducting

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Figure 1-16 Warm-Up Schedule for Steam Generator for Start-Up with MHD System

- e. Continue heating up duct refractory and increasing oil firing at the required rate allowing steam to bypass the main steam turbine into the condenser.
- f. When steam pressure and temperatures are suitable, warm-up and roll main steam turbine.
- g. Accelerate main steam turbine to operating speed and synchronize AC generator. Bring to minimum load of about 5 MW.
- h. When steam conditions permit, switch the compressor drive turbine from the auxiliary boiler steam to main steam.
- i. Continue to increase combustor air flow to meet duct heating requirements. Control temperature rise in the boiler by oil firing. Allow the turbine generator to pick up load as steam conditions permit.
- j. Approximately thirty (30) hr after initial start of duct warming using high temperature air flow, the refractory warm-up heat soak will be completed.
- k. Reduce air flow through the combustor to twenty-five (25) per cent for initial coal firing conditions. Allow the boiler combustion control system to maintain the air flow and oil firing as required for the bottoming plant.
- l. Start combustor slag removal system.
- m. Turn on the electro static precipitators.
- n. Begin feed coal to the primary combustor allowing the boiler warm-up oil guns to cutback to prevent excessive temperatures in the bottoming plant.
- o. Continue to increase coal firing and decrease warm-up oil gun firing until the oil guns can be removed from service. Deactivated oil guns are protected by cooling with inert flue gas.
- p. Start seed injection. Synchronize MHD generation as conditions allow.
- q. Slowly reduce main combustor air-fuel ratio to the required sub-stoichiometric levels. Simultaneously increase

secondary air flow to the final oxidation furnace, completing combustion in an excess air condition.

- r. Allow the unit to stabilize at minimum generation rate and verify that required control systems are in the automatic mode.

Completion of Cold Start-up

- a. The cold start-up is complete at this stage. The estimated time for the cold start-up to stable minimum generation is $30\frac{1}{2}$ days.
- b. Start-up seed regeneration system.
- c. Start-up bottoming plant ash removal system.

Plant Loading

- a. Increase plant generation loading as required by dispatch, keeping within the temperature ramp rates of the main steam turbine and AC generator which is approximately 3% load per minute.

Unit Warm Start-up

A unit warm start-up will be the type of start-up most frequently encountered and is defined by the criteria that the HTAH is at operating temperature. The magnet is off and the duct refractory is at less than operating temperature. Essentially it is assumed that the status of operating equipment is equivalent to start of day 30 of the unit cold start-up. Following is a brief description of the start-up sequence for the warm start-up. Details for some of the start-up requirements are defined in the cold start-up description.

Pre-Start Conditions

High Temperature Air Heaters at operating conditions

High Pressure Auxiliary Boiler operating

Auxiliary Steam Turbine and Air Compressor operating at unloaded condition

Air and Gas Blowers operating

Main Steam Turbine and AC Generator on turning gear ready for operation

Condensate system operating

Condenser and circulating Water System are operating

Feedwater System operating

Cooling Water Systems operating

Start - (Time Zero)

MHD Channel and Magnet

- a. Insure that cooling water systems are in operation
- b. Verify magnet cryogenic systems are at correct operating conditions
- c. Start the twenty-four hour energizing sequence for the magnet

Steam Generator (Boiler)

- a. Start procedure for the boiler to be the same as cold start-up requirements

Integrated Unit Warm-up

- a. The integrated unit warm-up shall be accomplished with the same procedure as stated in the cold start-up procedure. The length of time required can be shortened one hour for each 100°F the ductwork and boiler refractory are above ambient temperature.

Completion of Warm Start

- a. The warm start-up shall be completed in 24 to 30 hours dependent on residual temperature of the refractory
- b. Start seed regeneration system
- c. Start-up ash removal system

Plant Loading

- a. Increase plant generation loading as required by dispatch, keeping within the temperature ramp rates of the main steam turbine and AC generator which is approximately 8% load/min

Unit Hot Start-up

A unit hot start-up is the result of a shutdown for a relatively short period of time. The criteria for determining a hot start-up condition is the HTAH is at operating temperature, the magnet is fully

charged, the ductwork and boiler refractory temperatures are greater than ambient and the main steam turbine and AC generator are in a hot standby condition. Prior to hot start-up, the auxiliary systems shall have the following status.

- Condensate System operating
- Feedwater System operating
- Condenser and circulating water system operating
- High pressure auxiliary boiler operating
- Auxiliary steam turbine and air compressor operating at unloaded condition
- Coal injection system on standby
- Seed injection system on standby
- Main boiler in hot standby condition with recirculating pumps running

Following is a sequence of operations required for a hot unit start-up.

Start (Time Zero)

Integrated Unit Hot Start-up

- a. Start induced draft fan
- b. Start admitting hot air into the combustor. Increase air flow to 30% of MCR monitoring both ductwork and boiler refractory temperature rise.
- c. Allow for boiler purge. Then start windbox oil burners and continue to raise boiler temperature.
- d. By-pass generated steam around the turbine to the condenser until the steam temperature conditions match the turbine and bring up to operating speed. Synchronize and load up to 10%.
- e. Continue unit warm-up increasing air flow and boiler oil firing. Increase main steam turbine load as steam conditions allow.
- f. When steam conditions permit, transfer auxiliary turbine from the auxiliary boiler steam to main steam. Shut down auxiliary boiler.

- g. After required heating air flow has been attained and heat soak has been completed, reduce combustor air flow to 30% of MCR.
- h. Turn on electrostatic precipitators.
- i. Start combustor slag removal system.
- j. Begin coal feed to primary combustor. Allow the boiler warm-up oil guns to cutback, preventing excessive temperatures in the bottoming plant.
- k. Continue increasing coal firing and decreasing warm-up oil gun firing until the oil guns can be removed from service. The deactivated oil guns are to be protected by cooling with inert flue gas.
- l. Start seed injection. Synchronize MHD generator and load as conditions allow.
- m. Slowly reduce the main combustor air-fuel ratio to the required sub-stoichiometric levels. Simultaneously increase secondary air flow to the final oxidation furnace to complete combustion in excess air conditions.
- n. Allow the unit to stabilize at minimum generation rate and verify that required control systems are in the automatic mode.

Completion of Hot Start-up

- a. The hot start-up at this stage is complete. The estimated time for the hot start-up is dependent on the initial conditions. Time to stabilize minimum generation should be from two to twenty-four hours.
- b. Start-up seed regeneration system.
- c. Start-up ash removal system.

Plant Loading

- a. The plant generation can be increased at a rapid rate as the steam turbine metal temperature rise is within the turbine manufacturer's requirements of approximately 13% load/min.

Unit Normal Shutdown

A unit normal shutdown is defined as shutdown which can be conducted in an orderly, organized manner at the discretion of the operator. Prior to the shutdown, the unit including the primary combustor, MHD generator components, bottoming plant, HTAH and related components are operating at some load above minimum. The procedure for a normal shutdown is described as follows:

Pre-Shutdown Requirements

- a. Start auxiliary boiler and allow to operate in a standby mode.

Start (Time Zero)

- a. Reduce load to minimum stable load by ramping down primary combustor coal firing rate and air flow.
- b. Slowly increase air-fuel ratio to stoichiometric level and then to excess air conditions.
- c. Set the secondary air flow to suitable shutdown level.
- d. Allow adequate time with excess air operation to ensure that the MHD components and bottoming plant are purged of combustibles.
- e. Simultaneously ramp down coal firing rate and seed injection, monitoring temperatures and keeping them within the allowable temperature requirements of the refractory for both the MHD components and primary radiant furnace.
- f. When seed injection has been reduced to zero, establish ignition on windbox oil burners to help control boiler refractory temperature during further coal firing rate reduction.
- g. Ramp down coal firing rate to zero flow as MHD components and bottoming plant conditions allow.
- h. Shutdown electrostatic precipitators.
- i. Switch auxiliary steam turbine to high pressure auxiliary boiler steam as conditions require.
- j. Decrease oil firing, ramping down main steam turbine load to zero, at which time the turbine can be tripped off the line. The turbine speed is then allowed to coast down to the point at which the turbine can be placed on turning gear.

- k. Start boiler circulation system as required.
- l. Continue to decrease oil firing at a controlled rate until the burners can be shut down.
- m. After oil burners are shutdown, continue preheated air flow through the combustor and MHD component train into the bottoming plant for post-shutdown purge. Upon purge completion stop air flow.
- n. Shut down induced draft fan.
- o. Open boiler vents and drains as required.
- p. Shut down boiler circulation system as conditions permit.
- q. Close all dampers placing plant in "bottle-up" condition.

Post Shutdown Requirements

Upon completion of the normal shutdown, subsequent events are determined by the requirements initiating the shutdown which could include normal plant maintenance, forced maintenance of a particular piece of equipment due to some malfunction or requirements for specific equipment component modifications. These events may require further shutdown and/or cooldown of some portion of equipment or the entire plant. An example of this would be a major rebuild of the high temperature air heaters resulting in complete plant cool down with the removal of most plant equipment from service. Whereas modifications to the coal feed system may allow most of the plant components to remain in a hot standby "Bottled Up" mode.

Unit Emergency Shutdown

A unit emergency shutdown is the result of an event which requires an accelerated shutdown and could possibly result in some equipment or component damage. Following is a discussion of some of the events which would result in an emergency shutdown, and what actions are to be taken to lessen the severity of the resultant action and what further actions are required to restore the plant to full service or effect a complete unit shutdown. It has been assumed that each of the events being discussed occurs at some load above minimum unit load.

Main Fuel Trip

A Main Fuel Trip or failure of the coal injection system would cause an immediate stop to coal combustion. This situation could result in the possibility of thermal shock to the downstream components

Actions Required

- a. Initiate runback of induced draft fan to compensate for high negative furnace pressure
- b. Stop seed injection
- c. Allow MHD train and bottoming plant to purge then ignite oil burners to control temperatures in boiler
- d. Make decision to shutdown unit or to reestablish coal firing

Primary Air Trip

A Primary Air Trip would be the result of a major malfunction in the auxiliary steam turbine or the air compressor. This would cause not only a complete loss of primary air but also the loss of coal combustion which would result in excessive negative pressure in the Primary Radiant Furnace as well as allow the existence of combustibles in the bottoming plant.

Action Required

- a. Initiate runback of the Induced draft fan to compensate for high negative furnace pressure
- b. Increase inert gas flow to Primary Radiant Furnace to aid in preventing the negative pressure excursion
- c. Close secondary combustion air dampers
- d. Shutdown electrostatic precipitators
- e. Trip seed injection system
- f. The total system must be purged before a restart can be attempted

Seed Injection System Trip

A malfunction in the seed injection system causing the loss of seed feed would result in the immediate termination of MHD power generation in the channel. At MCR this would result in a loss of approximately 40 MW of enthalpy extraction with a corresponding increase in exhaust gas

temperature entering the bottoming plant. Sonic flow velocity would also develop in the MHD channel with a normal shock system located in some region of the diffuser.

Action Required:

- a. Increase flow of inert flue gas in oil gun windbox into the Primary Radiant Furnace to aid in controlling temperatures.
- b. Runback coal firing rate and increase air-fuel ratio as required to control temperature requirements.
- c. Once temperatures are stabilized at the required control level, the plant can be kept running until seed feed can be restored or the plant can be shutdown as circumstances dictate.

ID Fan Trip

An ID Fan Trip could cause possible overpressure in the bottoming plant which is essentially designed for balanced draft operation.

Action Required:

- a. Immediately reduce coal firing rate.
- b. Fully open ID Fan Dampers.
- c. Begin firing with excess air as soon as possible.
- d. Shutdown electrostatic precipitator to prevent ignition of combustibles.
- e. Stop seed flow when conditions permit.
- f. Terminate coal firing.
- g. Purge system of combustibles and reduce air flow to zero.
- h. Restart or continue plant shutdown.

Main Steam Turbine or AC Generator Trip

A malfunction in either the Main Steam Turbine or AC Generator resulting in a trip of either requires an immediate runback of coal firing rate. The runback will be of a magnitude to allow steam flow generated by the bottoming plant to be used by the Auxiliary Steam Turbine and bypassed to the condenser. When stability has been achieved and the malfunction can be corrected, the Main Steam Turbine can be reset and re-synchronized then reloaded. If the malfunction cannot be corrected, an orderly plan shutdown should be initiated.

Boiler Feed Pump Trip

A malfunction causing a Boiler Feed Pump trip will result in the immediate loss of primary cooling for the MHD Burner and diffuser, loss of boiler feedwater and ensuing loss of steam generation for Main and Auxiliary Steam Turbines. The Boiler Feed Pump trip should be considered to require the same emergency shutdown procedure as the primary air trip. In addition, immediate bypassing of the boiler feed pump must be initiated to provide cooling water to the MHD Burner and Diffuser.

Loss of Last Condensate Pump

A malfunction causing the loss of the last condensate pump will result in the immediate loss of cooling water to the MHD Generator and subsequent loss of boiler feedwater. Coal feed, primary air flow, seed injection and Main Steam Turbine should be tripped immediately. An emergency shutdown procedure similar to that for the primary air trip should be initiated to minimize possible damage.

1.4.4.2.1 Independent Operation of MHD System Components and Bottoming Plant

During the various stages of the test plan, there will be requirements to operate the MHD system separately from the bottoming plant or the bottoming plant separately from the MHD system. Following is the description of both of these two modes of operation.

Separate MHD System Operation

The separate operation of the MHD System requires a start-up similar to a typical total unit start-up except for the exclusion of the steam generator and the main steam turbine and generator. Following is a brief outline of the required start-up procedure. For a more detailed operational start-up of the various pieces of equipment, see the preceding unit start-up procedures.

- a. Start or verify operating the unit support auxiliaries.
- b. Warm-up HTAH as required.
- c. Start-up condensate and feedwater systems.
- d. Start-up auxiliary boiler.
- e. Establish condenser vacuum.

- f. Start-up auxiliary steam turbine and air compressor.
- g. Prepare the coal processing system for service.
- h. Place the gas bypass scrubber in service.
- i. Start heating ductwork refractory in accordance with the unit start-up requirements.
- j. Energize magnet if required.
- k. Start-up slag removal system.
- l. Upon completion of warm-up, start coal and seed injection at combustor.
- m. Load MHD generator as required.

Separate Bottoming Plant Operation

The steam generator and steam-turbine generator can be operated when the MHD components are not in service. The steam generator with its own combustion system can achieve ~ 30% of MCR. Following is a brief outline of the start-up procedure for the bottoming plant only.

- a. Start or verify operating the unit support auxiliaries.
- b. Start-up condensate, feedwater and circulating water systems.
- c. Start-up auxiliary boiler and establish condenser vacuum.
- d. Establish boiler recirculation systems.
- e. Start FD and ID fans.
- f. Establish furnish purge.
- g. Initiate oil gun firing upon purge completion.
- h. Warm-up steam generator in accordance with Figure 1-17 "Steam Generator Transients for Independent Start-up."
- i. Allow steam to bypass turbine to the condenser.
- j. Place steam turbine-generator in operation if required when steam conditions are acceptable.

The steam generator can be at maximum 30% MCR capacity in ~ 21 hours.

1.4.4.2.2 MHD ETF Failure Analysis

Following is a failure analysis table listing the types of failures which could occur in principal plant components. The table lists the type of failures which could occur, the cause of the failure, method of failure detection, subsequent effects and what corrective action must be taken.

In the event that a failure or malfunction occurs, the plant operating personnel must make the decision on whether or not a plant shutdown is required, or they must monitor the sequence of automatic events initiated by the plant safety interlock system.

Figure 1-17 Steam Generator Transients for Independent Start-Up

MHD ETF Failure Analysis

Component	Failure	Cause	Subsequent Effect	Method of Detection	Corrective Action
Channel	Internal water Leak	Seal or joint failure Excessive erosion of electrode or side walls	Loss in power output	Coolant flow rate monitoring. MHD power monitoring	Shut down and repair if leakage is excessive. If not excessive repair at next scheduled shutdown.
	External water leak	Failure of hose or connector.	Possible overheating and burnout of channel at channel wall	Coolant flow rate and pressure monitoring Channel temp monitoring.	Shut down and repair.
	Loss or reduction of coolant flow.	Leak in coolant supply system. Condensate pump trip or failure.	Overheating and burnout of channel wall.	Coolant flow rate and pressure monitoring.	Shut down and repair.
	External gas leak.	Structural failure. Burn-out of wall. Channel over-pressure.	Possible damage to power and instrumentation cables, magnet dewar.	IR scanner or gas sampling system in bore of dewar.	Shut down and repair.
	Electrical short-circuits.	Cable insulation failure. Contact with grounded surface. Foreign material.	Electrical arcing resulting in possible damage to channel structure or magnet.	Channel instrumentation monitoring.	Shut down to correct fault.
	Severe reduction or loss of seed.	Malfunction in seed injection system.	Reduction or loss of MHD power output.	MHD power output monitoring.	Operate at reduced firing rate to determine cause of malfunction. Repair if possible during operation or shut down to repair.
	Excessive inter-electrode shorting.	Degradation of inter-electrode insulators.	Non-uniform channel voltage distribution.	Channel instrumentation monitoring.	Redistribution of channel loading.
Consolidation Circuitry	Commutation Failure	Loss of Gate to SCR or SCR failure.	Extreme electrode current non-uniformities.	Monitor commutation capacitor voltages.	Attempt to restart shut down and repair or switch to redundant circuit.
	Short to ground.	Cable or transformer insulation failure.	Current overload in channel fire hazard.	Monitor channel instrumentation.	Shut down and repair
Channel Inlet Nozzle	Cooling tube leakage.	Excessive material erosion. Material over-temperature.	Loss in MHD power if leakage excessive.	Coolant flow rate and pressure monitoring. MHD power monitoring.	Shut down and repair.
Diffuser	Cooling tube leakage.	Excessive material erosion Material overtemp.	Loss in MHD power if leakage excessive.	Coolant flow rate and pressure monitoring. MHD power monitoring.	Shut down and repair if leakage excessive. If not severe repair at next scheduled shutdown.

MHD ETF Failure Analysis

Component	Failure	Cause	Subsequent Effect	Method of Detection	Corrective Action
Combustor	Cooling tube leakage.	Erosion, corrosion, weld failure, material over temperature.	Possible burnout of outer shell. Reduction in MHD power output.	Coolant flow rate and pressure monitoring. Shell temperature monitoring. MHD power monitoring	Shut down and repair or replace faulty tube
	Fuel injector blockage or burnout.	Insufficient coolant flow. resulting in fuel caking. Failure of coolant supply. Leakage in coolant supply line.	Reduction in MHD power output.	Coolant flow rate and pressure monitoring. Fuel flow rate and pressure monitoring. MHD power output monitoring.	Shut down to examine coolant supply circuitry and repair fault. Inspect injectors and clear or replace it if necessary.
	Seed injector blockage or burnout	Insufficient coolant flow. resulting in seed caking. Failure of coolant supply. Leakage in coolant supply line.	Reduction in MHD power output.	Coolant flow rate and pressure monitoring. Seed flow rate and pressure monitoring. MHD power output monitoring.	Shut down to examine coolant supply circuitry and repair fault. Inspect injectors and clear or replace it if necessary.
	Loss of refractory in inlet manifold	Erosion, thermal fracture, spalling.	Possible burnout of outer shell. Possible blockage of primary air injectors. Decrease in combustor performance. Reduction in MHD power output.	Manifold shell temperature monitoring. Primary air flow rate monitoring. Exhaust gas analysis. MHD power output monitoring.	Shut down and repair or replace refractory if severe fault. If conditions permit, repair at next planned shutdown.
	Partial or complete blockage of slag removal part.	Excessive slag build-up on walls from large number of stop-start cycles. Malfunction of slag removal system.	Excessive slag carry-over into MHD channel and downstream slag removal equipment.	Temperature monitoring in region of slag port. Decrease in slag removal rate.	Shutdown and clear slag port. Repair slag removal system if required.
	Electrical isolator.	Dirt, water, or other foreign material on dielectric insulators.	High voltage on upstream components.	Voltage monitoring on isolator sensing plate.	Shutdown and remove foreign material or replace dielectric.

MHD ETF Failure Analysis - Superconducting Magnet System

<u>Component</u>	<u>Failure</u>	<u>Cause</u>	<u>Subsequent Effects</u>	<u>Method of Detection</u>	<u>Corrective Action</u>
Vacuum System	Forepump	Pump or water failure; low oil level; forepump vacuum line breakage, etc.	Reduced efficiency of diffusion pumps. Possible build up of He gas in vacuum tank with increased load on refrigeration system.	Vacuum gauge or visual inspection.	Valve off offending pump and operate on second pump till repaired.
	Diffusion Pump	Low oil level, heater failure, joint leakage.	Same as above.	Vacuum gauge or pump boiler temperature.	Valve off offending pump, discharge magnet; maintain vacuum till repaired.
	Vacuum Gauge	Electric coil or mechanical failure.	None unless vacuum is destroyed.	Loss of vacuum in a part of system or a discrepancy between gauges.	Valve off offending portion and replace gauge when convenient provided vacuum can be maintained.
	Main Vacuum Tank Leak	Gasket failure or cooling line failure within tank.	Condensation on coldest structural parts.	Vacuum gauge and optical inspection system interruption.	Discharge magnet. Warm up system and repair.
Refrigeration System	Compressor Interrupt	Motor, mechanical or control failure.	Loss of refrigeration if continued for a long time.	Pressure gauge.	Repair. If shut down will be more than 1 day shut down magnet and warm up.
	Cold Box	Blockage or mechanical failure	Idem	Idem	Idem
	Transfer Lines	Mechanical failure	Loss of refrigeration	Gauges or visual inspection	May require magnet shut down and warm up depending on location of failure.
	Helium Storage Tank	Idem	Idem	Idem	Shut down magnet, warm up and repair.

MHD ETF Failure Analysis - Superconducting Magnet System

<u>Component</u>	<u>Failure</u>	<u>Cause</u>	<u>Subsequent Effects</u>	<u>Method of Detection</u>	<u>Corrective Action</u>
Magnet Power Supply	Power Leads	Idem	Magnet quench if shunt diodes are disconnected.	Magnet voltage.	Shut down power supply and repair.
	Shunt Diode Open Circuit	Burn out or mechanical failure.	No effect if only one of the group. Magnet quench if many.	Periodic inspection.	Repair at normal maintenance period if only one.
	Shunt Diode Short Circuit	Idem	Magnet discharges slowly.	Magnet voltage.	Replace or repair.
	Main Power Supply	Idem	Idem	Idem	Idem
Superconducting Magnet	Power Lead cooling not effective	Blockage or mechanical failure.	Low magnetic field. Possible safety circuit trip.	Voltage drop or visual inspection.	Shut down magnet, warm up and repair.
	Winding Open Circuit	Mechanical failure.	Magnet quench.	Magnet instrumentation.	Idem
	Winding Short Circuit	Idem	Low magnetic field.	Voltage drop or charging rate.	Idem
	Magnet instrumentation voltage taps, strain gauges, magnetic field gauge, or thermometers	Idem	Anxiety	Discrepancy between various readings.	Repair at normal maintenance period.
	Leakage of helium container or helium ducts	Idem	Loss of vacuum in vacuum tank.	Vacuum gauges.	Shut down magnet, warm up and repair.
	Leakage of cooling ducts in radiation shield	Idem	Idem	Vacuum gauges.	Idem
	Obstruction of J T expansion valve in vacuum tank	Blockage or mechanical failures.	Inefficient or loss of cooling.	Pressure gauges.	Idem

MHD ETF Failure Analysis

Component	Failure	Cause	Subsequent Effect	Method of Detection	Corrective Action
Regenerative Air Preheater	Checker work and combustion chambers side wall insulation	Over temperature, corrosion, erosion, thermal stress, mechanical failure	Increased heat loss, over temperature of shell, excessive thermal stress at matrix periphery	Shell temperature	Shut down and repair damaged refractory
	Checker work and combustion chambers dome insulation	Over temperature, corrosion, erosion, thermal stress, mechanical failure	Increased heat loss, over temperature of shell, plugging of matrix flow passages leading to increased pressure drop and possible reduction in fluid flow rates, damage to reheat burner, decreased air preheat	Shell temperature, visual inspection, pressure drop	Shut down and repair damaged refractory
	Ducting insulation	Over temperature, corrosion, erosion, thermal stress, mechanical failure	Increased heat loss, over temperature of piping, increased pressure drop	Shell temperature, pressure drop	If insulation is on inside of pipe, shut down and repair damaged refractory. If insulation is on outside of pipe, repair as required
	Refractory matrix	Over temperature, corrosion, erosion, thermal stress, mechanical failure	Plugging of matrix flow passages leading to increased pressure drop and possible reduction in fluid flow rates, decreased air preheat	Pressure drop, preheat temperature, combustion gas temperature	Shut down and install new matrix if pressure drop becomes excessive
	Matrix support	Over temperature, corrosion, mechanical failure	Misalignment of matrix flow passages leading to increased pressure drop, possible failure of entire matrix	Pressure drop, support temperature, visual inspection	Shut down and repair matrix support
	Checker work and combustion chamber pressure vessels	Over temperature, stress corrosion, over pressure	Loss of pressure, reduction in cycle air and/or reheat of combustion gas flow, over temperature of matrix and/or insulation	Visual inspection	Shut down and repair vessel shells
	Ducting pipe	Over temperature, over pressure, mechanical failure, stress corrosion	Loss of pressure, reduction in cycle air and/or reheat combustion gas flow	Visual inspection	Shut down and repair piping
	Isolation valving	Over temperature, over pressure, mechanical failure, stress corrosion	Leakage between fluids causing increased cycle air compressor power requirements	Cooling water temperature rise, mass flow rate, pressure drop	Shut down and repair and/or replace valve
	Reheat burner	Over temperature, over pressure, fouling, control problem, pulsations	Loss of reheat combustion gas flow, decreased air preheat	Visual inspection, pressure drop, temperature	Shut down and repair and/or replace burner

MHD ETF Failure Analysis

<u>Component</u>	<u>Failure</u>	<u>Cause</u>	<u>Subsequent Effects</u>	<u>Method of Detection</u>	<u>Corrective Action</u>
Boiler	Loss of Refractory from Water-Walls	Corrosion, erosion; materials problem; thermal stress.	•Excess heat absorption.	•Wall temp. monitoring. •Visual inspection.	•Shut down and purge boiler. •Repair or replace refractory.
	Slag Buildup	Slag tap plugging	•Possible structural damage. •Increased lower wall buildup. •Possible refractory damage.	•Slag removal rate. •Visual inspection.	•Shut down and purge boiler. •Free slag tap.
	Header or Tube Leakage	Materials; erosion, corrosion, or over-temps.	•Reduction in coolant flow. •Possible tube failure.	•Coolant flow and pressure monitoring. •Visual inspection.	•Shut down and repair.
	Boiler Circ. Pump or Pump Driver Failure	Hardware problem or control problem	•Reduction in coolant flow. •Possible tube failure.	•Circ. pump trip in control room. •Coolant flow monitoring.	•Shut down and repair or replace pump or driver. •Check control logic if driver cuts out.

MHD ETF Failure Analysis

<u>Component</u>	<u>Failure</u>	<u>Cause</u>	<u>Subsequent Effects</u>	<u>Method of Detection</u>	<u>Corrective Action</u>
Economizer	Tube or Header Failure	Materials problem; corrosion, erosion or overtemp.	<ul style="list-style-type: none"> • Loss of downstream cooling capacity with loss of flow. • Introduction of coolant to gas stream with wet mixture to ESP and ID fan; subsequent corrosion likely. 	<ul style="list-style-type: none"> • Economizer outlet pressure. • Visual inspection during planned outage. 	<ul style="list-style-type: none"> • Shut down and repair.
	Reduced Heat Transfer Effectiveness	Excessive fouling or plugging.	<ul style="list-style-type: none"> • Increased gas outlet temp. • Decreased economizer outlet temp. • Increased gas pressure drop with subsequent increase in fan power requirement. 	<ul style="list-style-type: none"> • Gas pressure monitoring. • Economizer temperature monitoring. 	<ul style="list-style-type: none"> • Increased usage of sootblowers. • Manual cleaning after shutdown.
	FW Stop Valve Malfunction	Materials or controls problem.	<ul style="list-style-type: none"> • Possible reduction in downstream cooling capacity with component overtemps and possible failures. 	<ul style="list-style-type: none"> • Flow meter. 	<ul style="list-style-type: none"> • Cut back to condition consistent with valve position if possible. • Check control logic to determine cause. • Shut down and replace or repair.

MHD ETF Failure Analysis

<u>Component</u>	<u>Failure</u>	<u>Cause</u>	<u>Subsequent Effects</u>	<u>Method of Detection</u>	<u>Corrective Action</u>
Superheater	Header or Tube Failure	Materials problem; corrosion, erosion or overtemp.	<ul style="list-style-type: none"> • Loss of, or reduction in steam flow. • Overtemp. of downstream components; also, probable failure of additional superheater sections. 	<ul style="list-style-type: none"> • Steam temperature and flow monitoring. • Visual inspection during planned outage. 	<ul style="list-style-type: none"> • Shut down and replace.
	Reduced Heat Transfer Effectiveness	Plugging or fouling.	<ul style="list-style-type: none"> • Drop in steam conditions. • Rise in downstream gas temperatures with possible component overtemps. • Increase in gas pressure drop with increase in fan power requirement. 	<ul style="list-style-type: none"> • Gas pressure monitoring. • Steam temperature monitoring. 	<ul style="list-style-type: none"> • Increased use of sootblowers. • Shut down and manually clean if above is ineffective.
	Main Steam Stop Valve Malfunction	Materials or controls problem.	<ul style="list-style-type: none"> • Fail closed (part): loss of or reduction in steam flow. • Fail open (part): loss of drum pressure; subsequent turbine overspeed. 	<ul style="list-style-type: none"> • Steam temperature and pressure monitoring. 	<ul style="list-style-type: none"> • Adjust firing rate to condition consistent with valve position. • Check control logic to determine possible cause. • Shut down and replace or repair.

MHD ETF Failure Analysis

<u>Component</u>	<u>Failure</u>	<u>Cause</u>	<u>Subsequent Effects</u>	<u>Method of Detection</u>	<u>Corrective Action</u>
Superheater (Cont'd.)	Desh. Failure	Materials or controls problem; loss of desh. flow.	• Improper SH conditions. • Possible overtemp. of finishing SH sections.	• Steam temp. and pressure monitoring. • Desh. flow monitoring.	• Adjust firing rate to condition consistent with max. allowable S/T inlet temp. • Replace and repair at next planned shutdown.

MHD ETF Failure Analysis

<u>Component</u>	<u>Failure</u>	<u>Cause</u>	<u>Subsequent Effects</u>	<u>Method of Detection</u>	<u>Corrective Action</u>
LTAH	Loss of Air	Failure of blower, blower drive, or ducting.	<ul style="list-style-type: none"> •Overtemp. of LTAH air sections. •Loss of HTAH function. •Loss of burnout capability. •Increase in gas outlet temp. with possible overtemp. of downstream components. 	<ul style="list-style-type: none"> •LTAH flow monitoring. •LTAH temp. or press. monitoring. •Motor control trip in control room. 	<ul style="list-style-type: none"> •Cut back MHD flow. •Shut down, or bypass MHD stream until load is cut back. •Cut back HTAH oil flow. •Check control logic to determine if cause. •Repair or replace failed equipment.
	Loss of Flow Recycle Gas	Failure of blower, blower drive, or ducting.	<ul style="list-style-type: none"> •Overtemp. of LTAH recycle gas section. •Reduction in HTAH performance due to reduced mass flow rate. •Possible overtemp. of downstream components. 	<ul style="list-style-type: none"> •LTAH flow monitoring. •LTAH temp. or poss. monitoring. •Motor control trip in control room. 	<ul style="list-style-type: none"> •Cut back flow thru boiler. •Bypass MHD stream. •Isolate HTAH so blower and ductwork can be worked on.
	Tube Failure	Materials	<ul style="list-style-type: none"> •Reduction of affected gas to HTAH with reduction in HTAH performance. •Probable increase in outlet temp. of the affected gas if leak is significant. 	<ul style="list-style-type: none"> •LTAH flow, temp., and pressure monitoring. 	<ul style="list-style-type: none"> •Go through standard procedures to take boiler off-line and shut down boiler. Repair or replace affected parts.

MHD ETF Failure Analysis

<u>Component</u>	<u>Failure</u>	<u>Cause</u>	<u>Subsequent Effects</u>	<u>Method of Detection</u>	<u>Corrective Action</u>
LTAH (Cont'd.)	Decrease in Heat Transfer Effectiveness	Fouling or Plugging	<ul style="list-style-type: none"> • Insufficient air/gas preheat to HTAH. • Increased outlet temp. for main gas stream. 	<ul style="list-style-type: none"> • Gas pressure monitoring. 	<ul style="list-style-type: none"> • Increase usage of sootblowers. • If severe or persistent, shut down and manually clean.
	HTAH WHR Recuperative Heat Exchanger Leakage	Corrosion, erosion; materials.	<ul style="list-style-type: none"> • Leakage of air or recycle gas to environment or to stack gas stream. • Possible inadequate flow to LTAH with subsequent LTAH overtemp. 	<ul style="list-style-type: none"> • Temperature monitors. • Damper settings. • Visual observation. 	<ul style="list-style-type: none"> • May not require system shutdown depending on severity of leak. • Repair as able.

MHD ETF Failure Analysis

<u>Component</u>	<u>Failure</u>	<u>Cause</u>	<u>Subsequent Effects</u>	<u>Method of Detection</u>	<u>Corrective Action</u>
ESP	Loss of Charging or Collecting Capacity	Electrical failure within ESP.	•Excessive particulates to ID fan and stack.	•Control room indicator. •Stack gas sampling.	•Shut down system and repair.
	Plugging of Collection Hoppers	Excessive solids accumulation.	•Inability to remove additional solids.	•Solids removal rate. •Level indicator.	•Shut down and remove blockage.
	Wire Failure	Mechanical.	•Decreased collection effectiveness.	•Visual inspection.	•Repair at planned shutdown.
	Explosive Mechanical Failure	Loss of secondary air flow; fuel-rich mixture to ESP along with air leakage.	•Loss of MHD gas stream to ambient. •Possible propagation of explosion effects back through boiler.	•Gas pressure indicators. •Visual observation.	•Shut down and repair or replace.

MHD ETF Failure Analysis

<u>Component</u>	<u>Failure</u>	<u>Cause</u>	<u>Subsequent Effects</u>	<u>Method of Detection</u>	<u>Corrective Action</u>
ID Fan	Loss of Flow	Motor or fan failure.	<ul style="list-style-type: none"> • Loss of exhaust suction with increased back pressure. • Possible component overtemps. and failures. • Gas leakage from furnace. 	<ul style="list-style-type: none"> • Gas pressure monitoring. • Motor circuit indicator in control room. 	<ul style="list-style-type: none"> • Shut down and repair or replace. • Check motor controls. • Check for structural damage.
	Inadequate Suction Pressure	Improper vane settings or failure of control mechanisms.	<ul style="list-style-type: none"> • Increased back pressure and reduced gas velocity thru boiler. • Possible overtemps. 	<ul style="list-style-type: none"> • Gas pressure monitoring. 	<ul style="list-style-type: none"> • Check control logic to determine possible malfunction. • At next planned shutdown, reestablish correct operating conditions - repair or replace vanes or vane controls.

MHD ETF Failure Analysis

<u>Component</u>	<u>Failure</u>	<u>Cause</u>	<u>Subsequent Effects</u>	<u>Method of Detection</u>	<u>Corrective Action</u>
Coal Processing System: Mill	Drive Motor Shutdown	Controls breaker or winding failure	<ul style="list-style-type: none"> •Pulverizer stops turning. •Coal buildup in feeder and mill. •Loss of coal feed to storage bin. 	•Mill ammeter.	<ul style="list-style-type: none"> •Stop hot gas flow to mill. •Stop ID fan, stop coal feed. •Shut cyclone air lock. •Inert gas to mill. •Restart and clear as able.
Mill	Drying Gas Flow Stoppage	Upstream failure in HTAH.	<ul style="list-style-type: none"> •Flow of coal to cyclone stops. •Negative pressure in mill from ID fan. 	<ul style="list-style-type: none"> •Mill ammeter. •Mill gas exit temperature. •ID fan ammeter and controls. 	<ul style="list-style-type: none"> •Stop coal feed. •Stop ID fan. •Stop mill, shut cyclone air lock. •Start inert gas to mill. •Restart and clear as able.
Mill	Fire/Explosion	High oxygen content in drying gas. Excess temperature.	•Fire could spread to cyclone, baghouse and/or storage bin.	•Temperature	<ul style="list-style-type: none"> •Reduce gas temperature to mill. •Extinguish baghouse and storage bin. •Shut down and inspect for damages as soon as fire put out.
Mill	Low Temperature Drying Gas from HTAH	Upstream failure in HTAH or WHR system.	•CPS unable to meet drying specifications.	•Mill temperature monitor.	<ul style="list-style-type: none"> •Shut down CPS until adequate flow restored. •Restart.

<u>Component</u>	<u>Failure</u>	<u>Cause</u>	<u>Subsequent Effects</u>	<u>Method of Detection</u>	<u>Corrective Action</u>
Storage Bin	Unable to Deliver Coal to Petrocarb	Extended mill outage. Vibrator failure. Scalping screen failure.	<ul style="list-style-type: none"> • Loss of coal flow to Petrocarb. 	<ul style="list-style-type: none"> • Storage bin load cells. • Petrocarb storage injector load cells. • Visual observation. 	<ul style="list-style-type: none"> • Correct failure and restart as able.
Baghouse	Fire/Explosion	Excess oxygen or temperature buildup.	<ul style="list-style-type: none"> • Flames exit explosion vents, damage to baghouse bags. • Extinguisher system in operation. 	<ul style="list-style-type: none"> • Visual observation of baghouse. • Baghouse temp. monitor. 	<ul style="list-style-type: none"> • CPS shutdown required to assess damage and locate cause. • Restart as able.
Petrocarb System	One or More Injector Lines Stops Coal Delivery	Control or ejector failure. Wet coal clogging.	<ul style="list-style-type: none"> • Proportional loss of coal feed to MHD combustor. 	<ul style="list-style-type: none"> • Petrocarb controls. 	<ul style="list-style-type: none"> • Correct cause, clear and restart.
Gravimetric Feeder	Shutdown	Component or control failure.	<ul style="list-style-type: none"> • Coal flow to mill stopped. 	<ul style="list-style-type: none"> • Coal flow rate monitor. 	<ul style="list-style-type: none"> • Correct and restart as able.
Coal and Gas Ducting	Pressure Integrity (leak)	Erosion, corrosion, weld failure, accidental damage.	<ul style="list-style-type: none"> • Depending on relative pressure air leaks in, coal leaks out or gas leaks out. • Possibility of fire or explosion. 	<ul style="list-style-type: none"> • Visual observation, fan and damper control settings. 	<ul style="list-style-type: none"> • CPS system shutdown and repair. Restart as able.
Cyclone	Airlock Failure	Jammed valve.	<ul style="list-style-type: none"> • Unable to isolate or drain cyclone during purge or filtering processes. • Coal buildup in cyclone, air enters cyclone. 	<ul style="list-style-type: none"> • Valve controls. 	<ul style="list-style-type: none"> • Free valve and restart system.

<u>Component</u>	<u>Failure</u>	<u>Cause</u>	<u>Subsequent Effects</u>	<u>Method of Detection</u>	<u>Corrective Action</u>
Inert Gas Compressor	Reduced or No Flow	Excess oxygen content trips control. HTAH shutdown.	<ul style="list-style-type: none"> • Loss of pressurizing gas to Petrocarb injector vessels. 	<ul style="list-style-type: none"> • Storage bin unable to be recharged. • Pressure falls in primary injector. • Coal flow to combustor is reduced rapidly. 	<ul style="list-style-type: none"> • Restore gas flow, restart as able.
ID Fan	Drive Motor Shutdown	Control failure motor failure.	<ul style="list-style-type: none"> • Drying gas flow severely reduced. • Coal flow to cyclone severely reduced. • Positive pressure in mill. 	<ul style="list-style-type: none"> • CPS ID fan ammeter and controls. 	<ul style="list-style-type: none"> • Stop coal feed, clear mill. • Reduce gas flow from HTAH. • Restart as able.
Overall System	High Oxygen Content in Drying Gas	Upstream failure in HTAH. Inlet leakage.	<ul style="list-style-type: none"> • Increased possibility of fire and explosion in mill, cyclone, baghouse, etc. 	<ul style="list-style-type: none"> • Oxygen monitor. 	<ul style="list-style-type: none"> • Cut coal flow and clear mill. • Shut cyclone air lock. • Reduce gas temperature. • Correct problem and restart.

MHD ETF FAILURE ANALYSIS

<u>COMPONENT</u>	<u>FAILURE</u>	<u>CAUSE</u>	<u>SUBSEQUENT EFFECTS</u>	<u>METHOD OF DETECTION</u>	<u>CORRECTIVE ACTION</u>
Steam Turbine Generator (STG)	Excessive Vibration; Thrust brg. movement excessive.	Brg.. failure. Improper Startup.	Possible Equipment damage.	Turbine Supervisory instrumentation.	STG trip ¹ . Shutdown and repair.
	Lube oil high temp.	Loss of cooling water; decreased oil flow; excess heat.	Possible equipment damage.	Bearing drain temperature.	Alarm. shutdown and correct.
	Lube oil pressure low.	Oil pump failure; Line blockage.	Possible equipment damage.	Oil pressure monitor.	Automatic initiation of backup oil pump. Possible shutdown.
	High back pressure or high exhaust temperature.	Loss or malfunction of circ. water (CW) or vacuum systems,	Excessive heating in turbine exhaust.	Back pressure and exhaust temp. monitors.	STG trip ¹ . Shutdown and repair
	Water induction.	Extraction line check valve malfunction.	Possible structural damage.	Differential temp. measurements.	STG trip ¹ . Shutdown and repair.
	Control oil pressure.	Oil pump failure; System blockage.	Loss of EHC system.	Oil pressure monitor.	Automatic initiation of backup pump. STG trip ¹ If problem persists. Shutdown and repair.

(1) See Unit Emergency Shutdown, Section 1.4.4.2

MHD ETF FAILURE ANALYSIS

<u>COMPONENTS</u>	<u>FAILURE</u>	<u>CAUSE</u>	<u>SUBSEQUENT EFFECTS</u>	<u>METHOD OF DETECTION</u>	<u>CORRECTIVE ACTION</u>
Turbine-Generator	Loss of DC control power.	DC power distribution system trouble.	Loss of EHC system.		STG trip ¹ . Shutdown and repair.
	Electrical fault.	Unit phase diff. High, ground fault. Loss of excitation.	Breakdown of electrical insulation or internal damage.	Generator instrumentation.	STG trip ¹ . Shutdown and repair.
	Loss of SJAE.	Loss of steam. Ejector malfunction.	Gradual increase in back pressure to non-condensibles.	Back pressure monitor.	Controlled shutdown. Correct problem.
	Loss of generator	Cooling water sys. failure. Loss of hydrogen.	Generator temperature rise.	Generator instrumentation.	Controlled shutdown. Correct problem.
Condensate System	Loss of one condensate pump.	Motor or pump failure.	Loss of cooling to MHD generator and flow available to BFP.	Control room monitoring.	Unit runback. Repair problem.
	Loss of all operating condensate pumps.	Motor or pump failures.	Loss of cooling to condensate and feedwater circuits.	Control room monitoring.	Unit trip ¹ . Shutdown and repair.
	Extreme low deaerator level.	Loss of condensate pumps.	Loss of cooling to condensate and feedwater circuits.	DA storage tank level control.	Unit trip ¹ . Shutdown and repair.
	Loss of BFP lube oil.	Lube oil system failure.	Possible pump damage.	Lube oil system pressure temperature monitors.	BFP trip ¹ .
	Loss of two condensate pumps.	System blackout.	Loss of cooling to condensate and feedwater circuits.	Control room monitoring.	Auto. start-up of emergency diesel gen. to restore 1 condensate pump.

(1) See Unit Emergency Trip, Section 1.4.4.2

MHD EFT FAILURE ANALYSIS

<u>COMPONENTS</u>	<u>FAILURE</u>	<u>CAUSE</u>	<u>SUBSEQUENT EFFECTS</u>	<u>METHOD OF DETECTION</u>	<u>CORRECTIVE ACTION</u>
Air Compressor	High bearing and lube oil temp.	Loss of cooling water; low oil flow; excess heat.	Possible equipment damage.	Bearing drain temperature.	Air compressor trip ¹ .
	Rotor vibrations.	Brg. or material failure.	Possible equipment damage.	Control room monitoring.	Air compressor trip ¹ .
	Excessive discharge temperature.	No flow.	Possible equipment damage.	Control room monitoring.	Air compressor trip ¹ .
	Low bearing lube oil pressure.	Oil pump failure; Line blockage.	Possible equipment damage.	Bearing oil pump pressure monitor.	Automatic initiation of backup oil pump. Air compressor trip ¹ if problem persists.
	High inlet pressure loss.	Blocked filters.	Reduced air flow - possible equipment damage.	Control room monitoring.	Operator decision to correct problem.
Air Compressor Turbine Drive	Lube oil low pressure or high temperature.	Oil pump failure; Line blockage.	Possible equipment damage.	Same as Air Compressor	Lube oil system is common to air compressor. Corrective actions are the same.
	Excessive vibration; Thrust bearing movement excessive.	Bearing failure.	Possible structural damage.	Turbine supervisory instrumentation.	Turbine trip ¹

(1) See Unit Emergency Shutdown, Section 1.4.4.2

MHD ETF FAILURE ANALYSIS

<u>COMPONENT</u>	<u>FAILURE</u>	<u>CAUSE</u>	<u>SUBSEQUENT EFFECTS</u>	<u>METHOD OF DETECTION</u>	<u>CORRECTIVE ACTION</u>
Air Compressor Turbine Drive	High backpressure; High exhaust temperature.	Loss of malfunction of circulating water (CW) or vacuum systems.	Excessive heating in turbine exhaust.	Backpressure and exhaust temp. monitors.	STG trip ¹ . Shutdown and repair.
	Loss of STAE.	Loss of steam; Ejector malfunction.	Gradual increase in backpressure due to non-con- densibles.	Backpressure monitor.	Controlled shutdown. Correct problem.

(1) See Unit Emergency Shutdown, Section 1.4.4.2

MDH ETF FAILURE ANALYSIS

<u>COMPONENTS</u>	<u>FAILURE</u>	<u>CAUSE</u>	<u>SUBSEQUENT EFFECTS</u>	<u>METHOD OF DETECTION</u>	<u>CORRECTIVE ACTION</u>
Circulating Water System	Loss of one circulating water (CW) pump.	Pump or electrical problems.	Rising turbine back pressure.	Control room monitoring.	Unit runback. Repair problem.
	Loss of two CW pumps.	Pump or electrical problems.	Loss of condenser vacuum.	Control room monitoring.	Unit trip. Repair problem.
	Loss of two cooling water pumps.	Pump or electrical problems.	Loss of cooling to auxiliary equipment.	Temperature monitoring of individual auxiliary equipment.	Unit trip. Repair problem.
	Loss of cooling tower capability.	Fan failure.	Increased CW temp.	Control room monitoring.	Unit trip for 100% cooling tower loss. Unit runback for partial cooling tower loss. Repair defective cooling tower cell.
Instrument Air Compressors	Loss of compressors.	Compressor or electrical problems.	Loss of air to pneumatic controls.	Control room monitoring.	Controlled shutdown. Repair problem.
DC Power Supply	Loss of DC power.	Electrical or physical failure.	Loss of DC central system and emergency motor power supplies.	DC power system monitors.	Unit trip.
Slag Flyash and Seed Collection	Loss of ash sluice pump, motive air sources, eductors or other system removal equipment.	Pump, blower or electrical problems; mechanical equipment failure.	Possible filling of available storage bins or silos.	Control room or local monitoring.	Repair problem while storage volumes are available. If available storage volumes are filled, shutdown and repair.

MHD ETF FAILURE ANALYSIS

<u>COMPONENT</u>	<u>FAILURE</u>	<u>CAUSE</u>	<u>SUBSEQUENT EFFECTS</u>	<u>METHOD OF DETECTION</u>	<u>CORRECTIVE ACTION</u>
Seed Feed System	Loss of all or some seed.	Control pressure; blocked lines.	Increase of gas temp. to boiler; Loss of MHD power.	Control room monitoring.	Unit runback ¹ .

(1) See Unit Emergency Shutdown, Section 1.4.4.2

MHD ETP FAILURE ANALYSIS
SEED REGENERATION PLANT

<u>COMPONENT</u>	<u>FAILURE OR MALFUNCTION</u>	<u>CAUSES</u>	<u>SUBSEQUENT EFFECTS</u>	<u>METHODS OF DETECTION</u>	<u>CORRECTIVE ACTION</u>	<u>REMARKS</u>
<u>SILO AIRVEYORS</u> RECOVERED SEED BALLAST LIME STORAGE COKE STORAGE FORMATE BALLAST FORMATE STORAGE	LOW FLOW	BLOWER MALFUNCTION SILO BLOCKAGE PIPE LINE BLOCKAGE LEVEL CONTROL MALFUNCTION	STARVE GRAVIMETRIC FEEDERS	FEEDER LOW LIMIT ALARM	REPAIR BLOWER UNBLOCK SILO MANUALLY CLEAR PIPELINES MANUALLY CORRECT LEVEL CONTROL MALFUNCTION OF FINALLY SHUT DOWN SRP	MANUALLY LOAD FEEDER HOPPERS DURING CORRECTIVE ACTION. DISSOLVER BALLAST IS 1 HOUR
	EXCESSIVE FLOW	LEVEL CONTROL MALFUNCTION	FLOOD GRAVIMETRIC FEEDER HOPPERS	FEEDER HOPPER HIGH LEVEL ALARM	DEENERGIZE AIRVEYOR BLOWER REPAIR LEVEL CONTROL	OPERATE AIRVEYOR CONTROLS MANUALLY DURING REPAIR
SPRAY DRIER	LOW FLOW	BLOWER MALFUNCTION, PIPELINE BLOCKAGE	DEPLETION OF BALLAST SILO	CYCLONE HIGH OR LOW LEVEL ALARM	DUMP CYCLONE AND REPAIR MALFUNCTION	OPERATE PETROCARB SYSTEM FROM FORMATE STORAGE SILO DURING REPAIR
<u>GRAVIMETRIC FEEDERS</u> RECOVERED SEED LIME COKE	LOW DISCHARGE	SILO AIRVEYOR FAILURE FEEDER MALFUNCTION	LOW PRODUCT YIELD LOW PRODUCT YIELD	FEEDER LOW LIMIT ALARM FEEDER LOW LIMIT ALARM; RISE IN T OF FLUE GAS EXITING SPRAY DRIER	SEE ABOVE SHUT DOWN RECOVERED SEED FEEDER REPAIR FEEDER WITHIN 1 HOUR OR SHUT DOWN SRP	DISSOLVER BALLAST IS 1 HOUR GASIFIER COKE RESERVE IS 200 HOURS
	EXCESSIVE DISCHARGE	FEEDER MALFUNCTION	OVERCHARGE DISSOLVER OR FLOOD GASIFIER FEED HOPPER	FEEDER HIGH LIMIT ALARM DISSOLVER OVERFLOW TO SEWER; COKE BUCKET ELEVATOR JAM	SHUT DOWN RECOVERED SEED FEEDER REPAIR FEEDER WITHIN 1 HOUR OR SHUT DOWN SRP	DISSOLVER BALLAST IS 1 HOUR GASIFIER COKE RESERVE IS 200 HOURS
<u>PUMPS</u> REACTOR FEED	LOW FLOW	STARVED SUCTION PLUGGED DISCHARGE PUMP MALFUNCTION	LOW PRODUCT YIELD	INCREASED CO CONCENTRATION IN REACTOR OFF GAS; RISE IN T OF FLUE GAS EXITING SPRAY DRIER	CORRECT DISSOLVER OPERATION, UNBLOCK SUCTION LINE, REPAIR PUMP WITHIN 1 HOUR OR SHUT DOWN SRP	REACTOR BALLAST IS 1 HOUR
FILTER FEED	LOW FLOW	STARVED SUCTION PLUGGED DISCHARGE PUMP MALFUNCTION	LOW PRODUCT YIELD	DECREASED PRESSURE DROP ACROSS FILTER; RISE IN T OF FLUE GAS EXITING SPRAY DRIER	CLEAR LINES OR REPAIR PUMP WITHIN X MINUTES OR SHUT DOWN SRP	FLASH TANK (1) BALLAST IS X MINUTES
SPRAY DRIER RECYCLE FEED	LOW FLOW	STARVED SUCTION PUMP MALFUNCTION	LOW PRODUCT YIELD FILTER FLOODS OR BYPASSES TO FILTER DISCHARGE STORAGE TANK (NOT SHOWN)	RISE IN T OF FLUE GAS EXITING SPRAY DRIER; DROP IN SOLUTE CONCENTRATION OF REACTOR FEED	CHECK FILTER; REPAIR PUMP WITHIN Y MINUTES OR SHUT DOWN SRP	FLASH TANK (2) BALLAST IS Y MINUTES

NOTE: SRP = Seed Regeneration Plant TRC = Temperature Recorder Controller FRC = Flow Recorder Controller

MID ETP FAILURE ANALYSIS
SEED REGENERATION PLANT

COMPONENT	FAILURE OR MALFUNCTION	CAUSES	SUBSEQUENT EFFECTS	METHODS OF DETECTION	CORRECTIVE ACTIONS	REMARKS
<u>PUMPS (CONTD)</u>						
GAS COOLER	LOW FLOW	NO WATER SUPPLY; TEMP. CONTROL MALFUNCTION PUMP MALFUNCTION	COMPRESSOR CUTS OFF; CO RESERVE CUTS IN; GASIFIER VENTS TO BOTTOMING PLANT OR FLARE; K LOSS IN FILTER CAKE DIS- CHARGE; PRODUCT YIELD DROPS	HIGH TEMP. ALARM IN GAS COOLER GAS DIS- CHARGE TRC; LOW FLOW IN FILTER WASH WATER;	ENERGIZE BYPASS SUPPLY OF SERVICE WATER (NOT SHOWN) TO FILTER WASH SYSTEM; REACTIVATE WATER SUPPLY TO COOL- ER; REPAIR TEMP. CON- TROLLER OR PUMP	IF REPAIR NOT PROMPT, DEENERGIZE COKE FEEDER AND GASIFIER
CO ₂ SCRUBBER	LOW FLOW	LOSS OF SCRUBBING SOLUTION (DIETHANOLA- MINE); PUMP MALFUNC- TION	CO ₂ IN GAS FEEDS TO TO REACTOR CAUSING SCALING AND/OR CaCO ₃ FORMATION	CO ANALYZER ALARM	REPLACE SCRUBBING SOL- UTION; REPAIR ANY LEAKS; REPAIR PUMP	
<u>FANS & BLOWERS</u>						
GASIFIER AIR SUPPLY	LOW FLOW	FAN MALFUNCTION	GASIFIER FIRE BOX FLAMEOUT; COKE FEEDER CUTS OUT COMPRESSOR CUTS OFF COMMERCIAL CO RESERVE CUTS IN	LOW TEMP. ALARM IN GASIFIER DISCHARGE TRC	REPAIR FAN; RE- ENERGIZE COKE FEEDER AND GASIFIER OPERA- TION	
SPRAY DRIER	LOW FLOW	FLUE GAS SUPPLY LOSS; FAN MALFUNCTION	WET FORMATE PRODUCT	LOW TEMP. ALARM IN FLUE GAS TRC;	SUBSTITUTE AIR FOR FLUE GAS; REACTIVATE FLUE GAS SUPPLY; REPAIR FAN	
<u>DISSOLVER</u>						
	SOLIDS BUILDUP	MIXER MALFUNCTION; LIME OR SEED FEEDER MALFUNCTION; SHORTAGE IN WATER FEED; CASCADE CONTROL MALFUNCTION	REACTOR FEED PUMP PLUGS; REACTOR FEED SHORT- AGE	MIXER MOTOR ALARM; FEEDER HIGH FLOW ALARMS; REACTOR FEED PUMP CUTOUT; TEST POINT Δ ; LOW FLOW ALARM IN WATER SUPPLY FRC	REPAIR MIXER; CORRECT FEEDER FLOWS; CORRECT WATER FEED; CORRECT CASCADE CONTROL; OR FINALLY SHUT DOWN SRP	REACTOR BALLAST IS 1 HOUR
	EXCESSIVE DILUTION	LIME OR SEED FEEDER MALFUNCTION; WATER FEED CONTROL MALFUNCTION	LOW PRODUCT YIELD	FEEDER LOW FLOW ALARM; TEST POINT Δ ; DIS- SOLVER HIGH LEVEL INDICATOR (NOT SHOWN) ALARM	CORRECT FEEDER FLOWS; CORRECT WATER FEED CASCADE CONTROL; OR FINALLY SHUT DOWN SRP	REACTOR BALLAST IS 1 HOUR
<u>GASIFIER</u>						
	LOW CO OUTPUT; FLAMEOUT	COKE FEEDER MAL- FUNCTIONS; AIR FEED MALFUNCTION	LOW PRODUCT YIELD	CO ANALYZER ALARM	ENERGIZE COMMERCIAL CO RESERVE TANK FLOW; COR- RECT COKE FEEDER OR GASIFIER FAN MALFUNC- TION; RE-ENERGIZE GAS- IFIER	

MHD ETP FAILURE ANALYSIS
SEED REGENERATION PLANT

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<u>COMPONENT</u>	<u>FAILURE OR MALFUNCTION</u>	<u>CAUSES</u>	<u>SUBSEQUENT EFFECTS</u>	<u>METHODS OF DETECTION</u>	<u>CORRECTION ACTIONS</u>	<u>REMARKS</u>
<u>GASIFIER (CONTD.)</u>	GAS LEAK	STRUCTURAL FAILURE	TOXIC BUILDING AIR	BUILDING CO DETECTOR (NOT SHOWN) ALARMS	CLEAR BUILDING; EMERGENCY AIR FLOOD TO BUILDING; EMERGENCY SHUT DOWN OF GASIFIER; ENERGIZE COMMERCIAL CO FLOW; REPAIR LEAK; ENERGIZE GASIFIER	
	HIGH OUTSIDE WALL TEMP.	REFRACTORY FAILURE	GASIFIER SHUTDOWN;	GASIFIER INSTRUMENTATION	ENERGIZE COMMERCIAL CO RESERVE TANK FLOW; REPLACE REFRACTORY; REENERGIZE GASIFIER	NORMAL REPLACEMENT ONCE PER OPERATING YEAR
<u>COMPRESSOR</u>	LOW OUTPUT PRESSURE OR CUTOFF	MACHINE FUNCTION	LOSS OF MANUFACTURED CO PRESSURE; GASIFIER SHUTDOWN	COMPRESSOR INSTRUMENTATION	ENERGIZE COMMERCIAL CO RESERVE TANK FLOW; REPAIR COMPRESSOR; RE-ENERGIZE GASIFIER	CO RESERVE WILL LAST 50 HOURS
<u>CO₂ REMOVAL SYSTEM</u>	LOW STRIPPING LOW ABSORPTION	STEAM SUPPLY CON- STRICTED LOSS OF AMINE SCRUB- ING SOLUTION	CaCO ₃ FORMATION IN REACTOR OR SCALING IN REACTOR	CO ANALYZER	BYPASS SCRUBBING SYS- TEM; CLEAR STEAM LINE; REFLESH DIETHANOLA- MINE SOLUTION	
<u>REACTOR</u>	INTERNAL FLOW REDUCTION	CaSO ₄ SALT BUILD- UP DUE TO HEMI- HYDRATE FORMATION; MIXER NOT EFFECTIVE	REDUCED MASS FLOW OUT OF REACTOR	OVERLOAD ON MIXER MOTOR;	SHUTDOWN SRP; CLEAR REACTOR; ESTABLISH NEW OPERATING CONDI- TIONS; REPAIR MIXER	PRE ETP SUPPORT DEVELOPMENT STUDIES SHOULD MINIMIZE THESE FAILURE MODES
	LOW OR NO REACTION	REACTION PARAMETERS; MIXER NOT EFFECTIVE	LOW PRODUCT YIELD	TEST POINTS Δ . Δ	SHUT DOWN SRP; CLEAR REACTOR; READJUST RE- ACTOR OPERATING PARA- METERS; REPAIR MIXER	
<u>FILTER</u>	LOW FILTRATE FLOW	SCREEN BLOCKAGE; FILTER COMPRESSOR MALFUNCTION	LOSS IN FILTRATE OUTPUT; STARVED SUCTION OF SPRAY DRIER FEED PUMP; LEVEL RISE IN FLASH TANK	CHANGED PRESSURE DROP ACROSS FILTER	SHUT DOWN SRP; ADJUST REACTOR OPERATING CON- DITIONS; REPAIR COMPRESSOR	
	SOLIDS IN FILTRATE	SCREEN BREAK	PRODUCT CONTAMINA- TION	TESTS POINTS Δ . Δ	SHUT DOWN SRP; REPAIR SCREEN	
<u>SPRAY DRIER</u>	LOW PRODUCT OUTPUT	PLUGGED FEED LINES; CYCLONE INOPERATIVE; ROTARY VALVE JAMMED; PLUGGED SPRAY DRIER OUTLET	LEVEL BUILDUP IN FLASH TANK; DUST COLLECTOR PLUGS; BUILDUP OF SOLIDS IN SPRAY DRIER	LOW LEVEL INDICATOR ALARM (NOT SHOWN) IN FORMATE BALLAST SILO	CLEAR FEED LINES; RE- PAIR CYCLONE; CLEAR ROTARY VALVE IN CY- CLONE; CLEAR SPRAY DRIER OUTLET; OR FIN- ALLY SHUT DOWN SRP	

END EKF FAILURE ANALYSIS
SEED REGENERATION PLANT

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<u>COMPONENTS</u>	<u>FAILURE OR MALFUNCTION</u>	<u>CAUSES</u>	<u>SUBSEQUENT EFFECTS</u>	<u>METHODS OF DETECTION</u>	<u>CORRECTION ACTIONS</u>	<u>REMARKS</u>
<u>SPRAY DRIER (CONTD.)</u>	WET PRODUCT	LOW FLUE GAS FRED TEMP.; REHEATER MALFUNCTION	PLUGGED FORMATE SILOS AND PNEUMATIC CONVEYOR LINES;	TEST POINT ; GAS DISCHARGE TPC; FLUE GAS TEMP.	REPAIR REHEATER; CHECK FLUE GAS SUPPLY CON- DITIONS	
INSTRUMENTATION	VARIOUS	VARIOUS HARDWARE MALFUNCTION	LOSS OF COMPONENT OR PLANT OPERATIONAL CONTROL	READOUTS; ALARMS	SWITCH TO MANUAL OPERA- TION IF POSSIBLE; REPAIR OR FINALLY SHUT DOWN SRP	

Inverter - See Section 2.16.1

1.5 ETF TEST PROGRAM

Summary

The objectives of the ETF Test Program are:

- To test and evaluate the performance, endurance, controls and operational characteristics of the various individual power system components and their interactions as part of an integrated power system.
- To test and evaluate the overall performance, operation and controls of a complete MHD/steam power system including part load operation and the dynamic response of the system to load variations as well as startup, shutdown, reliability and safety aspects.
- To identify and resolve critical problem areas.

The test schedule shown in Table 1-21 is arranged to provide a systematic and progressive test plan to achieve these objectives. The total time required to perform a comprehensive test and evaluation of the MHD components and the complete MHD/steam power plant is estimated to be 3 1/2 years. Testing and evaluation of components and the complete MHD/steam power system are first carried out with 2500°F air preheat temperature. After this the air preheat temperature is raised to its maximum design temperature of 3000°F. A 2000 hrs endurance test with 2500°F preheat temperature is completed within the first 1 3/4 year of the test program. With the achievement of this important milestone, the design phase of the first commercial MHD power plant is considered initiated.

An initial checkout test period is allocated as a first performance evaluation of the MHD components individually. These components are the HTAH system, coal combustor, MHD generator channel, superconducting magnet, and diffuser. To the extent possible, these components will initially be tested independently from one another except for those situations where the operation of one particular component is directly dependent upon the operation of one or more of the other components.

After the initial check test phase a series of tests will be conducted to evaluate the performance of the MHD topping plant independent of the steam generator bottoming plant. This will be accomplished by utilizing

TABLE 1-21

OVERALL ETF - MHD TEST PLAN

TEST ACTIVITY	YEAR 1 *	2	3	4
1. EVALUATION OF MHD SYSTEM IN BY-PASS MODE OPERATION WITH 2500°F AIR PREHEAT TEMP. (O ₂ ENRICHMENT AT REDUCED PREHEAT TEMP.)	—			
2. FLUSH AND CHECK-OUT STEAM BOTTOMING PLANT. CHECK-OUT BOP COMPONENTS AND SYSTEMS.	—			
3. CHECK-OUT OPERATION OF SEED REPROCESSING SYSTEM.	—			
4. INTEGRATE MHD SYSTEM WITH STEAM BOTTOMING PLANT AND SEED REPROCESSING SYSTEM.		—		
5. CHECK-OUT OF COMBINED MHD/STEAM CYCLE.		—		
6. 2000 HR. ENDURANCE TEST OF INTEGRATED SYSTEM.		—		
7. DETAILED INSPECTION OF ALL PLANT COMPONENTS.		—		
8. PARAMETRIC TESTING TO EVALUATE OVERALL PLANT PERFORMANCE CHARACTERISTICS.			—	
9. UPGRADE PLANT TO 3000°F AIR PREHEAT TEMPERATURE AND EVALUATE PERFORMANCE IN BY-PASS OPERATING MODE.			—	
10. INITIATE DESIGN OF COMMERCIAL MHD PLANT.		Δ		
11. 2000 HR. ENDURANCE TEST OF INTEGRATED SYSTEM AT 3000°F PREHEAT.			—	
12. INSPECTION AND PARAMETRIC TESTING.				—
13. CONVERT HTAH SYSTEM FOR USING COAL GASIFIER FUEL.				—
14. EXTENDED FACILITY OPERATION WITH COAL GASIFIERS.				—

* YEAR 1 OF THE TEST PROGRAM CORRESPONDS TO YEAR 6 OF THE PROJECT SCHEDULE SHOWN ON TABLE 1.6.3-1.

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the exhaust bypass system which accepts the hot exhaust gases directly from the diffuser discharge and provides the necessary quenching and scrubbing functions before the gases are emitted to the atmosphere. The exhaust bypass system permits progressive testing with increased degree of complexity in operation of the facility. Thus, with use of the exhaust bypass system the total MHD power train can be tested and characterized without the additional operational complexity imposed by the combined operation of this with the steam generator bottoming plant. Initial independent characterization of MHD component performance and operation will also contribute to safeguarding the bottoming plant against unpredictable behavior. The bypass mode operation will also facilitate initial maintenance procedures or modifications to any of the MHD components which may be indicated by the test results. The final test in the bypass mode with 2500°F air preheat will require ~1 year. During this period the steam bottoming plant will be flushed and thoroughly checked out preparatory to mating it with the topping plant. The seed recovery system will also be checked out to verify its satisfactory operation.

After the MHD and bottoming plants have been combined, approximately six months of testing will be conducted with the complete combined MHD/steam power system. This will first be directed primarily toward proper functioning and operation as a complete integrated power system. Following this, a 2000 hr endurance test at rated plant design conditions with 2500°F air preheat temperature will be conducted. Upon completion of this test, a detailed inspection of all plant components and systems will be carried out and appropriate maintenance functions will be performed.

Subsequent to the inspection period approximately four months of testing will be performed to evaluate overall plant performance over a wide range of load and parametric test conditions in order to obtain a comprehensive facility performance map. Up to this point of the test program all testing has been carried out at 2500°F air preheat temperature. The subsequent part of the test program is then devoted to testing and evaluation of MHD components and the complete MHD/steam power system at maximum air preheat temperature of 3000°F to demonstrate possible improvements in plant performance resulting from this maximum preheat temperature.

Testing at the highest preheat temperature of 3000°F is also conducted in a progressive manner with sequential testing of MHD components and the complete power system as before at 2500°F preheat temperature.

Subsequent activity will consist of converting the HTAH system to using LBTu fuel gas provided by the coal gasifier plant. Introduction of the gasifier is deliberately delayed until this stage of the test program to benefit from possible advances in gasifier technology which may be forthcoming from the ongoing development effort in this area. It is believed that the facility can thereafter be placed on line to furnish power to the grid or continue to serve as an engineering test facility to test and evaluate new or improved MHD components and system design concepts.

1.5.1 Pre-Operation Checkout

A thorough pre-operation checkout of all the facility components, auxiliaries, and subsystems will be conducted to establish their readiness for subsequent operational tests. This will consist of all those tasks which can be performed short of actual operation such as:

1. Flush and hydraulic test of cooling circuits.
2. Checkout of electrical power and control wiring.
3. Calibrate instrumentation and controls.
4. Proper operation of pumps, fans, and actuators.
5. Mechanical alignment.

Wherever possible, simulated operation will be employed. Test procedures provided by equipment suppliers or specified by onsite manufacturer's test personnel will be used in conducting these tests.

Scheduling of the pre-operation tests will be determined by the construction and equipment installation schedule and will proceed as soon as possible after all the necessary services for a given piece of equipment have been provided. It is estimated that a period of approximately six months toward the conclusion of the construction phase will be required for the performance of these tests.

1.5.2 Independent Testing of MHD System at 2500°F Preheat Temperature

As stated earlier the first phase of operational testing will be in the bypass mode, i. e., the exhaust gas from the MHD diffuser will not enter the steam bottoming plant but will be delivered to the stack via the bypass duct and

exhaust scrubber system. During the initial phase of this test period the performance of the individual MHD components will be evaluated. Therefore, the test schedule, as shown in Table 1-22, is arranged in a sequential fashion in order to complete the testing of those components which must be functionally operational to permit the testing of subsequent components. Detailed test specifications will be prepared to provide specific test procedures and evaluation criteria for each component or system. The test procedures presented in this document are merely preliminary and intended only to present a logical sequence of tasks which must be performed to evaluate individual component performance.

1.5.2.1 Checkout of S. C. Magnet and Support Systems

As shown in the test schedule, the first MHD component to be tested is the S. C. magnet and the supporting cryogenic refrigeration and vacuum systems. These tests will be performed prior to channel installation to obtain full access to the bore of the magnet. The tests to be performed are the following:

Refrigeration System

The refrigeration system will be bought from vendors who will install their equipment, instrumentation and controls. Checkout will consist of starting the equipment and verifying that it works properly.

Vacuum System

The same remarks apply to the vacuum system. Checkout of the main vacuum tank will consist of closing the tank, starting the pumps, and measuring the degree of vacuum achieved.

Magnet Instrumentation - Checkout

When the magnet is assembled, instruments such as voltage taps, strain gauges, thermocouples, and magnetic field probes will be incorporated. Some of these can be tested when the magnet is a room temperature and some during cool down. Instrumentation testing will proceed continuously during the magnet checkout.

Magnet Operation - Checkout

After these preliminaries, the magnet can be cooled and operated. During cool down, refrigeration will be supplied from the compressor/refrigerator. The instrumentation will be monitored to be sure that the

TABLE 1-22

TEST SCHEDULE FOR OPERATION IN BY-PASS MODE

TASK DESCRIPTION	MONTH 1	2	3	4	5	6	7	8	9	10	11	12
1 CHECK-OUT CRYOGENIC AND VACUUM SYSTEM, START-UP AND EVALUATE PERFORMANCE OF SC MAGNET												
2 INSTALL CHANNEL AND INSTRUMENTATION IN MAGNET												
3 CHECK OUT COOLING WATER CIRCUITS AND INSTRUMENTATION IN BURNER, CHANNEL AND DIFFUSER												
4 VALIDATE ELECTRICAL INTEGRITY OF CHANNEL, POWER CABLES, INSTRUMENTATION LEADS												
5 COLD FLOW TESTS THROUGH MHD FLOW TRAIN												
6 PRELIMINARY EVALUATION OF HTAH SYSTEM												
7 PREHEATED AIR FLOW TEST THROUGH MHD FLOW TRAIN												
8 CHECK-OUT BURNER OPERATION AT VARIOUS LEVELS OF MASS FLOW, STOICHIOMETRIC RATIOS, COAL MOISTURE CONTENT												
9 CHECK-OUT CHANNEL AND DIFFUSER WITH HOT GAS FLOW GAS CONDUCTIVITY MEASUREMENTS												
10 EVALUATE CHANNEL PERFORMANCE AT VARYING MAGNETIC FIELD WITH FULL MASS FLOW POWER TO RESISTIVE LOAD BANK AND/OR INVERTER SYSTEM												
11 EVALUATE SYSTEM PART LOAD PERFORMANCE - FULL MAGNETIC FIELD, VARYING MASS FLOW												
12 100 HR TEST OF COMPLETE SYSTEM AT DESIGN RATED POWER												

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cool down rate is proceeding in accordance with the schedule. Also, the strain gauges will be monitored to determine whether there is unexpected stress. When the magnet is finally cold the power supplies can be started. Charging will proceed in accordance with the initial charging plan and the magnet field measured by magnet field probes. Once again, the appearance of unusual conditions will be looked for. It is conceivable that the magnet will have to be partially charged, then discharged, and recharged to a higher level in several steps. This will only be required the first time and its purpose is to set the windings within the structure. After the magnet is fully charged, it can be discharged and recharged as often as required in accordance with the MHD test plan. The channel may be installed and removed without warming the magnet. Also, operation of the MHD channel at fields less than the maximum is reasonable and practical.

Schedule

Refrigeration, vacuum system and power supply checkout will be part of the procurement and installation of these purchased components. Magnet checkout will require 6 weeks including initial cool down. Testing will be done by the regular operating personnel and will be in charge of the magnet operator. Additional test personnel will be the chief magnet project engineer and chief magnet design engineer or designees for 5 weeks.

1.5.2.2 Evaluation of HTAH System

Checkout and evaluation of the HTAH system will begin during the first month of the operational test period. This work will be conducted concurrently with the S. C. magnet because it can be performed on a noninterference basis. The complexity of the HTAH system and the necessity for very carefully controlled drying and heat-up rates of the refractory matrix and insulation requires that a fairly extensive time period be allotted for the checkout and testing of this system. As shown on the schedule, the time period for this activity spans months 1 through 5. A detailed account of the checkout and startup procedure for the HTAH system is presented in Section 2.11.

In the event that unforeseen technical difficulties with the HTAH system limit the deliverable preheat temperature to some value below the 2500°F temperature required for the initial stages of the test program, the MHD

combustion air will be enriched with gaseous oxygen to compensate for the temperature deficit. Based upon prior high temperature operating experience this situation is unlikely to develop, however, the provision to employ oxygen enrichment is included in the facility design so that, should these difficulties arise, the other test activities can proceed according to schedule.

1.5.2.3 Checkout and Evaluation of Combustor

After the magnet tests have been completed, the channel can be installed to complete the MHD flow train which will permit initiation of the combustor tests. Consideration was given to using a dummy flow channel for the combustor tests rather than the generator channel but because of the relatively low risks involved it was believed that the added cost of a dummy channel or instrumented test channel was not warranted. The sequence of tests which will be conducted on the combustor are:

1. Calibrate fuel and seed injection systems.
2. Checkout cooling water circuitry.
 - a. Look for internal & external water leaks.
 - b. Ascertain rated flow exists in all coolant circuits.
 - c. Checkout instrumentation.
3. Cold flow tests.
 - a. Run unheated air through combustor at 40, 70, 100% of design flow.
 - b. Check for gas leaks and general mechanical integrity.
 - c. Checkout instrumentation.
4. Run with only preheated air at increasing mass flow rate during final stage of HTAH warm-up cycle.
 - a. Determine heat flux to combustor walls.
 - b. Validate proper functioning of cooling circuitry and I&C system.
5. Begin burner operation at 40% mass flow rate and gradually increase fuel, seed, and mass flow to design conditions.
 - a. Check for ignition and general combustion stability.
 - b. Check slagging characteristics and slag collection system.
 - c. Check heat flux to walls.
 - d. Determine overall performance from gas temperature and conductivity measurements in channel.
 - e. Determine burner performance over a range of stoichiometric ratios.
 - f. Determine effect of varying coal moisture content on overall burner performance.

1.5.2.4 Checkout and Performance Evaluation of MHD Generator Channel

The sequence of tests which will be conducted to checkout and evaluate the channel performance characteristics are those given below. The principal activity and the parameters to be investigated for each series of tests are indicated and discussed in the subsequent text.

Sequence of Channel Evaluation Tests

1. Checkout cooling water circuits.
 - a. Look for internal and external water leaks.
 - b. Ascertain rated flow exists in all coolant circuits.
 - c. Checkout instrumentation - flow meters, pressure sensors thermocouples, control elements.
2. Checkout electrical integrity of channel, power cables, instrumentation leads.
 - a. Validate proper hook-up of all channel power cables and instrumentation leads.
 - b. Determine that there are no ground loops in channel and all other wiring.
3. Cold flow tests.
 - a. Run unheated air through channel at 40, 70, 100% of rated flow.
 - b. Check for gas leaks and general mechanical integrity.
 - c. Checkout related instrumentation.
4. Run with only preheated air at various mass flow rates during final stage of HTAH warm-up cycle.
 - a. Determine heat flux to channel walls.
 - b. Validate proper functioning of cooling circuitry, I&C systems.
5. Run with MHD burner on but without power extraction (no magnetic field) at 40, 70, 100% of rated mass flow.
 - a. Determine burner performance by gas temperature and conductivity measurements.
 - b. Determine heat flux to channel walls.
 - c. Determine slagging characteristics of channel walls.
 - d. Evaluate diffuser performance.
 - e. Validate proper functioning of cooling circuitry and I&C systems.

6. Run at rated mass flow at various levels of magnetic field strength. Electrical power absorbed by resistive load bank.
 - a. Determine channel electrical characteristics.
 - b. Determine channel aerodynamic characteristics.
 - c. Determine wall heat flux variation with power generation.
 - d. Validate proper functioning of cooling circuitry and I&C systems.
 - e. Evaluate diffuser performance.
7. Run with rated magnetic field at 40, 60, 80, 100% of design mass flow. Electrical power initially absorbed by resistive load bank. Final tests with power to inverter system.
 - a. Determine part load performance characteristics.
 - b. Determine wall heat flux variation with power generation.
 - c. Validate proper functioning of cooling circuitry and I&C systems.
 - d. Evaluate diffuser performance.
 - e. Determine performance characteristics of inverter system and dynamic interaction with channel.
8. Run for 100 hrs at full design mass flow and channel output power.
 - a. Determine performance characteristics of all major MHD components under extended operation.
 - b. Determine channel endurance capabilities.

A number of important checkout procedures will be carried out on the channel after it has been placed inside the bore of the S. C. magnet prior to firing of the burner. These are indicated in the test procedures 1 through 4. In the next series of tests, Item 5, the channel will primarily be used to determine burner performance by measuring gas temperature and conductivity. The method of gas temperature measurement will be resolved during the detailed design phase but will probably be by optical means or water-cooled temperature probes. The gas conductivity will be determined by the conventional method of applying a dc potential across an axial segment of the channel. From measuring the potential gradient in this segment and the current flow, the conductivity is readily calculated from Ohm's law.

Also during this test series, measurements will be made to determine the heat flux to the channel walls and to evaluate the slagging characteristics of the walls. This period will also provide the opportunity to establish an adequate slag coating on the channel walls prior to the power extraction runs.

In step 6 of this test series, power will be generated by the MHD channel. The mass flow will be maintained at essentially 100% of design flow and the magnetic field will be increased in steps from a low level to the design level of 6 T peak. The power produced during this period will be absorbed by the resistive load bank to permit the application of discrete electrical loads on the MHD generator. This will permit a complete mapping of the channel performance characteristics. A portion of the power generated will be routed through the inversion system for preliminary checkout and evaluation of the individual inverter circuits. The channel electrical characteristics to be evaluated include power output, voltage and current distribution as a function of load, current concentrations, and excessive interelectrode voltage. The aerodynamic characteristics will be determined from the measured pressure distribution.

Data will also be obtained from these tests to determine heat flux to both the electrode walls and the insulating walls of the channel as a function of the power produced and the slagging characteristics of the channel. Performance of the diffuser will also be evaluated during this test series by measuring the axial static pressure distribution in the diffuser.

1.5.3 Independent Checkout of Steam Bottoming Plant and Seed Regeneration System

During the period in which the MHD plant is operating in the bypass mode, a number of tests will be performed on the steam generator and other components and systems in the facility. This activity can be conducted on a noninterference basis and will in no way impede the testing of the MHD system.

Checkout of the steam generator will closely follow the startup procedures employed in conventional steam plants. This will include hydrostatic testing, boilant and acid cleaning of the boiler and steam lines; setting of safety valves; calibration and tuning of controls. Refractory curing in the radiant furnace section and reduced load thermal performance testing will be accomplished by oil firing of the warm-up burners which have the capacity of developing ~ 30% of the rated steam generator output. During these tests a temporary closure panel will be used to seal the opening in the forward wall of the radiant furnace through which the MHD exhaust gas would normally pass.

Also, during this period, prior to the firing of the MHD combustor, the coal processing system will be checked out and the controls calibrated and adjusted. The HTAH Waste Heat Recovery System will also be checked out during this time. This will include performance evaluation of the fans and heat exchangers in the system and the calibration and tuning of controls.

The bypass mode operating period will also afford a convenient time for checkout of the Seed Regeneration System. This will consist of first testing the individual components of the seed regeneration plant for proper functioning followed by a test of the complete system. Initially K_2SO_4 will be used as the raw material feed stock and the KCOOH produced will be stored for subsequent use. A more detailed discussion of the test plan for evaluating operation of the system is given in Section 2.16.2.

1.5.4 Integration and Testing of Combined MHD/Steam Plant at 2500°F Preheat Temperature

Following the successful completion of the 100 hr test of the MHD system in the bypass mode, the diffuser to boiler transition section will be installed to duct the MHD exhaust gas to the steam generator. Because the steam generator and all the other BOP components will have been checked out in prior tests, operation of the integrated plant should be able to proceed within a relatively short time. It is assumed that the S. C. magnet is energized and available to deliver the rated magnetic field. It is also assumed that the HTAH system will have been maintained in a hot standby status after the previous test series so that it can be returned to normal operating temperature quite rapidly. The limiting factor will probably be the warm-up rate of the refractory lined duct from the HTAH to the combustor.

Another important consideration is the allowable heating rate of the refractory in the steam generator. Prior to firing the combustor this will be controlled by regulating the mass flow rate of high temperature air from the HTAH in combination with the firing rate of the oil-fired warm-up burners. When the refractory has attained operating temperature the combustor can be turned on at a low firing rate and gradually increased while simultaneously reducing the heat input from the warm-up burners. When steam temperature and pressure are suitable the ac generator turbine can be warmed up and begun to roll and accelerated to synchronous speed. As the steam supply increases the compressor drive turbine can be switched from the auxiliary steam supply to the main steam supply.

At $\sim 40\%$ mass flow rate the seed system can be turned on and power extracted from the MHD generator. The firing rate to the combustor can then be increased within the temperature ramp rates allowable for the main steam turbine. A more detailed discussion of the plant startup procedure giving the required sequence of events and operating limits for individual components is given in Section 1.4.4.2.

The primary objectives of the testing during this period are to investigate the problems associated with the startup of the integrated MHD/steam plant and to demonstrate that the overall control system will provide for safe and reliable operation of the facility. The performance of individual control elements and the capability of the integrated control system to provide the dual functions of plant safety and operational control will be closely monitored. Adjustments of control set points and tuning of the overall system will be performed as indicated by the test result.

During this period of testing the performance of the steam generator with the seed-laden exhaust products will also be evaluated. Of particular interest will be the thermal absorption capabilities of the various sections and the seed collection efficiency. The performance of the seed regeneration system operating with ash-contaminated seed material will also be determined.

Another important operating parameter to be investigated during the test period is the ability of the plant to comply with all the applicable environmental pollution regulations. These will include exhaust gas composition and particulate loading, wastewater discharge contamination, noise level, and all other pertinent federal, state and local standards.

1.5.4.1 Two Thousand Hour Endurance Test

After the comprehensive checkout of the combined MHD/steam plant has been completed and satisfactory performance from all the major components and overall control system has been attained, preparations will be made to conduct a long term endurance test of the facility. This will be primarily aimed to demonstrate the durability of the MHD channel under sustained load conditions because this component is generally regarded as being the most susceptible to damage from extended operation. As a consequence, the channel will be thoroughly instrumented to permit continuous

monitoring of all the critical parameters. This will enable the test conductor to make a critical assessment of the condition of the channel throughout the course of the test. The electrical and mechanical integrity of the channel electrode and insulating walls will be evaluated in relation to the severe electrical and mechanical stresses imposed by the test conditions. Effects of erosion, oxidation and corrosion from chemical attack by the seed and slag will also be determined. This test will be conducted at or near design rated power output depending upon the ability of the utility system to accept the power developed by the ETF. The conditions imposed by this test will, therefore, also demonstrate the endurance characteristics of the other MHD components as well as the steam generator portion and other major systems of the plant. Specific performance parameters and areas to be scrutinized in the steam generator during the course of the test are addressed in detail in Appendix B.

It is believed that successful conclusion of the 2000 hr test will be of sufficient significance to provide the confidence required for initiating design of a commercial size MHD plant.

1.5.4.2 Detailed Plant Inspection

Upon completion of the 2000 hr endurance test, the plant will be shut down for a period of ~ 3 months to conduct a thorough inspection of all the components. This will primarily be a visual inspection but other techniques such as metallurgical or radiographic inspection will be employed if conditions indicate them to be necessary. The primary purpose for the inspection is to assess the condition of the MHD components, steam generator and other items of plant equipment after the 2000 hrs of testing. On the basis of this information, necessary repairs or refurbishing will be performed and recommendations made for future design changes. An effort will also be made at this time to estimate the useful operating life of the various components to establish a realistic maintenance and replacement schedule for future operation.

After the inspection period ~ 4 months of testing will be conducted to determine overall plant performance over a range of parametric test conditions. Included in these tests will be operation for a sufficiently extended time to evaluate performance using Illinois #6 fuel. Other test parameters

to be investigated will include part load performance, response to load change, control system performance, combustor stoichiometry, fuel moisture content, channel loading, oxidizer enrichment, and any others which may be found to be necessary to obtain comprehensive plant performance characteristics.

This series of tests will conclude operation of the plant at 2500°F preheat temperature. Data from these tests in combination with the data previously obtained will provide an extensive fund of information for evaluating the performance of a combined MHD/steam power plant of this size.

1.5.5 Independent Testing of MHD System at 3000°F Preheat Temperature

During the early part of the third year of the test program the HTAH system will be upgraded to provide combustion air to the MHD cycle at 3000°F preheat temperature. This will not require extensive modification to the system for the original design specifications will have been prepared to permit operation under these conditions. The 2500°F preheat temperature used for the previous testing was selected to apply a conservative approach to the overall test program and to provide the opportunity for observing the performance of the HTAH system and the other major plant components under somewhat less severe conditions. At this stage of the program it is believed that sufficient operating experience and component performance and endurance information will have been acquired to begin operation of the plant at the higher temperature level with a sufficient degree of confidence.

Test activities during this period will closely parallel those conducted during the initial phase of the test program at 2500°F preheat temperature. The plant will be configured to operate in the bypass mode to permit testing of the individual MHD components and the complete MHD system independent from the steam bottoming plant. A new MHD channel and inlet nozzle will be installed at this time because a slightly modified channel geometry is required to provide optimum flow conditions at this higher temperature level. The sequence of tests will be similar to that previously followed for the 2500°F preheat condition and essentially the same type of information sought. A number of tests such as those originally conducted on various cooling water circuits and control systems which were not disrupted after the prior testing will obviously not have to be repeated at this time.

The primary objectives of the testing conducted during this period will be to determine the performance characteristics of the individual MHD components and also the performance of the complete MHD system at the higher preheat temperature. Because future commercial MHD plants will in all probability be designed to operate at $\sim 3000^{\circ}\text{F}$ preheat temperature it is particularly important to obtain sufficient operating and performance data at this condition to supply the necessary engineering and design data required for these plants. It is anticipated that the bypass mode testing at 3000°F preheat temperature can be completed in ~ 6 months.

1.5.6 Integration and Testing of Combined MHD/Steam Plant at 3000°F Preheat Temperature

Upon completion of the bypass mode testing described in the previous section, the facility will again be configured to duct the MHD exhaust gas from the diffuser directly to the steam generator. Because operating experience with the combined MHD/steam plant has already been obtained at the 2500°F preheat level, considerably less time will be required at this point in the test program to integrate the two systems. The startup procedure established during the earlier test period will be followed and necessary adjustments to individual control elements and the overall plant control system will be performed.

When it has been determined that the overall system is operating satisfactorily, a 2000 hr endurance test at 3000°F preheat temperature will be conducted. As in the previous 2000 hr test, the test conditions will be at or near design rated power output depending upon the ability of the utility system to accept the power produced by the ETF.

The primary objective of the test is to determine the endurance characteristics of the major MHD components and the steam generator. Of particular importance will be a measure of the MHD generator output as a function of operating time to determine if a correlation between performance degradation and operating hours can be established. This information will be helpful in extrapolating the useful operating life of the generator channel.

Specific performance parameters in the steam generator section and the exhaust gas cleaning system which will be carefully monitored during the course of the test are discussed fully in Appendix B.

After completion of the 2000 hr test the plant will be shutdown to conduct a thorough inspection of all the principal components and systems. Necessary repairs or parts replacement will be performed and design changes to improve upon the endurance characteristics of deficient components will be recommended. An estimate of the remaining useful operating lifetime of the various components will also be made to help in establishing a realistic maintenance and overhaul schedule for future plant operation.

Following the inspection period a parametric test program will be conducted to determine plant performance over a range of operating conditions. Test parameters to be investigated will be essentially the same as those described in Section 1.5.4 for the parametric test program conducted as 2500°F preheat temperature.

1.5.7 Integration and Testing of Complete MHD/Steam Plant with Coal Gasifier System

Midway in the fourth year of the test program the HTAH system will be converted from using #2 fuel oil to a low Btu fuel gas produced by Wellman-Galusha coal gasifiers. The ETF will become at that time an all coal burning power system. Introduction of the coal gasifiers into the facility was deliberately delayed until this stage of the test program so that advances in gasification technology which may accrue from the ongoing development effort in this area can be incorporated. The Wellman-Galusha gasifier is specified at this time for reasons stated elsewhere in this report (Section 2.12) but other systems will be considered on the basis of new information generated by several DOE programs.

Because the reheat burners of the HTAH system are designed for dual fuel operation virtually no modifications will need to be made to the burners. Also the ducting and valves required to deliver and control the flow of gas from the gasifier trunk line to the individual burners will have been installed during initial construction.

Operation of the HTAH system with low Btu gas can be evaluated without discharging the high temperature preheated air to the MHD flow train by directing the high temperature air to the bypass line which has the capability for quenching the air by water sprays before it is sent to the stack.

After satisfactory operation of the HTAH system with low Btu gas has been demonstrated testing of the complete MHD/steam plant in combination with the gasifier system will be conducted to evaluate the overall performance of an all coal fired power system. The energy source for supplying the high temperature air to the MHD cycle should have no impact upon the performance of the combined MHD/steam plant so the testing will be primarily directed toward establishing optimum operating procedures for the integrated all coal fired facility.

One of the questions which has been raised concerning the use of coal gasifiers as the fuel source for the HTAH system is the long term effect of tars and particulate matter which may be present in the LBtu gas. This will be investigated, however, the data available on the Wellman-Galusha product gas indicates that it is sufficiently low in ash so that there should be no problem on this score. Also the temperature at which the gas is used is well above the condensation temperature of any liquids which may be carried over.

Demonstration of satisfactory operation of the overall facility using coal as the only fuel source will essentially complete the formal test program. Thereafter the facility can be used as an engineering test bed to evaluate new or improved component design concepts or it can be placed online to furnish power to the utility grid.

1.6 PROJECT COSTS AND SCHEDULES

1.6.1 Engineering and Construction Costs

The summary estimate format and code of accounts supplied in this report, Table 1-23, has been developed in accordance to the Department of Energy's directives and closely follows standard estimating practices. The cost elements identified for each line item are those costs for materials (divided into major components and balance of plant), costs for field installation, indirect costs, a specific contingency, and a total cost for that item. Costs are in 1977 $\frac{1}{2}$ dollars and are shown in thousands of dollars.

"Major Components" have been identified as those items which are engineered, designed, fabricated, shipped, and in some cases, erected by one supplier. MHD components are analogous with components of high technology.

"Balance of Plant" items are normally designed, engineered, and purchased by the engineer. All material costs include charges for shipment and delivery to the site.

The "Installation" portion of the direct cost includes wage costs for all manual labor, foremanship, all wage related benefits and costs mandated by labor agreement. Payroll taxes, payroll premium costs, and workmen's compensation insurance costs are built into the wage rate of direct labor costs. Also included is special construction equipment associated with certain civil work items to which the costs can be charged directly, and contractor fees. Auxiliary labor for unloading, storing, sorting materials and equipment, general and final cleanup, and other miscellaneous activities directly associated with the installation of the work are also charged to the direct account.

"Indirect Costs" for construction are those cost items, which include facilities, equipment and services, that are required to directly support the construction operations but which cannot be conveniently so charged by the constructor or general contractor directly to a single estimating account. For conceptual estimates, indirect construction costs are expressed as a percentage of the direct cost. Field offices and temporary facilities, transportation, safety equipment, construction tools and equipment, expendable supplies, non-manual labor, construction services and testing contracts, and insurance and bonds are all examples of indirect costs.

TABLE 1-23
ETF PROJECT COST ESTIMATE

ACCOUNT NUMBER	ACCOUNT DESCRIPTION	UNIT	QUANTITY	MATERIAL		INSTALLATION COST	INDIRECT COST	CONTINGENCY	TOTAL COST
				MAJOR COMPONENT	BALANCE OF PLANT COST				
310.	Land and Land Rights	JOB	1	-	242	-	-	36	278
311.	Structures and Improvements	JOB	1	-	13,781	11,339	5,670	4,757	35,547
311.1	Improvements to Site	JOB	1	-	395	1,671	836	435	3,337
311.2	MED Building	CF	1,950,000	-	2,242	1,755	878	731	5,606
311.3	Bottoming Plant Building	CF	650,000	-	748	585	292	244	1,869
311.4	Steam Generator Building	CF	990,000	-	1,138	891	446	371	2,846
311.5	HTAH Building	CF	4,980,000	-	5,727	4,482	2,241	1,868	14,318
311.6	Maint. Serv., Warehouse and Office Buildings	CF	890,000	-	1,024	801	400	334	2,559
311.7	Other Buildings	SF	30,000	-	660	270	135	160	1,225
311.8	Cranes and Hoists	JOB	1	-	350	50	25	64	489
311.9	On-site Waste Treatment	JOB	1	-	1,497	834	417	550	3,298
312.	Boiler Plant	JOB	1	-	22,307	7,435	3,718	6,356	39,816
312.1	Coal Handling	JOB	1	-	2,878	2,146	1,073	1,065	7,162
312.11	Unloading and Yard Storage	JOB	1	-	1,352	1,154	577	462	3,545
312.12	Reclaim and Delivery	JOB	1	-	1,526	992	496	603	3,617
312.2	Slag and Ash Handling	JOB	1	-	600	108	54	152	914
312.4	Steam Generator	JOB	1	-	13,979	3,729	1,863	3,912	23,483
312.41	Steam Generator	JOB	1	-	12,258	3,273	1,636	3,433	20,600
312.42	I.T. Air Heater	JOB	1	-	666	177	88	186	1,117
312.43	Instrumentation & Controls	JOB	1	-	372	98	49	102	621
312.44	Auxiliaries	JOB	1	-	683	181	90	191	1,145
312.5	Effluent Control	JOB	1	-	2,102	603	304	601	3,610
312.51	Precipitator and Breeching	JOB	1	-	1,118	295	148	312	1,873
312.52	Chimney	EA	1	-	415	225	112	150	902
312.53	Bypass System	JOB	1	-	569	83	44	139	835
312.6	Auxiliary Boiler System	JOB	1	-	1,207	181	90	222	1,700
312.7	Other Boiler Plant System	JOB	1	-	1,541	668	334	404	2,947
312.71	Condensate and Feedwater Sys.	JOB	1	-	341	78	39	92	550
312.72	Water Treatment System	JOB	1	-	900	540	270	256	1,966
312.73	Water Supply System	JOB	1	-	300	50	25	56	431

MAIN

TABLE 1-23
ETF PROJECT COST ESTIMATE (CONTINUED)

ACCOUNT NUMBER	ACCOUNT DESCRIPTION	UNIT	QUANTITY	MATERIAL		INSTALLATION COST	INDIRECT COST	CONTINGENCY	TOTAL COST
				MAJOR COMPONENT	BALANCE OF PLANT COST				
314.	Steam Turbine Gen. and Aux.	JOB	1	-	8,520	3,821	1,912	2,245	16,498
314.1	Steam Turbine and Aux.	JOB	1	-	3,871	2,635	1,318	1,174	8,998
314.2	Condenser and Auxiliary	JOB	1	-	450	175	88	107	820
314.3	Circulating Water System	JOB	1	-	1,566	556	278	360	2,760
314.31	Pumps, Valves, Piping and Struct	JOB	1	-	840	318	159	198	1,515
314.32	Cooling Towers	JOB	1	-	726	238	119	162	1,245
314.4	Steam Piping Systems	JOB	1	-	908	230	115	191	1,444
314.41	Main Steam	JOB	1	-	468	120	60	97	745
314.42	Auxiliary Steam	JOB	1	-	50	10	5	13	78
314.43	Misc. Piping Systems	JOB	1	-	390	100	50	81	621
314.5	Other Turbine Plant & Mech Equip.	JOB	1	-	1725	225	113	413	2476
315.	Accessory Electric Equipment	JOB	1	-	6,498	2,122	1,060	1,915	11,595
315.1	Station and Aux. Transf.	EA	1	-	128	17	8	23	176
315.2	Misc. Motors	JOB	1	-	200	40	20	39	299
315.3	S.G. and MCCs	JOB	1	-	890	120	60	214	1,284
315.4	Conduit Tray, Cable and Busswork	JOB	1	-	1,450	600	300	470	2,820
315.5	Misc. Electrical Equipment	JOB	1	-	840	240	120	240	1,440
315.6	Integrated Control System	JOB	1	-	1,400	600	300	460	2,760
315.7	Data Acquisition System	JOB	1	-	900	385	192	295	1,772
315.8	Emergency Power Systems	JOB	1	-	690	120	60	174	1,044
316.	Misc. Power Plant Equipment	JOB	1	-	985	200	100	194	1,479
316.1	Fuel Oil Handling System	JOB	1	-	215	50	25	44	334
316.2	Fire Protection System	JOB	1	-	390	130	65	88	673
316.3	Machine Shop and Maint. Shop	JOB	1	-	380	20	10	62	472

MAIN

TABLE 1-23
ETF PROJECT COST ESTIMATE (CONTINUED)

ACCOUNT NUMBER	ACCOUNT DESCRIPTION	UNIT	QUANTITY	MATERIALS		INSTALLATION COST	INDIRECT COST	CONTINGENCY	TOTAL COST
				MAJOR COMPONENT	BALANCE OF PLANT COST				
317.	MHD Topping Cycle	JOB	1	\$1,497	6,208	10,497	5,256	21,081	94,539
317.1	Combustion Equipment	JOB	1	2,292	1,728	797	400	1,323	6,540
317.11	Coal Drying	JOB	1	-	1,728	459	230	483	2,900
317.12	Coal Injection	JOB	1	1,012	-	269	135	425	1,841
317.13	Combustor	JOB	1	1,020	-	24	12	317	1,373
317.14	Slag Coll. System	JOB	1	260	-	45	23	48	426
317.2	MHD Generator	JOB	1	4,404	-	166	84	1,395	6,049
317.21	Nozzle	EA	1	1180	-	5	3	56	244
317.22	Channel (3000°F)	EA	1	3,600	-	90	45	1,120	4,855
317.23	Diffuser & Trans. Section	JOB	1	624	-	71	36	219	950
317.3	Magnet Subsystem	JOB	1	15,885	-	227	114	4,868	21,094
317.31	Structure	EA	1	1,754	-	48	24	548	2,374
317.32	Winding Assembly	EA	1	8,341	-	10	5	2,507	10,863
317.33	Dewar	EA	1	4,192	-	20	10	1,267	5,489
317.34	Refrigeration System	JOB	1	1,015	-	15	8	311	1,349
317.35	DC Power Instr. & Control Other	JOB	1	583	-	134	67	235	1,019
317.4	Inverters and Electrode Con.	JOB	1	4,351	-	1,299	650	1,890	8,190
317.41	Inverters	JOB	1	2,508	-	818	409	1,120	4,855
317.42	Electric Consolidation Circ.	JOB	1	1,843	-	481	241	770	3,335
317.5	Oxidizer Preheat System	JOB	1	22,923	2,969	6,290	3,148	10,189	45,519
317.51	Air Compressor and Drive	JOB	1	-	2,476	396	198	614	3,684
317.52	Comb. Air and Gas Piping, Fans	JOB	1	-	378	109	57	76	620
317.53	L.T. Air Heater	JOB	1	-	115	30	15	32	192
317.54	H.T. Air Heater	JOB	1	22,923	-	5,755	2,878	9,467	41,023
317.541	Vessels	EA	4	14,824	-	4,037	2,018	6,264	27,143
317.542	Reheat Burner	EA	4	3,530	-	1,115	558	1,561	6,764
317.543	Ducting	JOB	1	2,939	-	603	302	1,153	4,997
317.544	Auxiliary Equipment	JOB	1	1,630	-	0	0	489	2,119
317.6	Seed Subsystem	JOB	1	1,642	-	699	350	808	3,499
317.61	Seed Regeneration Process	JOB	1	1,192	-	411	206	543	2,352
317.62	Seed Injection System	JOB	1	450	-	288	144	265	1,147
317.7	Other MHD Topping Equipment	JOB	1	-	1,511	1,019	510	608	3,648
317.71	O. System	JOB	1	-	11	19	10	8	48
317.72	Gasifier System	JOB	1	-	1,500	1,000	500	600	3,600

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TABLE 1-23
ETF PROJECT COST ESTIMATE (CONTINUED)

ACCOUNT NUMBER	ACCOUNT DESCRIPTION	UNIT	QUANTITY	MATERIAL		INSTALLATION COST	INDIRECT COST	CONTINGENCY	TOTAL COST
				MAJOR COMPONENT	BALANCE OF PLANT COST				
318.	Research Equipment	JOB	1	6,140	-	1,120	561	2,345	10,166
318.1	MDH Channel (2500 °F)	EA	1	3,600	-	90	45	1,120	4,855
318.2	Channel Nozzle (2500 °F)	EA	1	180	-	5	3	56	244
318.3	Test Instrumentation	JOB	1	2,360	-	1,025	513	1,169	5,067
319.	Simulation Equipment	JOB	1	-	45	20	10	15	90
319.1	DC Resistive Load Bank	JOB	1	-	45	20	10	15	90
350.	Transmission Plant	JOB	1	-	1,203	469	235	285	2,192
350.1	Structures and Improvements	JOB	1	-	142	116	58	47	363
350.2	Main Transformers	JOB	1	-	649	131	66	127	973
350.3	Switchyard	JOB	1	-	412	222	111	111	856
	Subtotal - Direct Accounts			57,637	59,789	37,023	18,522	39,229	212,200
	Engineering Services	JOB	1	-	-	14,000	-	2,800	16,800
	Preliminary Engineering	JOB	1	-	-	3,500	-	700	4,200
	Detailed Design	JOB	1	-	-	7,000	-	1,400	8,400
	Construction Management	JOB	1	-	-	3,500	-	700	4,200
	Other Costs	JOB	1	-	-	5,125	-	1,025	6,150
	TOTAL Estimated Construction Cost			57,637	59,789	56,148	18,522	43,054	235,150

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"Contingency" represents the total contingency that has been applied to each line item. As owner committed monies for purchase material and negotiated contracts proceed toward 100% of total project cost, necessary contingency factors may be reduced in a manner to reflect lessened possibilities of unforeseeable circumstances occurring before project completion. As project engineering nears completion, estimating may deal with more precise information and can more accurately predict material quantities and respective project costs. Items incorporated into contingency management considerations include:

- Design (but not major scope) changes
- Market conditions
- Labor productivity
- State of project definition
- Unreliable and non-current estimating data
- Unpredictable field conditions
- Material and labor markets instabilities
- Uncertainties in project timing
- Errors and omissions
- Weather
- Short term strikes, walkouts, and other labor disputes
- Other unforeseeable occurrences and conditions which would delay or otherwise increase material and/or installation costs.

The contingency factors used in this report reflect varying degrees of development and uncertainty. In the case of Balance of Plant (BOP) structures, improvements and well-defined mechanical systems 15% is used. This factor includes recognition of the conceptual nature of the engineering work, so it is higher than the 5-10% normally associated with such items. For certain other BOP items, a factor of 20% is used in recognition of a greater degree of uncertainty with respect to engineering. For the higher technology components, a factor of 30% is used to reflect the relatively high degree of uncertainty for these items.

"Total Cost" represents the total for all material, installation, indirect, and contingency costs for each account.

Professional services include project management, preliminary engineering, detailed design and engineering, construction management,

procurement services, architectural design, shop inspection, expediting, and startup and testing (not including the extensive operational testing of the MHD facility once it begins operation). For this project, professional service charges have been subdivided into preliminary engineering (2% of total direct and indirect costs), detailed design and engineering (4% of total direct and indirect costs), and construction management (2% of total direct and indirect costs).

The "other costs" category includes such items as the Government's field staff, legal fees and ad valorem taxes. Normally these are owner's charges, but in the case of the MHD Test Facility these costs would be incurred by the DOE. A factor of 2.5% of the Direct, Indirect, and Professional Services subtotal costs was assumed for this category. As directed by DOE, no costs for escalation or allowance for funds during construction (AFDC) have been included in this estimate. Estimated annual rates of escalation for the period through 1985 are shown on Table 1-24.

1.6.2 Operating Personnel

The plant operating personnel will consist of all personnel necessary for the day-to-day operation and maintenance of the plant. They will be responsible for the safe and efficient operation of all plant equipment. They will perform all routine maintenance as necessary to keep the plant operational. The Test and Instrumentation and Control personnel will be available for conducting routine tests, calibration of instruments and maintenance of plant controls. They will be available for calibration and installation of instruments used for special tests by outside personnel.

Major tests of equipment such as the MHD channel or high temperature air heaters will be conducted with the manufacturer's personnel. The plant personnel will operate all equipment but with guidance and advice from the manufacturer's personnel. The remaining plant personnel may be used to supplement the manufacturer's test personnel if they are available.

The plant manager will be at the job site approximately one (1) year ahead of the plant start-up.

The plant operators (control room operator, shift formen, etc.) will be at the plant site approximately six (6) months prior to plant

TABLE 1-24
CHAS. T. MAIN POWER PLANT
ESCALATION COST INDEX

E = Estimated

Average Index for Year:	1967	=	100.0	
				$\Delta = +2.6\%$ Change from 1967 to 1968
	1968	=	102.6	
				$\Delta = +3.4\%$
	1969	=	106.1	
				$\Delta = +3.7\%$
	1970	=	110.0	
				$\Delta = +3.6\%$
	1971	=	114.0	
				$\Delta = +2.6\%$
	1972	=	117.0	
				$\Delta = +7.6\%$
	1973	=	125.9	
				$\Delta = +22.2\%$
	1974	=	153.0	
				$\Delta = +11.5\%$
	1975	=	171.5	
				$\Delta = +6.3\%$
	1976	=	182.3	
				$\Delta = +6.2\%$
	1977	=	195.1	
				$\Delta = +7.2\%E$
	1978	=	209.1E	
				$\Delta = +6.4\%E$
	1979	=	222.5E	
				$\Delta = +6.0\%E$
	1980	=	235.8E	
				$\Delta = +6.0\%E$ Change from 1980 to 1981
	1981	=	249.9E	
				$\Delta = +6.0\%E$

TABLE I--24
CHAS. T. MAIN POWER PLANT
ESCALATION COST INDEX (CONTINUED)

Average Index for Year:	1982	=	264.9E $\Delta = + 6.0\%$
	1983	=	280.8E $\Delta = + 6.0\%$
	1984	=	297.6E $\Delta = + 6.0\%$
	1985	=	315.4E

start-up. The first three (3) months will consist of plant familiarization consisting of studying the various manufacturer's instruction manuals. They will physically locate each pipe, valve, etc. throughout the plant using the flow diagrams as a check sheet. They will also prepare detailed operating instructions.

The remaining time prior to actual pre-op and start-up will consist of formal classroom instruction and training. Specialists from the major manufacturers will be brought in to provide detailed instructions as to the operation of the equipment.

The plant personnel will be broken down into four (4) basic categories. A brief description of each category is as follows:

Administration

The administration of the ETF plant will consist of approximately eighteen (18) people.

The major administrator will be the plant manager. He will have complete authority of all operations in the plant. He will be responsible for the safe and efficient operations of the plant. He will assure that operating personnel are trained and available to perform their assigned tasks as necessary for routine operations or special tests.

The other administrative personnel will consist of the assistant plant manager, the necessary clerks and typists, the storekeepers and plant security personnel.

The assistant manager will be available and have the capability to act for the plant manager should he be absent from the plant site.

Operations

The Operations category is one of the most vital groups in the plant. This group will consist of approximately thirty-two (32) people initially and, as the ETF program advances, the operating staff will be increased. No increase in staff is anticipated before year six.

The operating staff consists of all the operating personnel required to safely and efficiently operate the plant equipment. This includes the personnel who will man the control room on a continuous twenty-four (24) hour basis. It will also include the equipment operators who will rove around the plant and be available to manipulate valves, check over equipment

for start-up and/or shutdown and, in general, perform any service as directed by the control operators in the central control room.

Included in the operations group will be the yard or outside auxiliary personnel such as the coal handling personnel and personnel to operate the seed plant equipment as well as other outside facilities.

Initially, the seed plant will be operated and staffed by a separate group until such time as the seed recovery process is proven and can be remotely operated from the central control room.

The key individual in the operations group is the operations supervisor. He will be responsible for the overall operations plans for the facility. He will answer to the plant manager. He will have the four shift supervisors answering directly to him. The four shift supervisors will be shift personnel. There will be one shift supervisor on each eight hour shift. The fourth is required for the rotation schedule. The shift supervisor is responsible for the operation of the plant. He will back up and instruct all of the operating personnel. He will also be responsible for clearance permit procedures before equipment is cleared for maintenance. During the evening and night shifts, he will be the senior man in the plant and be responsible for all functions necessary for the plants safety.

Maintenance

The maintenance group is by far the largest division in the plant consisting of thirty-nine (39) people initially and expanding to fifty-seven (57) as the plant is proven and components begin to age.

The maintenance staff is primarily mechanical and electrical to perform routine maintenance on all plant equipment and auxiliary systems. Routine maintenance will include, but is not limited to, packing valves and pumps, replace bearings, balance fans, replace motors, repair and/or replace pipe lines, power circuits, etc.

The maintenance staff is also expected to perform required maintenance work of the channel and other MHD components with directions supplied by special MHD test personnel at the site or from respective equipment designers and manufacturers. In case of extensive repair work of the channel or other MHD components the equipment may be shipped to the manufacturer for overhaul.

It is not anticipated that this staff would be large enough for major outages and repairs such as turbine generator overhaul or major repairs to the high temperature air heaters. This type of work will require hiring outside contractors. It would not be economical to maintain on a continuous basis the necessary personnel to perform this type of work.

The maintenance group will be headed by a maintenance supervisor. He will be responsible for the day-to-day maintenance, planning, major outage planning, etc. He will coordinate the maintenance planning with the operations supervisor to assure that maximum availability of equipment is achieved.

The maintenance supervisor will have two (2) mechanical and two (2) electrical foremen under his direct supervision.

The maintenance foreman will schedule and plan the day-to-day activities of the personnel under their supervision. They will assure that the personnel are trained and instructed in proper maintenance procedures and methods. They will also be responsible to assure the maintenance personnel perform their work safely.

Plant Engineering

The plant engineering group will consist of the personnel required for the efficient operation of the plant. The plant engineers will be responsible for monitoring the efficiency of equipment and operating methods and the general overall performance of the plant. The instrument and control engineers and technical staff would maintain all instrumentation and control equipment such as calibration of monitoring devices, fine tuning of control equipment, etc. They would install and calibrate all test equipment for any tests required by the various manufacturers who may want to run specialized tests. For any major equipment performance or specialized tests, the specific manufacturer would supply any additional test personnel that may be required.

The Results and Instrumentation and Control group will be headed by a chief Results, Instrumentation and Control supervisor. There will be four (4) results engineers who will perform various performance tests and calculate equipment efficiencies and make recommendations to improve performance.

The chief will also have the Instrumentation and Control technicians under his direct supervision. These people will be responsible for maintenance and calibration of all instrumentation, control and monitoring apparatus.

The laboratory personnel would be available to monitor and test all chemical variables, coal analysis, seed recovery analysis, etc.

Costs

It is anticipated that the annual operating costs for operation will be \$2,811,086 in 1977 $\frac{1}{2}$ dollars through years one to five. By year six, the staff will be increased and cost approximately \$3,465,560 in 1977 $\frac{1}{2}$ dollars. This has not been escalated. An estimated total cost for twenty (20) years of operation will be \$66,038,830 including a 1.45 increase for overhead and fringe benefits. These figures are exclusive of repair parts and expendable items such as fuel, seed, oils and chemicals required. See Appendix for detail breakdown of operation, maintenance and supervisory costs.

In addition, the major equipment manufacturers will have a projected cost of \$1,591,200 for supplying start-up and testing services necessary for the early operation of the ETF equipment. See Appendix for details.

The total plant operating, and maintenance costs are shown on Table 1-25. This table shows that the plant will cost \$153,621,460 to operate over a 20 year life and 50% load factor. This table includes all costs including manpower, fuel, extraordinary maintenance and routine maintenance costs. The costs breakdown for each year during the first five years of the ETF operation are provided in the Appendix. The expected hours of operation for each of the first five years are also shown. The remaining 15 years are lumped as one item based on a 50% load factor.

1.6.3 Engineering and Construction

Project Schedule

The project engineering and construction schedule is based on the following assumptions:

TABLE 1-25

PLANT OPERATING COSTS

(Based on 20 Year Life - 50% Load Factor)

	<u>Total Cost</u> (x 1000)
Plant Operating and Maintenance Manpower	\$ 66,038.83
Start-up and Testing Services	1,591.00
Consumables (including Coal & Fuel Oil)	70,269.46
Major Maintenance Material & Labor	14,242.00
Balance of Plant Maintenance & Spare Parts	<u>1,480.17</u>
TOTAL	\$153,621.46

Note: See Appendix for Item Breakdowns for first five years and estimates for last fifteen years.

- 1) Preliminary engineering will start 18 months prior to detailed design.
- 2) Six (6) months is allowed for DOE approval and selection of a contractor. Three to four months after contractor selection, orders are placed for the major equipment.
- 3) Detailed design will be performed by an architect engineer.
- 4) Phased-Multiple contract construction will be directed by a construction manager, probably the architect engineer. Foundation work will begin in the high temperature air heater area followed by the MHD area and the bottoming plant.
- 5) Check out of the components necessary for testing of MHD components in the by-pass mode will begin at the end of the fifth year. By-pass mode testing of MHD components will be initiated in the middle of the sixth year. A separate schedule shows this activity. See Table 1-26.

Check out and trial operation of the bottoming plant, which can be performed independently, will begin in the fourth month and the sixth year.

Construction of the gasifier will begin in the seventh year. Completion of the gasifier is phased into the overall test program schedule.

Only those major, critical items with the longest engineering, manufacture, and construction durations which determine the overall length of the project are shown on the Conceptual Project Schedule. The continuous path of activities with the longest cumulative sum of activity durations is called the "critical path", and all the activities which form that path are called "critical activities". The critical path for this project follows through the design, fabrication, erection, and start-up of the high temperature air heaters and the MHD combustor and magnet.

Any delay in these critical activities will result in an equivalent delay of the project completion date, unless possibly circumvented by alternate logic, premium time, or other usually expensive and nonrecommended alternatives.

	YEAR 1	YEAR 2	YEAR 3	YEAR 4	YEAR 5	YEAR 6
PRELIMINARY DESIGN						
DETAILED DESIGN						
MANUFACTURE AND DELIVERY						
H.T. AIR HEATERS						
BOILER						
STEAM TURBINE						
MHD COMBUSTOR						
MHD CHANNEL & DIFFUSER						
MHD MAGNET						
CONSTRUCTION						
SITE PREPARATION						
STATION FOUNDATIONS						
CIRCULATING WATER SYSTEM						
BUILDINGS						
COAL HANDLING						
HTAH-VESSEL SUBASSEMBLY						
VESSEL CONSTRUCTION						
REFRACTORY MATRIX						
PIPING VALVES & BURNERS						
MHD -COMBUSTOR						
CHANNEL & DIFFUSER						
MAGNET						
MAGNET DEWAR						
BOILER STRUCTURAL STEEL						
BOILER PRESSURE PARTS						
COAL DRYING & PROCESSING						
STEAM TURBINE						
BYPASS SYSTEM						
SEED REPROCESSING						
INVERSION EQUIPMENT						
AUX. MEC. -ELECTRICAL						
CHECKOUT-BYPASS MODE						
CHECKOUT, TRIAL OPERATE-BOTTOMING PLANT						

NOTE:

1. LEAD TIME FOR GASIFIER DETAILED DESIGN & MANUFACTURE IS FROM THE 1st MONTH OF THE 7th YEAR TO THE 6th MONTH OF THE 8th YEAR (18 MOS. DURATION). GASIFIER CONSTRUCTION IS FROM THE 7th MONTH OF THE 8th YEAR TO THE 6th MONTH OF THE 9th YEAR (12 MOS. DURATION).

2.0 SUBSYSTEM/COMPONENTS DESIGN

2.1 MHD CHANNEL

2.1.1 Introduction and Summary

A major consideration in the design of the ETF channel is that its performance and size be adequate to provide a realistic basis for scaling to baseload size. Performance criteria selected as of major interest in determining scalability were the enthalpy extraction and the isentropic efficiency. The target value for enthalpy extraction was chosen as 12.5%, and the target value for isentropic efficiency was chosen as 50%. These target values imply a certain minimum channel size (i. e., mass flow capability) since they cannot be attained in too small a channel. A channel of larger than the minimum size required to demonstrate scalability requires a larger facility, and hence a higher overall cost.

In order to establish the channel size, a parametric study was performed in which were considered effects of mass flow rate, inlet Mach number, stagnation pressure and electrical load parameter. Air preheat temperatures of 2500°F and 3000°F were considered. Prior to the availability of the actual magnet design developed as part of this design work, a simplified magnetic field distribution was assumed as shown in Figure 2-1. Detailed channel calculations were subsequently carried out with the actual magnetic field distribution for the selected ETF channel size. Details of the channel calculations are described in subsequent sections of this report.

A summary of the tradeoffs considered and the various design features of ETF reference channels are presented in Tables 2-1 through 2-5 inclusive:

Table 2-1 - General characteristics of the ETF reference channels at air preheat temperatures of 2500°F and 3000°F.

Table 2-2 - Comparison of ETF channel performance with that of other channel designs.

Figure 2-1 Simplified Magnetic Field Distribution

TABLE 2-1

SUMMARY

CHARACTERISTICS OF ETF REFERENCE CHANNELS AT THE FULL LOAD OPERATING POINT

	<u>AIR PREHEAT</u>	
	<u>2500 °F</u>	<u>3000 °F</u>
FUEL	MONTANA SUB-BITUM.	MONTANA SUB-BITUM.
THERMAL INPUT	250 MW	270 MW
MASS FLOW ($\dot{m}$)	65 kg/s	65 kg/s
INLET STAGNATION PRESSURE (P_{o_i})	4.0 atm	5.4 atm
INLET STAGNATION TEMP. (T_{o_i})	2675 °K	2794 °K
INLET MACH NO. (M_i)	0.85	0.85
LOADING PARAMETER (FARADAY)	0.69	0.69
CHANNEL DESIGN MODE	CONSTANT VELOCITY	CONSTANT VELOCITY
POWER OUTPUT	35.5 MW	45.6 MW
ENTHALPY EXTRACTION	14.2 %	16.9 %
ISENTROPIC EFFICIENCY	57 %	57.8 %
EXIT STAGNATION PRESSURE (P_{o_f})	1.2 atm	1.2 atm
EXIT MACH NO.	0.9	0.92
DIFFUSER EFFICIENCY	46 %	46 %
LENGTH	9 m	9.0 m
INLET CROSS-SECTION	0.48 x 0.48 m	0.42 x 0.42 m
EXIT CROSS-SECTION	0.90 x 0.90 m	0.91 x 0.91 m
PEAK MAGNETIC FIELD (B)	6 T	6 T
AVERAGE MAGNETIC FIELD	5 T	5 T
MAX. TRANSVERSE CURRENT DENSITY (J_y)	0.85 A/cm²	1.1 A/cm²
MAX. HALL FIELD (E_x)	2.5 kV/m	2.4 kV/m
MAX. HALL PARAMETER ($\omega\tau$)	4.4	4.4
AVERAGE POWER DENSITY	8.4 MW/m³	11.6 MW/m³

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TABLE 2-2
COMPARATIVE MHD CHANNEL PERFORMANCE DATA

						COMMERCIAL BASELOAD			
		AVCO MK-VI	CDIF # 1	ETF		BASELINE 1		BASELINE 2	ECAS
				2500 °F PREHEAT	3000 °F PREHEAT	2500 °F PREHEAT	3000 °F PREHEAT	2500 °F PREHEAT	2500 °F PREHEAT
THERMAL INPUT	MW _{th}	15	40 - 50	250	270	1397	1511	2850	5300
PRESSURE RATIO		2.9 - 3.5	5.5	4.0	5.4	5.5	6.5	7.0	8.0
INLET TEMP. (STAG.)	°K	2900	2875	2675	2794	2744	2845	2780	2830
MAX. MAGNETIC FIELD	TESLA	2.8	3	6	6	6	6	6	6
MASS FLOW RATE	kg/s	2.5	8 - 10	65	65	353	353	700	1270
POWER OUTPUT	MW	0.2-0.6	1.5-2.5	35.5	45.6	288	342	634	1420
MAX. TRANSV. CURRENT	amps/cm ²	0.8-1.5	0.9-1.5	0.65	1.1	0.7	0.9		1.0
MAX. AXIAL FIELD	kV/m	1.0-3.0	2.0-3.0	2.5	2.4	1.9	1.7		2.7
HALL PARAMETER	$\omega\tau$	1.5-2.5	1.6-2.3	4.4	4.4	3.6	3.9		4.1
POWER DENSITY (AVG)	MW/m ³	5-15	7-12	8.4	11.6	6.4	8.7	10.5	10.0
ENTHALPY EXTRACTION	%	1.5-4.0	4 - 6	14.2	16.9	20.6	22.6	22.3	26.5
ISENTROPIC EFFICIENCY	%	≈ 40		57.0	57.8	68.5	71.5		76.0
DIFFUSER EFFICIENCY	%	≈ 45		46	46	61	61		70

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TABLE 2-3
ETF CHANNEL DESIGN
GENERAL PERFORMANCE CONSIDERATIONS*

PARAMETER/VARIABLE	DESIGN CONSIDERATIONS/TRADE-OFFS	ADDITIONAL COMMENTS
1. MAGNETIC FIELD B	<ul style="list-style-type: none"> COST OF MAGNET $\sim$ STORED ENERGY $\sim (V_0 B^2)^n$ HIGH-B FIELD MEANS HIGH POWER DENSITY & SHORT LENGTH; BUT ALSO HIGH E_x, ωT & J_y (CONSTRAINTS ON LATTER MAY BE EXCEEDED) STATE-OF-THE-ART FOR MHD MAGNETS IS PEAK FIELD OF APPROX. 6T IN ACTIVE VOLUME 	$E_x = \omega T \mu B (1-K) \sim B^2$ $\omega T \sim B/\rho$ $J_y = \sigma \mu B (1-K)$ $V_0 =$ VOLUME OF MAGNET BORE
2. CHANNEL LENGTH	<ul style="list-style-type: none"> ELECTRICAL CONDUCTIVITY: $\sigma \sim 1/\sqrt{P_{01}}$; THEREFORE LENGTH: $L \sim (P_{01})^{3/2} / B^2$ (REFER R. ROSA**) GENERAL AIM: DEFINE CHANNEL LENGTH TO ACHIEVE DESIRED PERFORMANCE; ARGUES FOR OPTIMUM LOADING PARAMETER & ENTHALPY EXTRACTION HIGH P_{01} IMPLIES: <ul style="list-style-type: none"> HIGHER MASS FLOW & POWER OUTPUT, & GREATER COST HIGHER THEORETICALLY ATTAINABLE ENTHALPY EXTRACTION & ISENTROPIC EFFICIENCY HIGH B-FIELD IMPLIES SHORT LENGTH BUT HIGH E_x, ωT & J_y (REFER ITEM #1) 	<ul style="list-style-type: none"> P_{01} = INLET STAGNATION PRESSURE REFER ALSO TO DISCUSSION OF "MHD CHANNEL CALCULATIONS" IN SECTION 2.2.2
3. POWER OUTPUT $\dot{P}$	<ul style="list-style-type: none"> $\dot{P}$ PROPORTIONAL TO MASS FLOW. $\dot{P} \sim P_{01} D^2 \dot{m}$ IF CHANNEL L/D IS APPROX. CONSTANT AS FUNCTION OF SIZE, THEN $\dot{P} \sim P_{01} L^2$ (REFER ITEM #2) HIGH $\dot{m}$ MEANS HIGH OUTPUT POWER BUT ALSO HIGHER MAGNET/PLANT COST AS CHANNEL SIZE & $\dot{m}$ INCREASE, SURFACE / VOLUME RATIO DECREASES WHICH IMPLIES: <ul style="list-style-type: none"> IMPROVED ENTHALPY EXTRACTION IMPROVED ISENTROPIC EFFICIENCY 	$\dot{P} \sim \sigma \mu^2 B^2 D^2 L$ D = CHANNEL DIAMETER L = CHANNEL LENGTH (REFER ITEM #2) $\sigma \sim 1/\sqrt{P_{01}}$
4. INLET STAGNATION PRESSURE & PRESSURE RATIO	<ul style="list-style-type: none"> SPECIFYING P_{01} DETERMINES PRESSURE RATIO SINCE EXIT PRESSURE, P_{02}, IS DETERMINED BY GIVEN DIFFUSER EFFICIENCY P_{01} AFFECTS CHANNEL LENGTH (REFER ITEM #2) P_{01} AFFECTS MASS FLOW, POWER OUT & MAGNET / PLANT COST REQUIREMENTS ON ENTHALPY EXTRACTION & ISENTROPIC EFFICIENCY GOVERNS CHOICE OF PRESSURE RATIO 	P_{01} RELATED TO DIFFUSER RECOVERY FACTOR AS FOLLOWS $C_D = \frac{P_1 - P_2}{P_{01} - P_1} \approx 0.46 \text{ (FOR ETF)}$ $P_{01} = f(M_1) \approx P_1$ $\left. \begin{array}{l} \text{ENTHALPY EXTRACTION} \\ \text{ISENTROPIC EFFICIENCY} \end{array} \right\} = f(K, P_{01} / P_{02})$
5. ENTHALPY EXTRACTION & ISENTROPIC EFFICIENCY	<ul style="list-style-type: none"> THE LARGER THE CHANNEL SIZE ($\dot{m}$) THE MORE FAVORABLE IS CHANNEL SURFACE / VOLUME RATIO, THEREFORE, POTENTIAL ENTHALPY EXTRACTION & ISENTROPIC EFFICIENCY IS GREATER DEFINE PRESSURE RATIO FOR OPTIMUM THERMODYNAMIC PERFORMANCE & ELECTRO/GASDYNAMIC CONDITIONS 	<ul style="list-style-type: none"> APPROXIMATE APPLICABLE EQUATIONS FOR ADIABATIC PROCESS (SEE R. ROSA**): $\left. \begin{array}{l} \text{ENTHALPY EXTRACTION} \\ \text{ISENTROPIC EFFICIENCY} \end{array} \right\} = \frac{1 - \left[\frac{P_{02}}{P_{01}} \right]^{\eta_p (\gamma-1)/\gamma}}{1 - \left[\frac{P_{02}}{P_{01}} \right]^{\gamma-1/\gamma}}$ η_p = POLYTROPIC EFFICIENCY = $f(K, M)$ REFER ALSO TO DISCUSSION OF "MHD CHANNEL CALCULATIONS" IN SECTION 2.2.2
6. MASS FLOW RATE $\dot{m}$	<ul style="list-style-type: none"> MASS FLOW PROVIDES MARGIN IN ACHIEVING PERFORMANCE, I.E., CHOOSE $\dot{m}$ SLIGHTLY GREATER THAN MINIMUM NECESSARY TO ACHIEVE PERFORMANCE GOALS 	---
7. ELECTRICAL & POLYTROPIC EFFICIENCY	---	$\eta_{is} > \eta_o > \eta_p$ $\eta_p = \frac{\eta_o}{1 + \frac{\gamma-1}{2} M^2 (1-\eta_o)}$ $\eta_o = K$ - FARADAY LOADING PARAMETER

* FOR DETAILED DESCRIPTION OF CHANNEL DESIGN CALCULATION, REFER TO DISCUSSION OF "MHD CHANNEL CALCULATION", SECTION 2.2

** R. ROSA, "MHD ENERGY CONVERSION", MCGRAW-HILL BOOK CO., NEW YORK, N.Y.

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TABLE 2-4
ETF CHANNEL DESIGN
ELECTRICAL CONSTRAINTS & AERODYNAMICS

TRADE-OFFS				
DESIGN ITEM	DESIGN CHOICE	REASON(S) FOR DESIGN CHOICE	OTHER CONSIDERATIONS AND/OR DISADVANTAGES OF DESIGN CHOICE	ADDITIONAL COMMENTS
1. MACH NUMBER	SUBSONIC ($M < 1$)	RELATIVE TO SUPERSONIC FLOW, ADVANTAGES ARE: • GREATER CHANNEL/PLANT EFFICIENCY, GIVEN CONSTRAINTS ON E_x , ωT & J_y • BETTER CHANNEL PART-LOAD PERFORMANCE	LIMITED EXPERIENCE WITH SUBSONIC FLOWS AT HIGH MHD INTERACTION; APPLIES TO BOTH CHANNEL & DIFFUSER	IMPROVED PERFORMANCE USING SUBSONIC FLOW DIMINISHES AS PLASMA INLET STAGNATION TEMPERATURE INCREASES
2. MODE OF OPERATION	CONSTANT VELOCITY ($du/dx \approx 0$)	• YIELDS GOOD OVERALL PERFORMANCE; IN GENERAL, BETTER THAN ACCELERATING FLOW • AVOIDS SEPARATION OF BOUNDARY LAYER ON ELECTRODE WALL	DECELERATING FLOW ($du/dx < 0$) MAY YIELD BETTER PERFORMANCE BUT INVOLVES RISK OF BOUNDARY LAYER SEPARATION	--
3. MAXIMUM ALLOWABLE HALL ELECTRIC FIELD (ELECTRODES)	$E_x \approx 4 \text{ kV/m}$	APPROXIMATE STATE-OF-THE-ART	• SIGNIFICANTLY HIGHER VALUE RISKS SEVERE ARCING • SIGNIFICANTLY LOWER VALUE ADVERSELY AFFECTS CHANNEL SCALING	<ul style="list-style-type: none"> V_g AND E_x RELATED VIA SEGMENTATION PITCH USED: $V_g = p E_x$ CRITERION ON GAP VOLTAGE APPEARS TO TAKE PRECEDENCE HALL FIELD RELATED TO MAGNETIC FIELD: $E_x = \omega T \mu B(1-K) \sim B^2$ INTER-ELECTRODE ARC DRIVEN INTO ANODE WALL
4. MAXIMUM ALLOWABLE INSULATOR GAP VOLTAGE (ELECTRODES)	$V_g \approx 40 \text{ V/GAP}$	APPROXIMATE STATE-OF-THE-ART	• SIGNIFICANTLY HIGHER VALUE RISKS SEVERE ARCING • SIGNIFICANTLY LOWER VALUE ADVERSELY AFFECTS CHANNEL SCALING	
5. MAXIMUM ALLOWABLE TRANSVERSE CURRENT DENSITY	$J_y \approx 1.0\text{--}1.5 \text{ A/cm}^2$	APPROXIMATE STATE-OF-THE-ART	• SIGNIFICANTLY HIGHER VALUE RISKS SEVERE ELECTRODE WEAR • SIGNIFICANTLY LOWER VALUE ADVERSELY AFFECTS CHANNEL SCALING	<ul style="list-style-type: none"> J_y RELATED TO MAGNETIC FIELD $J_y = \sigma \omega B(1-K)$ DAMAGE FROM CURRENT TRANSPORT APPEARS RELATED TO CURRENT PER ELECTRODE; THEREFORE ALLOWABLE J_y RELATED TO SEGMENTATION PITCH: $I_y \sim p J_y$
6. MAXIMUM ALLOWABLE HALL PARAMETER	$\omega T \approx 4$	APPROXIMATE STATE-OF-THE-ART	SAME AS FOR E_x BUT IN ADDITION: • EFFECTS OF BULK PLASMA NON-UNIFORMITIES ARE MORE SEVERE THE HIGHER ωT ; DEPENDENCE IS ON $(\omega T)^2$	ωT IS A FUNCTION OF MAGNETIC FIELD & CHANNEL PRESSURE LEVEL: $\omega T \sim \frac{B}{p}$
7. MAGNETIC FIELD STRENGTH	PEAK OF 6 TESLA (AVERAGE OF 5 TESLA)	<ul style="list-style-type: none"> STATE-OF-THE-ART YIELDS ACCEPTABLE VALUES OF E_x AND E_x FOR CHOSEN CHANNEL PRESSURE LEVEL YIELDS REASONABLE CHANNEL LENGTH 	HIGHER MAGNETIC FIELD YIELDS MORE COMPACT CHANNEL: $\text{LENGTH} \sim \frac{1}{B^2}$	COST OF MAGNET IS A FUNCTION OF PRODUCT: $V_g B^2$, WHERE V_g = VOLUME OF WARM BORE; THEREFORE, MAGNET COST IS A FUNCTION OF CHANNEL POWER OUTPUT
8. EXIT STAGNATION PRESSURE	$P_{0f} \approx 1.2 \text{ ATM}$ (PRIOR TO DIFFUSER)	ENSURE RECOVERY VIA DIFFUSER TO ATMOSPHERIC PRESSURE	LOWER VALUE OF P_{0f} WOULD YIELD BETTER CHANNEL PERFORMANCE, ALL ELSE HELD CONSTANT, BUT WOULD REQUIRE MORE EFFICIENT DIFFUSER	ASSUMED DIFFUSER EFFICIENCY = 48% I.e., $C_p = 0.48 = \frac{P_2 - P_1}{P_{01} - P_1}$
9. SEGMENTATION (ELECTRODE PITCH)	PITCH, $p = 1.0\text{--}1.5 \text{ cm}$	<ul style="list-style-type: none"> APPROXIMATE STATE-OF-THE-ART YIELDS ACCEPTABLE GAP VOLTAGES AND CURRENT PER ELECTRODE 	FINER SEGMENTATION MEANS INCREASED MECHANICAL COMPLEXITY BUT WOULD HAVE ADVANTAGES IN: • LOWER CURRENT (I_y) PER ELECTRODE • HIGHER ALLOWED FREE-STREAM E_x	<ul style="list-style-type: none"> REFERE ITEMS # 3, 4, & 5 CURRENT PER ELECTRODE MAY ALSO BE REDUCED BY SEGMENTING PARALLEL TO B DIRECTION

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TABLE 2-5
ETF CHANNEL DESIGN
MECHANICAL DESIGN CONSIDERATIONS

TRADE-OFFS				
DESIGN ITEM	DESIGN CHOICE	REASON(S) FOR DESIGN CHOICE	OTHER CONSIDERATIONS OR DISADVANTAGES OF DESIGN CHOICE	ADDITIONAL COMMENTS
1. COOLING WATER	LOW PRESSURE BOILER FEED WATER	<ul style="list-style-type: none"> ● MAX. COOLING WATER TEMP $\approx 220^{\circ}\text{F}$. ALLOWS G-10, G-11 GLASS EPOXY TO BE USED IN WALL CONSTRUCTION ● G-10, G-11 STATE-OF-THE-ART MATERIALS IN AVCO CHANNEL DESIGNS ● ALLOWS USE OF AVAILABLE (INSULATING) RUBBER HOSE 	---	<ul style="list-style-type: none"> ● G-10, G-11 USED AS BACKING MATERIAL FOR ELECTRODE WALLS / INSULATOR WALLS ● STRENGTH/DEFLECTION CHARACTERISTICS OF LAMINATED PLASTICS LIMITS OPERATING TEMP. TO $\leq 300^{\circ}\text{F}$ ● READILY AVAILABLE RUBBER (INSULATING) HOSE LIMITED TO $\leq 250^{\circ}\text{F}$
2. BACKING MATERIAL FOR ELECTRODES	G-10, G-11 GLASS REINFORCED EPOXY	<ul style="list-style-type: none"> ● GOOD ELECTRICAL INSULATOR ● GOOD STRUCTURAL CHARACTERISTICS ● STATE-OF-THE-ART MATERIAL IN AVCO CHANNEL DESIGNS 	<ul style="list-style-type: none"> ● G-10, G-11 ARE CHARRING MATERIALS ● G-7 (SILICON BASE) IS NON-CHARRING BUT WEAKER & DIFFICULT TO MACHINE, DELAMINATES EASILY 	<ul style="list-style-type: none"> ● G-11 STRONGER THAN G-10 BUT MORE EXPENSIVE ● REFER ITEM # 1
3. DESIGN OF ELECTRODE & SIDE WALL ELEMENTS (INCLUDING MATERIAL)	COPPER BAR-SEGMENT	<ul style="list-style-type: none"> ● STATE-OF-THE-ART ● ADVANTAGES OF COPPER <ul style="list-style-type: none"> - READILY WATER COOLED - HEAT FROM ARCS RAPIDLY DISSIPATED THUS MINIMIZING DAMAGE FROM ARC PHENOMENA - LOW OPERATING TEMP MINIMIZES EROSION DUE TO CHEMICAL ATTACK (OXIDATION, ELECTROLYTIC ACTION) 	---	<ul style="list-style-type: none"> ● COPPER ELECTRODE DESIGN EVOLVED AFTER EXTENSIVE TESTING OF MANY ALTERNATE DESIGN CONCEPTS ● COPPER EXTRUSIONS READILY AVAILABLE
4. ELECTRODE-WALL MODE OF OPERATION (APPLIES ALSO TO SIDE WALLS EXCEPT AS NOTED)	SLAGGING	<ul style="list-style-type: none"> ● WALLS ARE SLAGGING, BY DEFINITION, REASONABLE ASSUMPTION BASED ON 2000 HR PROJECTED OPERATING TIME OF ETF CHANNEL ● SLAG LAYER REDUCES HEAT TRANSFER LOSSES TO COOLING WATER 	<ul style="list-style-type: none"> ● CHEMICAL ATTACK OF SUBSTRATE COPPER BY SLAG ● ON CATHODE WALL SLAG LAYER BECOMES CONDUCTIVE, SHORT CIRCUITS GROUPS OF ELECTRODES 	<ul style="list-style-type: none"> ● SURFACES ARE GROOVED & PRE-FILLED WITH ZIRCONIA TO PROVIDE ATTACHMENT POINT FOR SLAG ● PROBLEM OF CATHODE-WALL SHORTING NEEDS FURTHER INVESTIGATION
5. CLADDING OF ELECTRODES	<ul style="list-style-type: none"> ● ANODES MATERIAL RESISTANT TO OXIDATION & ELECTROLYTIC ACTION ● CATHODES MATERIAL RESISTANT TO ELECTROLYTIC ACTION 	<ul style="list-style-type: none"> ● PROTECTS AGAINST SURFACE OXIDATION, NEGATIVE OXYGEN IONS FROM PLASMA & SLAG ATTRACTED TO ANODE ● CANDIDATE MATERIALS Pt, Pd, Ni, Au, Ag, INCONEL ● PROTECTION AGAINST ELECTROLYTIC ACTION, UPSTREAM EDGES BOTH CATHODES & ANODES, SAME CANDIDATE MATERIALS AS ABOVE 	FABRICATION AND/OR MATERIAL COST	---
6. INSULATOR SPACERS (ELECTRODE WALLS)	BORON NITRIDE (BN) DENSE (HARD) SPACERS	STATE-OF-THE-ART MATERIAL IN AVCO CHANNEL DESIGNS <ul style="list-style-type: none"> ● GOOD ELECTRICAL INSULATOR ● GOOD THERMAL CONDUCTIVITY, COOL INSULATOR MINIMIZES ELECTROLYTIC EROSION OF ELECTRODES 	BN IS EXPENSIVE MATERIAL BUT EASILY MACHINED	<ul style="list-style-type: none"> ● BN IS NOT WETTED BY SLAG ● OTHER CERAMICS, PARTICULARLY CASTABLES, TEND TO BE DISSOLVED BY SLAG
7. INSULATOR SPACERS (SIDE WALLS)	BORON NITRIDE (BN) DENSE (HARD) SPACERS	STATE-OF-THE-ART MATERIAL IN AVCO CHANNEL DESIGNS <ul style="list-style-type: none"> ● GOOD ELECTRICAL INSULATOR ● GOOD THERMAL CONDUCTIVITY, COOL INSULATOR MINIMIZES ELECTROLYTIC EROSION OF ELECTRODES 	USE OF, FOR EXAMPLE, ALUMINA DENSE (HARD) SPACERS MAY BE PERMISSABLE FOR NON-PEG WALL CHANNEL, BUT COST BENEFIT IS QUESTIONABLE	USE OF BN FOR PEG WALL APPEARS NECESSARY TO PROTECT O-RING SEAL, I.E., PROVIDE ADEQUATE COOLING BECAUSE OF GOOD THERMAL CONDUCTIVITY OF BN

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Table 2-3 - Summary of various factors considered in the parametric analysis: mass flow, pressure ratio, etc., including tradeoffs.

Table 2-4 - Summary of channel aerodynamic consideration and electrical constraints, including tradeoffs.

Table 2-5 - Summary of channel mechanical design features, including tradeoffs.

Results of the parametric studies are shown in Figures 2-2 and 2-3 as a function of mass flow rate. The upper curves are for 3000°F preheat and the lower ones are for 2500°F preheat. Both the power output and the enthalpy extraction increase with increasing mass flow rate. It is seen that the relative gain in enthalpy extraction becomes markedly less beyond a mass flow rate of 65 kg/s as indicated by the knee bends of the two curves in Figure 2-3. At this mass flow rate the calculated values of enthalpy extraction and isentropic efficiency (Table 2-1) satisfies our target values of 50% and 12.5% respectively for these two important channel performance parameters. It is also noted that the target values are reached at the lowest preheat temperature and that certain margin is provided. Therefore, a total mass flow rate of 65 kg/s was selected for the ETF channel. The reasons can be summarized as follows:

1. It satisfies the target values of enthalpy extraction (12.5%) and isentropic efficiency even at the lower preheat temperature of 2500°F.
2. It provides a certain margin over what is considered the minimum channel performance requirements.
3. The relative gain in performance beyond 65 kg/s is relatively small.

The remaining paragraphs of this section deal briefly with the subject of channel configuration, and with the design of gas-side surfaces in a slagging environment.

2.1.1.1 Channel Configuration

The statement-of-work requires a description of a Faraday type channel, and both a single load and multiload diagonal generator. The channel configuration ultimately incorporated in the ETF and a baseload

Figure 2-2 ETF MHD Channel Outputs as a Function of Mass Flow Rate for 2500°F and 3000°F Preheat Temperatures

Figure 2-3 ETF Channel Enthalpy Extraction (η_e) as a Function of Mass Flow Rate for 2500°F and 3000°F Preheat Temperatures

plant may well be a hybrid; namely a design that combines features of each individual channel type. It is useful to review the advantages and disadvantages of each type.

Three variations of the Faraday MHD channel concept are shown schematically in Figures 2-4(a), (b) and (c). The differences between the three are only in a specific design of the insulator wall (or side wall). Illustrated in Figure 2-4(a) is the classic multiple-load Faraday design using metallic peg wall construction. In general, the segmentation of the pegs is about equal to that of the electrodes so that gap voltages, in all directions parallel to the plane of the side wall, can be kept to a minimum. Square pegs are illustrated but polygon cross sections can be and have been used. An all-ceramic side wall and peg wall are approximate electrical equivalents, but the feasibility of ceramic walls for long duration has not yet been demonstrated. In contrast, several promising peg wall schemes have been evolved and tested.

The advantage of the peg-wall Faraday channel is that its performance is generally considered to be the best of all possible channel types. It represents the standard against which all other channel types are judged for both full load and part-load operation. Its disadvantage is associated with the mechanical complexity and the question of long term reliability of a peg wall. The number of pegs is large. Provision for adequate and reliable cooling can be a formidable design problem, given the mechanical and electrical stress conditions present in an MHD generator.

Figure 2-4(b) shows another Faraday type channel with its characteristic loading of individual electrode pairs. In this instance, the insulator wall is constructed of long bars rather than pegs. Some segmentation of the bars may be incorporated to control gap voltages and current densities. The bars are oriented along the equipotentials predicted for the free-stream plasma. The orientation angle, ϕ , of the bars with respect to the $\vec{U} \times \vec{B}$ direction is given by the equation:

$$\phi \equiv \arctan E_y/E_x = \arctan \left[\frac{K}{\omega \tau (1 - K)} \right]$$

where $\omega \tau$ is the Hall parameter and K is defined as the loading parameter or, $K = E_y/uB$. The gross performance of the Figure 2-4(b) channel is equivalent to that of the peg wall type of Figure 2-4 (a), at the full load or

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Figure 2-4 Faraday Channels

design point. The obvious and demonstrated advantage of a bar wall is the mechanical simplicity and reliability of this kind of construction. Its disadvantage is in part-load (or off-design) operation where the fixed bars will be at an incorrect angle. The resulting circulation of Hall current is likely to make performance less than that of a part-load, peg wall design.

A possible compromise for side wall construction suggests itself for a Faraday channel, and this is depicted in Figure 2-4(c). In this concept the bars are highly segmented but the segmentation is not as great as that of a peg wall. Namely, the length to width ratio, l/w , of a typical "short bar" or "long peg" is greater than one. If $l/w = 1$, then one has a true peg wall. The idea here is to try and approach a peg wall design with its good part-load performance, and yet retain some of the mechanical simplicity of bar wall construction. This concept needs further investigation within the context of part-load performance requirements.

Two variations of the basic single-load diagonal channel are illustrated in Figure 2-5. Figure 2-5(a) shows the window frame design in which each electrode/bar pair is an integral unit; no external connections are necessary. The window frame technique appears to represent a simple and low cost type of construction; however this advantage may be more apparent than real, particularly for large channels of ETF and baseload size. In any case, window frames have a serious fault in that severe current concentrations tend to exist on the bars near the corners (assuming a channel flow cross section that is either square or rectangular). These current concentrations cause unacceptable long term damage as has been observed in experiments performed at Avco. To alleviate the difficulty, the troublesome current path is interrupted by separating the bars from the electrodes, and each bar is split into at least two elements. The electrode pair previously connected by virtue of the window frame technique must now be connected by external wires.* This

* The existence of electrode voltage drops means that cross-channel equipotentials are roughly in shape of the letter "Z". In practice, one might stagger cross-channel connections to minimize gap voltages. Namely, referring to Figure 2-5(b), one would externally connect a to 1'; b to c'; and so forth.

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Figure 2-5 Single-Load Diagonal MHD Channels

arrangement is shown in Figure 2-5(b). In this design the window frame method technique cannot be used, but the basic mechanical simplicity of a diagonal channel is retained.

There are two disadvantages to the single load diagonal channel concept. First, even at the full load design point, ideal single load diagonal channel performance will be less than that of the multiloaded Faraday machines. This is so because one cannot adjust the loading parameter to an arbitrary distribution along the channel's length. The single load precludes this option. Secondly, part-load performance may be significantly less than that of any of the Faraday options discussed previously and shown on Figure 2-4. Multiple loading offset these disadvantages. Two multiple loading schemes are depicted on Figure 2-6. These apply to either window frame or split bar designs. As the number of loads, L_n , approaches the number of electrode pairs, one has in the limit the approximate equivalent of a bar-wall Faraday machine (Figure 2-4(b)). The pros and cons of the latter were discussed previously.

It seems clear that if full load MHD channel operation were the only mode being considered, then one would tend to choose for a baseload MHD plant either: (1) the bar-wall Faraday type channel, with multiple loads and some bar segmentation; or (2) the split bar diagonal channel with several loads. (A two terminal diagonal channel for baseload application is not considered practical, because of its lower efficiency compared to a multiterminal channel.) These two channel types are nearly identical in mechanical design and differ primarily in the loading scheme employed. The development of practical consolidation circuitry at a reasonable cost would throw weight toward the Faraday design. But, the requirement for good part-load performance complicates the situation. Calculations have been performed at Avco showing that a Faraday type channel (assuming peg walls) has excellent part load performance characteristics. These computations are summarized in a later section. If, for example, the latter two prove to have unacceptable part-load performance, then one is driven toward the Faraday option incorporating either true pegs or short bars. Alternately, of course, bar wall design may prove to have good part-load characteristics. Evaluation of part-load channel characteristics involves not only gross performance figures, such as enthalpy extraction, but also the important matter of excessive gap

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Figure 2-6 Multi-Load Diagonal MHD Channels

voltage and/or excessive local current densities that might result from operation at off-design conditions.

2.1.1.2 Gas-Side Surface

The gas-side surfaces of the channel (electrode walls and insulating walls) are designed to operate coated with slag. The principal advantages of a slag coating are: (a) reduction of heat transfer losses by virtue of the slag's high gas-side surface temperature, and (b) protection of the electrode (or substrate) material against erosion due to severe arcing phenomena. Here, the term arcing refers to either a strong pinching of transverse current (I_y) at the slag surface or within the slag layer itself; or an electrical breakdown between adjacent elements making up a channel wall. Both effects may occur simultaneously and, for an electrode wall, may be aggravated by the usual current concentration associated with the Hall effect. Arcs are merely regions of high current density (A/cm^2) and the principal mechanism for damage is a local high heating rate which melts or boils away material. In the special case of the anode wall, a breakdown arc between adjacent elements tends to be driven into the wall by the local electromagnetic body force ($\vec{J} \times \vec{B}$). Erosion is also caused by oxidation and by electrolytic action; these will be discussed below.

The attainment of diffuse or arc-free transport of transverse current from the plasma to the electrodes has proven to be an elusive goal. In fact, arc transport may always be present for the electrodynamic conditions projected for commercial MHD channels. Therefore, an additional ground rule in the present Avco ETF channel designs is that arc transport will always exist to some degree, and must be taken into account in electrode design. A slag layer can be helpful in achieving the goal of a long-lived electrode. Arcs may destroy the layer locally, but such areas can be "healed" by deposition of slag from the free stream plasma.

Other observed and important mechanisms causing electrode damage are erosion by oxidation and by electrolytic action. Oxidation is significant on the anode wall since negative oxygen ions, present in both the plasma and the slag coating, are attracted. Electrolytic erosion occurs on both anode and cathode walls. Each electrode element on either wall is at a positive potential relative to its immediate, upstream neighbor. Therefore, the

upstream face of each element acts as an anode with respect to the upstream element, and experiences an electrolytic loss of material. Molten slag, with or without dissolved seed, is the electrolyte. Maintaining the electrode material at a low temperature minimizes loss of material by both of the aforementioned mechanisms. Low temperature causes a reduced rate of chemical activity and in addition, the slag will freeze if it is in good contact with the cold surfaces. However, current concentrations due to arc-mode transport or the Hall effect will tend to locally liquify the slag (Hall effect current concentrations occur on the upstream edge of anodes and the downstream edge of cathodes).

The basic electrode design evolved at Avco after extensive testing is a water-cooled copper bar, clad with a thin layer of oxidation resistant material over vulnerable surfaces (e.g., Figures 2-52 and 2-54). Cladding materials under consideration include platinum, palladium, gold, silver, inconel and nickel. Extensive long duration testing is necessary to determine which materials are best, considering both technical performance and cost. To date, good results have been achieved with platinum-clad electrodes, which have been operated successfully at AERL for 250 hours under electrical and thermal conditions approximating those in the ETF channel. The plasma-side surface of the electrode has grooves, prefilled with ceramic (e.g., zirconia), to provide attachment point for the deposited slag and thereby facilitate development of a slag layer. The high thermal conductivity and high thermal diffusivity of the copper substrate makes it more suitable as an electrode than other materials. Specifically, the local high heating associated with arcing phenomena is more readily dissipated in copper. Arc damage is thereby alleviated or avoided. Boron nitride (BN) hard spacers are employed as insulators in present Avco electrode wall designs. BN is a good thermal conductor, is resistant to thermal shock, and is not wetted by coal slag.

Potential arc damage is also influenced by electrode segmentation, both in the flow direction and parallel to the magnetic field vector. First, the possible damage associated with arc transport of the transverse current tends to be dependent on the absolute magnitude of the current (I_y) entering a single electrode. Secondly, the so-called fault power associated with inter-electrode breakdown is proportional to the product of electrode current and nominal gap (Hall) voltage:

$$\text{Fault Power} \sim I_y V_x = J_y E_x L p^2.$$

L is the length of the electrode in the $\vec{B}$ direction, and p is the pitch in the flow direction. In MHD channel design, J_y and E_x are usually specified. Fault power, therefore, can be lowered by reducing L and P .

Although it was stated that all ETF channel surfaces will be coated with slag, it is important to describe an experimental observation that may support arguments for a non-slagging cathode. On Avco's Mk VI channel experiment it has been observed that the slag layer on the cathode wall short-circuits groups of electrodes in well defined intervals down the length of the channel (gross channel performance remains unaffected). It is speculated that this may be due to absorption of positive potassium (seed) ions. The slag short-circuits some number of electrodes until the accumulated and concentrated current on a single electrode is sufficient to burn through the layer to the metallic surface. A relatively large voltage is simultaneously accumulated across a single insulator gap. So far, at least for Mk VI Faraday configured channels, this phenomenon does not appear to have an adverse effect on channel life. But for duration times $\gg 100$ hrs and approaching those required for ETF channels, cathode slag-shortening may prove to be of great significance. For example, the high accumulated gap voltage may increase electrolytic erosion. To eliminate or control cathode slag-shortening, several options are available. The first is to design a non-slagging cathode which essentially means a smooth, cold surface. It may prove, however, that erosion by slag particle impact will be unacceptable; or that pitting of the surface by impact and subsequent filling of the voids by slag will result in the formation of a layer anyway. Alternately, one might have to accept the existence of a cathode slag layer and attempt to devise external circuitry to control or distribute any damage effects associated with slag shorting.

The Avco state-of-the-art design for insulator walls is the same as that for the electrodes, and essentially for the same reasons. Namely, water cooled copper bar-segments are used, with grooves on the gas side surface to facilitate slag layer development. Dense aluminum oxide spacers may be used as gap insulators rather than BN to minimize cost. The various erosion difficulties are not as severe on insulator walls compared to electrode walls. But in the special case of a peg wall, experimental evidence indicates that BN is necessary. BN's capability to be better cooled than Al_2O_3 , and its good resistance to thermal shock affords better protection of the numerous O-ring seals associated with a peg wall.

2.1.2 Working Fluid

Thermodynamic properties of the combustion gases produced by burning coal with air and oxygen enriched air were calculated for use in the ETF design work. Table 2-6 lists the calculations made for the two coals specified by the ETF work statement.

2.1.2.1 Results

Figures 2-7, 2-11 and 2-14 are mollier charts computed for Montana subbituminous and Illinois No. 6 coals respectively. Seed mass, coal moisture content, and air equivalence ratios are given by Table 2-5, Item 1. Figure 2-15 shows enthalpy as a function of temperature for 5% moist Montana coal, 95% equivalence ratio, and 6 atm pressure. Figure 2-16 is similar, but includes 100% equivalence ratio. Pressure is 1 atm. Figures 2-17 and 2-18 extends Figure 2-16 to lower temperatures. Additional equivalence ratios corresponding to 5% and 10% excess air are included in these latter figures.

Figures 2-19, 2-20 and 2-21 show the variation of calculated adiabatic flame temperatures with coal moisture content, air equivalence ratio, and air preheat as specified by Table 2-6, Item 3.

Figures 2-22 and 2-23 shows calculated flame temperature for low equivalence ratios which might be encountered in the first stage of a two-stage burner. Item 4 of Table 2-5 stipulates conditions.

Figure 2-24 shows the variation of adiabatic flame temperature with oxidizer preheat temperature and various amounts of oxygen enrichment. The baseline curve is for air at 21.2% oxygen by volume. Figure 2-25 shows the variation of the total mass flow rate of combustion products for various degrees of oxygen enrichments for constant coal feed rate.

Figures 2-26 and 2-29 are mollier charts for Montana subbituminous coal dried to 5% moisture, at 0.95 equivalence ratio, and 1% K for the various percents of oxygen enrichment given by Item 7 of Table 2-6.

TABLE 2-6
COAL COMBUSTION CALCULATIONS

Item No.	Calculation	Montana Sub-bituminous Coal	Illinois No. 6 Coal
1.	Mollier charts		
	Pressure range	0.6 - 12 atm	0.6 - 12 atm
	Temperature range	1700°K - 3100°K	1700°K - 3100°K
	Mass of potassium	1%	1%
	Coal moisture content	2%, 5%	1%, 2%
	Equivalence ratio, air/fuel	0.95, 1.00	0.95, 1.00
	Air composition	NEMA	NEMA
	Figure Nos.	1-4, 9-11	5-8
2.	Mollier chart extension		
	Pressure	1 atm	
	Temperature range	500°K - 1600°K	
	Mass of potassium	1%	
	Coal moisture content	5%	
	Equivalence ratios	0.95, 1.00, 1.05, 1.10	
	Figure Nos.	9, 10, 11, 12	
3.	Flame temperature, sensitivity to coal moisture content		
	Pressure	5.4 atm	5.4 atm
	Mass of potassium	1%	1%
	Coal moisture content	0 - 25%	0 - 25%
	Equivalence ratios	0.95, 1.00	0.95, 1.00
	Air preheat temp.	2500°F, 3000°F	3000°F
	Figure Nos.	13, 15	14
4.	Flame temperature, effect of equivalence ratio		
	Pressure	6 atm	
	Mass of potassium	1%	
	Coal moisture content	5%	
	Equivalence ratios	0.35, 0.40, 0.50, 0.60	
	Air preheat	2500°F, 3000°F	
	Figure Nos.	16, 17	
5.	Adiabatic flame temperature as function of oxidizer preheat temperature and % O ₂ by volume		
	Pressure	6 atm	
	Mass of potassium	1%	
	Coal moisture content	5%	
	Equivalence ratio	0.95	
	Air preheat	500 - 3000°F	
	% O ₂ by volume	21.2% (air) - 50%	
	Figure No.	18	
6.	Fraction of air/coal combustion product mass flow rate as function of % O ₂ by volume in oxidizer	Same conditions as item 5	
	Figure No.	19	
7.	Mollier Charts		
	Pressure	0.6 - 12 atm	
	Temperature	2000 - 3000°K	
	Coal moisture content	5%	
	Equivalence ratio	0.95	
	% O ₂	35.7, 33.3, 31.3, 29.4	
	Figure Nos.	20, 21, 22, 23

Figure 2-7 Mollier Chart, Montana Sub-Bituminous Coal, 2% Moisture Content, Air/Fuel Equivalence Ratio 0.95, 1% K by Mass

Figure 2-8 Mollier Chart, Montana Sub-Bituminous Coal, 5% Moisture Content, Air/Fuel Equivalence Ratio 0.95, 1% K by Mass

Figure 2-9 Mollier Chart, Montana Sub-Bituminous Coal, 2% Moisture Content, Air/Fuel Equivalence Ratio 1.0, 1% K by Mass

Figure 2-10 Mollier Chart, Montana Sub-Bituminous Coal, 5% Moisture Content, Air/Fuel Equivalence Ratio 1.0, 1% K by Mass

Figure 2-11 Mollier Chart, Illinois No. 6 Coal, 1% Moisture Content, Air/Fuel Equivalence Ratio 0.95, 1% K by Mass

Figure 2-12 Mollier Chart, Illinois No. 6 Coal, 2% Moisture Content,
Air/Fuel Equivalence Ratio 0.95, 1% K by Mass

Figure 2-13 Mollier Chart, Illinois No. 6 Coal, 1% Moisture Content, Air/Fuel Equivalence Ratio 1.0, 1% by Mass

Figure 2-14 Mollier Chart, Illinois No. 6 Coal, 2% Moisture Content, Air/Fuel Equivalence Ratio 1.0, 1% K by Mass

H1291

Figure 2-15 Enthalpy as Function of Temperature at 6 Atm, Combustion Product of Montana Sub-Bituminous Coal with 5% Moisture Content, 95% Stoichiometric Air, 1% K (In K_2CO_3 Seed)

H1290

Figure 2-16 Enthalpy as Function of Temperature at $p = 1$ Atm, Combustion Products of Montana Sub-Bituminous Coal Dried to 5% Moisture Content, Stoichiometric and 95% Stoich. Air, 1% K by Weight

HI287

Figure 2-17 Enthalpy as Function of Temperature at 1 Atm, Combustion Products of Montana Sub-Bituminous Coal with 5% Moisture Content

HI293

Figure 2-18 Enthalpy as Function of Temperature at $p = 1$ Atm, Combustion Products of Montana Sub-Bituminous Coal Dried to 5% Moisture Content with Various Air Ratios, 1% K by Weight

H1292

Figure 2-19 Flame Temperature as a Function of Coal Moisture Content for Montana Sub-Bituminous Coal, 95% Stoichiometric Air Preheated to 3000°F, 100% Stoichiometric Air, 1% K by Weight

H1289

Figure 2-20 Flame Temperature as a Function of Coal Moisture Content for Illinois No. 6 Coal, 95% Stoichiometric Air Preheated to 3000°F, 100% Stoichiometric Air, 1% K by Weight

HI286

Figure 2-21 Flame Temperature as a Function of Coal Moisture Content for Montana Sub-Bituminous Coal, 95 % Stoichiometric Air Preheated to 2500°F, 100 % Stoichiometric Air Preheated to 2500°F, 1% K by Weight

H1288

Figure 2-22 Effect of Air/Fuel Equivalence Ratio on Flame Temperature in 1st Stage of Combustor for Montana Sub-Bituminous Coal, 5% Moisture Content, 1% K by Weight

Figure 2-23 Adiabatic Flame Temperature vs Air/Fuel Equivalence Ratio for Various Air Preheat and Burner Pressures

Figure 2-24 Adiabatic Flame Temperature as a Function of Oxidizer Preheat Temperature and % O_2 by Volume, Montana Sub-Bituminous Coal, 5% Moisture Content, 0.95 Air/Fuel Equivalence Ratio, 1% K by Mass, Total Pressure = 6.0 Atm

Figure 2-25 Fraction of Air/Coal Combustion Product Mass Flow Rate as Function of Percent O_2 in Oxidizer

Figure 2-26 Mollier Chart, Combustion Products of Montana Sub-Bituminous Coal and Air with Oxygen Enrichment
 5% Moisture Content
 0.95 Equivalence Ratio
 1% K by Mass
 35.7% O₂

Figure 2-27 Mollier Chart, Combustion Products of Montana Sub-Bituminous Coal and Air with Oxygen Enrichment
 5% Moisture Content
 0.95 Equivalence Ratio
 1% K by Mass
 33.3% O₂

Figure 2-28 Mollier Chart, Combustion Products of Montana Sub-Bituminous Coal and Air with Oxygen Enrichment
 5% Moisture Content
 0.95 Equivalence Ratio
 1% K by Mass
 31.3% O₂

Figure 2-29 Mollier Chart, Combustion Products of Montana Sub-Bituminous Coal and Air with Oxygen Enrichment
 5% Moisture Content
 0.95 Equivalence Ratio
 1% K by Mass
 29.4% O₂

2.2 MHD CHANNEL CALCULATIONS

2.2.1 Calculation Procedure

Four MHD channel computer programs are used for the ETF channel design calculations. These are referred to as PDG 73, PDG 79, PDG 95 and MHD 2. Program PDG 73 is used for establishing initial channel design. It calculates the channel loft and operating parameters for specified channel inlet conditions, axial velocity profile magnetic field distribution, and to electrical loading. This program assumes segmented Faraday mode of operation. Programs PDG 79 and PDG 95 are used to calculate channel operating parameters for a specified channel loft. These allow off-design performance to be calculated for a channel of specified geometry. Program PDG 79 is used for channels operating in the segmented Faraday mode, and Program PDG 95 is used for channels operating in the diagonally connected mode. The PDG series programs concentrate on the channel core flow, and have a relatively crude treatment of the channel boundary layer. These programs permit calculations of the basic channel loft and operating characteristics, at relatively low cost in computer time.

The basic channel calculation procedure for the PDG series programs is schematically outlined in the diagram on Figure 2-30. Both the electrical and gasdynamic calculation subroutines of the model separate the flow into a uniform core and a boundary layer. An additional subroutine calculates the composition of the plasma assuming chemical equilibrium at each step in the channel integration.

The inlet boundary conditions are established by the various selected system parameters for fuel, oxidizer, seed, etc., together with assumed combustion design conditions. This also defines the thermodynamic thermochemical and electrical properties of the working fluid. The exit pressure is determined by the diffuser performance and pressure loss in the downstream bottoming plant and exhaust system.

Program MHD 2 incorporates a more sophisticated treatment of channel wall phenomena including slagging effects, and boundary layer losses. These refinements are obtained at the expense of additional complexity in calculation procedures and the use of more computer time. Because of this, the MHD 2 program is used for more detailed calculations of boundary layer losses, heat transfer and overall channel performance, in

- (1) THE DESIGN PROGRAM CALCULATES CHANNEL LOFT FOR A SPECIFIED VELOCITY DISTRIBUTION
THE OFF DESIGN PROGRAM CALCULATES VELOCITY ETC FOR A GIVEN CHANNEL LOFT
- (2) ELECTRODE AND BOUNDARY LAYER VOLTAGE LOSS AS SPECIFIED BY AN EQUATION BASED ON
EXPERIMENTAL MEASUREMENT AS WELL AS THEORETICAL CALCULATION
- (3) BARRED QUALITIES ARE AVERAGES OVER THE CHANNEL CROSS SECTION
- (4) SEPARATE PROGRAMS ARE USED FOR THE CALCULATIONS OF THREE DIMENSIONAL CURRENT
DISTRIBUTIONS AND END EFFECTS

Figure 2-30 Basic MHD Channel Calculation Procedure

order to establish these with a higher degree of accuracy after basic channel design and operating characteristics have been obtained by the use of the PDG series programs.

Appendix A contains more detailed information on the channel computer programs.

2.2.2 Selection of Basic Design Parameters

The purpose of the following discussion is to review briefly some of the factors involved in optimization of the design of an MHD generator for central station power plant application. As an aid in this discussion some approximate calculations are presented to indicate the values of the parameters involved. However, the final choice of optimum design parameters must in each case be made on the basis of detailed calculations of both generator and overall cycle performance.

The overall cycle efficiency of an MHD/Steam Power Cycle is, in simple terms:

$$\eta = \frac{P_{\text{MHD}} + P_{\text{steam}} - P_{\text{aux}}}{Q_{\text{fuel}}}$$

where:

- P_{MHD} = net power output from MHD generator
- P_{steam} = gross power output from steam plant
- P_{aux} = compressor + plant auxiliary power (incl. inverter losses)
- Q_{fuel} = fuel heat input

The basic way to increase the cycle efficiency is to raise the relative fraction of power from the MHD generator. Thus, in order to optimize the thermal efficiency of an MHD power plant, one seeks to obtain a relatively high percentage of enthalpy extraction from the gas in the MHD-generator and to do so at the highest possible machine efficiency (isentropic efficiency).

In the design of the generator, this means to seek to employ a relatively high pressure ratio, relatively high loading parameter and relatively low Mach number. In addition, in order to optimize channel reliability and lifetime relatively low current density and low voltage gradients are sought. All of these factors tend to increase the length of the generator.

Other factors, on the other hand, favor a short length for the generator. A decrease in generator length decreases heat loss, friction loss, and capital cost, in particular the cost of the magnet. Last but not least, generator construction materials and fabrication methods must be taken into account. All of this means that a compromise must be reached between all factors involved in channel design in order to optimize generator performance, size and length.

The magnitude of the gas conductivity is of particular importance for this problem. An estimate of its importance can be obtained by examining the equations for the heat transfer, friction loss, and electrical power output. These are:

$$\text{Electrical} \quad \rho u \frac{dh_o}{dx} = k(1-k) \frac{\sigma}{G} u^2 B^2 \quad (2-1)$$

$$\text{Friction} \quad \left(\frac{dP_o}{dx} \right)_{\text{friction}} = \frac{4}{D_h} C_f \frac{1}{2} \rho u^2 \quad (2-2)$$

$$\text{Heat Loss} \quad q = St \rho u (h_\infty - h_w) \quad (2-3)$$

where:

- ρ = gas density
- u = gas velocity
- h_o = gas stagnation enthalpy
- k = load parameter
- σ = gas electrical conductivity
- B = magnetic field strength
- G = a nonuniformity factor
- A = flow cross-section
- D_h = channel hydraulic diameter = $4 \times \text{flow cross section} \div \text{perimeter}$
- C_f = friction factor
- St = Stanton number
- h_∞ = gas enthalpy in the core
- h_w = gas enthalpy at the wall

From these basic equations the following equations can be derived for:

Electric power output IP_m per unit length,

$$\frac{d IP_m}{dx} = A k (1-k) \frac{\sigma}{G} u^2 B^2 \quad (2-4)$$

Heat loss IP_q per unit length,

$$\frac{d IP_q}{dx} = \frac{4A}{D_h} St \rho u (h_\infty - h_w) \quad (2-5)$$

Friction loss IP_f per unit length,

$$\frac{d IP_f}{dx} = \frac{4A}{D_h} \left(\frac{\rho}{\rho_o} \right) k C_f \frac{1}{2} \rho u^3 \quad (2-6)$$

Here the stagnation pressure loss due to friction has been converted into an equivalent power loss using the relation,

$$d IP_f = \frac{\rho}{\rho_o} k A u dP_o \Big|_{\text{friction}}$$

Thus, for a given stagnation pressure increment, IP_f is the additional useful power the generator could have delivered if friction loss had been absent.

The ratio between friction and heat loss is, from Eqs. (2-5) and (2-6);

$$\frac{d IP_f}{d IP_q} = \frac{\rho}{\rho_o} K \left(\frac{C_f/2}{St} \right) \frac{u^2}{h_\infty - h_w} \quad (2-7)$$

Under typical conditions, the right-hand side is never far from unity so that losses due to friction and heat tend to be of comparable size. Assuming that they are of equal size, then the ratio of heat plus friction loss to electrical output can be expressed using Eqs. (2-4) and (2-5) as the following:

$$\frac{d (IP_q + IP_f)}{d IP_m} \approx \frac{8 St G \rho (h_\infty - h_w)}{D_h k(1-k) \sigma u B^2} \quad (2-8)$$

Typical channel end conditions considered for the ETF MHD generator size with total thermal input to the channel in the order of 250 MW are:

$$\begin{array}{ll}
 D_h \approx 0.9 \text{ meter} & \rho \approx 0.13 \text{ kg/m}^3 \\
 St \approx 0.001 - 0.002 & h_\infty - h_w \approx 200 \text{ cal/gm} \\
 k \approx 0.70 & B \approx 4 \text{ Tesla} \\
 u \approx 800 \text{ m/s} & G \approx 2.5
 \end{array}$$

Hence;

$$\frac{d IP_q}{d IP_m} \approx \frac{1-2}{\sigma}$$

Thus, in a generator of the ETF size considered, the incremental losses due to friction and heat losses will approach the incremental power output when the conductivity at the channel exit approaches about 1-2 mho/m.

An increase of viscous losses leads to a decrease of the generator machine efficiency and to an increase in the required compressor power. Furthermore, the utilization of heat losses from the channel becomes less efficient as the heat loss increases since this effects and decreases the obtainable steam bottoming cycle efficiency. Therefore, the required channel exit gas conductivity and corresponding temperature is expected to be somewhat higher than that calculated above so as to avoid inefficiencies in generator and overall plant performances. A similar example as above can also be calculated for any other generator size, such as for a full scale generator.

Here, it would also be important to compare the incremental power output with incremental equipment costs, in particular for the magnet. For the ETF, these incremental costs are also important but technical considerations in this case are more important.

The choice of Mach number is influenced by the following considerations:

- a) Minimum heat plus friction loss. This is obtained at the velocity that makes heat and friction loss equal, typically $M \approx 1$.
- b) Minimum length. This is obtained at the velocity that maximizes average $\sigma u / \rho$; for ETF conditions, this is typically supersonic with $M \approx 1.5$.

- c) Minimum diffuser loss is obtained at the lowest possible Mach number.
- d) Performance is not accurately predictable for Mach numbers close to one.
- e) At present, there is considerably more MHD experience at $M > 1$.

Although items (b) and (e) favor $M > 1$, the rest favor $M < 1$, so that in general subsonic velocities are considered for efficient operation. It is also noted that part load performance appears to be favored by subsonic operation because preliminary investigations indicate that part load operation can be obtained with no substantial decrease in enthalpy extraction ($\sigma u/\rho \approx \text{constant}$) or generator performance. However, it should be noted that as the preheat temperature increases the difference in viscous losses between supersonic and subsonic operation diminishes.

The exact choice of Mach number depends upon many factors, including subjective ones such as item e, and poorly understood ones such as diffuser performance under MHD conditions. Thus, detailed machine design calculations must be complemented with a subjective assessment of channel design and operating parameters according to MHD experience.

The choice of loading parameter, K , must be based upon a compromise between length, and the length-dependent losses due to friction and heat transfer on the one hand, and electrical losses on the other hand. An estimate of the optimum load parameter can be obtained as follows:

The stagnation pressure drop in the generator due to power generation and friction is given by,

$$\delta P_o = \frac{\rho_o}{k} \delta h_{oe} - \frac{4}{D_h} C_f \frac{1}{2} \rho u^2 \delta x \quad (2-9)$$

where δh_{oe} is the change in stagnation enthalpy due to power generation only. Using the energy equation,

$$-\delta h_{oe} = k(1-k) \frac{\sigma}{G} \frac{uB^2}{\rho} \delta x \quad (2-10)$$

to eliminate dx in Eq. (2-9) gives,

$$\delta P_o = \left[\frac{\rho_o}{k} + \frac{2 C_f \rho^2 u}{D_h \sigma / G B^2 k(1-k)} \right] \delta h_{oe} \quad (2-11)$$

It can be seen that, for a given δh_{oe} , δP_o will be infinite for $k = 0$ and 1 and must have a minimum between. Differentiating with respect to k , and assuming, to first order, that none of the other parameters varies as k varies,

$$k_{opt}^2 - 2 C k_{opt} + C = 0 \quad (2-12)$$

where:

$$C \equiv 1 + \frac{2 C_f \rho^2 u G}{D_h \rho_o B^2 \sigma} \quad (2-13)$$

The solution is;

$$k_{opt} = C \left[1 \pm \sqrt{1 - \frac{1}{C}} \right]$$

Typical parameter values for the channel exit and entrance, and the resulting values of optimum k are;

	Entrance	Exit
C_f	0.004	0.004
D_h	0.5	1
B	6	4
ρ/ρ_o	1	1
ρ	0.75	0.15
u	1000	1000
k_{opt}	0.85	0.79

In a more comprehensive optimization study, electrical losses in the boundary layers, wall leakage, growth of the nonuniformity factor, G , wall heat loss, and capital costs must be considered. These will tend to reduce the optimum value of k . On the other hand, for a given amount of overall enthalpy extraction, increased k results in a lower channel inlet pressure, hence lower density, ρ and larger hydraulic diameter, D_h .

2. 2. 3 Commercial Baseload Channel Calculations

The basis and objective of the baseload MHD channel calculations described in this section has been presented in Section 1. 3. 2, and the various baseload plant design condition considered are presented in Section 1. 3. 2. 1.

Results of various initial baseload channel calculations are presented in Tables 2-7 and 2-8. These channel calculations pertain to a MHD power cycle with combustion of air preheated in separately fired high temperature air preheaters. The thermal input with coal to the MHD combustor is 1000 MW and the air preheat temperatures considered are 2500°F and 3000°F respectively.

Figure 2-31 shows important design and performance data for the selected baseload channels for 2500°F and 3000°F preheat temperatures based upon the calculations performed.

Tables 2-9 and 2-10 show results from some of the baseload channel calculations considering oxygen enrichment of the combustion air together with more moderate preheat temperatures from 1100°F to 1400°F. Table 2-9 pertains to an oxygen enrichment of the air of 30% ($N_2/O_2 = 2.356$) and Table 2-10 to an oxygen enrichment of 34% ($N_2/O_2 = 1.953$). The flame temperatures achieved for these two degrees of oxygen enrichments are roughly the same as the flame temperatures attained with combustion of air preheated to 2500°F and 3000°F respectively.

The channel calculations conducted for 30% oxygen enrichment of the combustion air for 1450 MW coal thermal input resulted in the selection of the following values of inlet stagnation pressures and electrical load parameters for the three preheat temperatures considered:

Oxidizer Preheat Temperature	°F	1100	1200	1400
Inlet Stagnation Pressure	atm	5.0	5.0	6.0
Channel Electrical Load Parameters	α	0.77	0.78	0.75

For the case with 34% oxygen enrichment of the combustion air the selected values of inlet stagnation pressures and electrical load parameters for the same coal thermal input of 1450 MW and the same preheat temperatures of 1100, 1200 and 1400°F are listed below:

Oxidizer Preheat Temperature	°F	1100	1200	1400
Inlet Stagnation Pressure	atm	6	6	6.75
Channel Electrical Load Parameters	α	0.77	0.78	0.77

TABLE 2-7
BASELOAD CHANNELS AT PREHEAT TEMPERATURE 2500°F

		Magnetic Field Distribution: X									
		B		0		0.75		3		22	
				4		6		6		3.15	
Mass Flow Rate (kg/s)	353	353	353	353	353	353	353	353	353	353	353
Stagnation Enthalpy (cal/g)	222	222	222	222	222	222	222	222	222	222	222
Electrical Load Parameter	0.75	0.75	0.75	0.75	0.775	0.775	0.775	0.8	0.8	0.8	0.8
Inlet Mach #	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85
Stagnation Pressure (atm)	4.5	5.0	5.5	6.0	5.5	5.75	6.0	4.5	5.0	5.25	5.5
Max. Axial Field (kV/m)	4.5 ^(7m)	3.3 ^(9m)	2.5 ^(11m)	2.0 ^(8m)	1.9 ^(8m)	1.7 ^(7m)	1.6 ^(6m)	2.3 ^(8m)	1.8 ^(7m)	1.6 ^(6m)	1.4 ^(6m)
Max. Hall Parameter ($\omega\tau$)	5.5 ^(7m)	4.9 ^(9m)	4.4 ^(12.5)	3.4 ^(16m)	3.5 ^(15m)	3.1 ^(15m)	2.7 ^(14m)	4.7 ^(10.5)	3.7 ^(15m)	3.2 ^(14m)	2.8 ^(14m)
*Electric Power Output (MW)	256	264	280	289	283	(272)	(254)	260	274	(265)	(246)
Enthalpy Extraction %				20.5	20.1				19.4		
Isentropic Efficiency %				65.6	67.1				68.5		
Channel Length (m)				19.3	18.1	(> 22)	(> 22)		16.9	(> 22)	(> 22)
Compressor Power (MW)				78.2	73.7				68.9		
Net Power (MW)				204.8	203.5				199.6		
Heat Loss (MW)	15.1	19.5	26	39.0	36.3	(43.7)	(45.0)	21	33.5	(42.8)	(44.1)

*Based on Diffuser Recovery Factor 0.5

TABLE 2--8

BASELOAD CHANNELS AT PREHEAT TEMPERATURE 3000°F

	Magnetic Field Distribution												
	X 0 0 75 3 22 B 4 6 6 3 15												
Mass Flow Rate (kg/s)	353	353	353	353	353	353	353	353	353	353	353	353	353
Stagnation Enthalpy (cal/g)	290	290	290	290	290	290	290	290	290	290	290	290	290
Electrical Load Parameter	0 75	0 75	0 75	0 75	0 78	0 79	0 775	0 775	0 8	0 8	0 8	0 8	0 8
Inlet Mach #	0 85	0 85	0 85	0 85	0 85	0 85	0 85	0 85	0 85	0 85	0 85	0 85	0 85
Stagnation Pressure (atm)	6 0	6 5	7 0	7 5	6 5	6 5	6 5	7 5	6 0	6 25	6 5	7 0	7 5
Max Axial Field (kV/m)	4 1 ^(7 5)	3 3 ^(9 5)	2 5 ^(10 m)	2 0 ^(9 m)	2 0 ^(9 m)	1 7 ^(9 m)	2 2 ^(9 m)	1 4 ^(8 m)	1 9 ^(8 m)	1 6 ^(8 m)	1 5 ^(9 m)	1 2 ^(7 m)	1 1 ^(5 m)
Max. Hall Parameter ($\omega\tau$)	5 5 ^(7 5)	5 0 ^(9 5)	4 5 ^(12 m)	3 8 ^(16 m)	4 3 ^(13 m)	3 9 ^(16 m)	4 4 ^(12 m)	2 8 ^(15 m)	4 4 ^(13 m)	3 9 ^(15 m)	3 4 ^(15 m)	2 6 ^(15 m)	2 1 ^(14 m)
*Electric Power Output (MW)	310	320	336	342	330	333	327	(312)	320	329	(327)	(289)	(256)
Enthalpy Extraction %				22 6		22				21 8			
Isentropic Efficiency %				66 7		69 7				69 9			
Channel Length (m)				15 7		15 5		(> 22)		15 4	(> 22)	(> 22)	(> 22)
Compressor Power (MW)				90 4		82 5				80 4			
Net Power (MW)				244 9		244 2				243			
Heat Loss (MW)	19 5	24 5	30 9	39 4	32 6	37 8	31	(53 4)	31 5	37 4	(50 5)	(53 2)	(55 8)

	Magnetic Field Distribution												
	X 0 0 75 3 22 B 4 6 6 3 72												
Mass Flow Rate (kg/s)	353	353	353	353									
Stagnation Enthalpy (cal/g)	290	290	290	290									
Electrical Load Parameter	0 79	0 79	0 8	0 8									
Inlet Mach #	0 75	0 8	0 75	0 8									
Stagnation Pressure (atm)	6 5	6 5	6 25	6 25									
Max Axial Field (kV/m)	1 2 ^(8 m)	1 5 ^(9 m)	1 2 ^(8 m)	1 5 ^(9 m)									
Max Hall Parameter ($\omega\tau$)	3 2 ^(17 m)	4 0 ^(17 m)	3 3 ^(17 m)	4 0 ^(17 m)									
*Electric Power Output (MW)	(324)	337	320	334									
Enthalpy Extraction %		22 3		22 1									
Isentropic Efficiency %		70 5		70 9									
Channel Length (m)	(> 22)	17	(> 22)	17									
Compressor Power (MW)	82 5	82 5	80 4	80 4									
Net Power (MW)		248 2		248									
Heat Loss (MW)	(50 5)	40 8	(50)	40 5									

* Based on Diffuser Recovery Factor 0.5

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Figure 2-31

Baseload MHD Channels Design Data for 2500°F
 and 3000°F Preheat Temperatures

TABLE 2-9
BASELOAD CHANNELS WITH $N_2/O_2 = 2.356$

	Magnetic Field Distribution:														X	0	0.75	3	22
														B	4	6	6	3.15	
Coal Thermal Input (MW)	1330	1330	1330	1330	1330	1450	1450	1450	1450	1450	1450	1450	1450	1450	1450	1450	1450	1450	
Mass Flow Rate (Kg/s)	355.8	355.8	355.8	355.8	355.8	388.9	388.9	388.9	388.9	388.9	388.9	388.9	388.9	388.9	388.9	388.9	388.9	388.9	
Preheat Temperature (°F)	1100	1100	1100	1200	1400	1100	1100	1200	1200	1200	1200	1200	1400	1400	1400	1400	1400	1400	
Stagnation Enthalpy (Cal/g)	12	12	12	24	48	12	12	24	24	24	24	24	48	48	48	48	48	48	
Electrical Load Parameter	0.75	0.76	0.77	0.78	0.75	0.77	0.78	0.76	0.77	0.775	0.78	0.79	0.75	0.76	0.77	0.75	0.76	0.765	
Inlet Mach #	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	
Stagnation Pressure (Atm)	5	5	5	5	6	5	5	5	5	5	5	5	5.75	5.75	5.75	6	6	6	
Stagnation Temperature (°K)	2703	2703	2703	2718	2759	2703	2703	2718	2718	2718	2718	2718	2756	2756	2756	2759	2759	2759	
Max. Axial Field (kV/m)	2.3 ^(9m)	2.05 ^(8m)	1.83 ^(8m)	1.77 ^(8m)	1.89 ^(9m)	1.88 ^(8m)	1.68 ^(7m)	2.36 ^(9m)	2.07 ^(9m)	1.94 ^(8m)	1.83 ^(8m)	1.62 ^(7m)	2.23 ^(10m)	1.95 ^(9m)	1.72 ^(8m)	1.95 ^(9m)	1.7 ^(8m)	1.63 ^(8m)	
Max. Hall Parameter (ωτ)	12.5m ^(12.5m)	14.5m ^(14.5m)	16m ^(16m)	15.7m ^(15.7m)	16m ^(16m)	16m ^(16m)	15m ^(15m)	11.5m ^(11.5m)	13m ^(13m)	14m ^(14m)	15m ^(15m)	15m ^(15m)	12.8m ^(12.8m)	14.5m ^(14.5m)	16m ^(16m)	15.5m ^(15.5m)	17m ^(17m)	3.07 ^(16m)	
*Electric Power Output (MW)	252	256	258	264	289.5	284	286	282	286	288	289	291.5	312	313	316	317	319.4	320.5	
Enthalpy Extraction %	17.3	17.5	17.7	17.9	19.1	17.8	17.9	17.4	17.7	17.8	17.9	18.1	18.8	18.9	19.1	19.2	19.3	19.4	
Isentropic Efficiency %	64.8	65.9	66.3	67.5	65.9	66.8	67.3	65.9	66.9	67.4	67.6	68.2	66.6	66.8	67.4	66.1	66.6	66.8	
Channel Length (m)	12.5	14.5	16.5	15.7	16	16	20	11.5	13	14	15	18.5	12.8	14.5	17	15.5	18	21.5	
Compressor Power (MW)	66.2	66.2	66.2	66.2	75.1	72.4	72.4	72.4	72.4	72.4	72.4	72.4	79.8	79.8	79.8	82.1	82.1	82.1	
Net Power (MW)	185.8	189.8	191.8	197.8	214.4	211.6	213.6	209.6	213.6	215.6	216.6	219.1	232.2	233.2	236.2	234.9	237.3	238.4	
Heat Loss (MW)	27.5	31.1	34	34	37.5	35	42.3	27	30.5	32.1	34	40.7	32.3	36	41	38	43.8	50	
*Based on Diffuser Recovery Factor 0.5																			

TABLE 2-10
BASELOAD CHANNELS WITH $N_2/O_2 = 1.953$

	Magnetic Field Distribution:													X	0	0.75	3	22
	B	4	6	6	3.15													
Coal Thermal Input (MW)	1330	1330	1330	1330	1330	1450	1450	1450	1450	1450	1450	1450	1450	1450	1450	1450	1450	1450
Mass Flow Rate (Kg/s)	321.1	321.1	321.1	321.1	321.1	351	351	351	351	351	351	351	351	351	351	351	351	351
Preheat Temperature (°F)	1100	1100	1100	1200	1400	1100	1100	1200	1200	1200	1400	1400	1400	1400	1400	1400	1400	1400
Stagnation Enthalpy (Cal/g)	-7	-7	-7	3	24	-7	-7	3	3	3	24	24	24	24	24	24	24	24
Electrical Load Parameter	0.76	0.77	0.78	0.78	0.77	0.77	0.78	0.77	0.78	0.79	0.75	0.76	0.77	0.775	0.78	0.75	0.76	0.77
Inlet Mach #	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85	0.85
Stagnation Pressure (Atm)	6	6	6	6	6.75	6	6	6	6	6	6.75	6.75	6.75	6.75	6.75	7	7	7
Stagnation Temperature (°K)	2791	2791	2791	2802	2834	2791	2791	2802	2802	2802	2834	2834	2834	2834	2834	2837	2837	2837
Max. Axial Field (kV/m)	1.89 ^(9m)	1.84 ^(9m)	1.43 ^(8m)	1.54 ^(8m)	1.47 ^(9m)	1.7 ^(9m)	1.48 ^(8m)	1.86 ^(9m)	1.6 ^(9m)	1.39 ^(8m)	2.1 ^(10m)	1.79 ^(10m)	1.54 ^(9m)	1.43 ^(8m)	1.33 ^(8m)	1.84 ^(10m)	1.58 ^(9m)	1.38 ^(8m)
Max. Hall Parameter (φτ)	3.8 ^(13m)	3.5 ^(15.1m)	3.1 ^(16m)	3.5 ^(15.5m)	3.28 ^(16m)	3.65 ^(14.5m)	3.22 ^(16m)	3.9 ^(13m)	3.6 ^(14.5m)	3.2 ^(16m)	3.95 ^(12m)	3.75 ^(13.6m)	3.42 ^(16m)	3.17 ^(17m)	2.95 ^(16m)	3.7 ^(14m)	3.4 ^(16m)	2.9 ^(16m)
*Electric Power Output (MW)	267	270	272	275	293.8	296	298	298	301.3	304	314	318	321.3	322.7	318	318	321	321.1
Enthalpy Extraction %	18.7	18.9	19.1	19.1	20.	19.	19.1	18.9	19.1	19.3	19.5	19.8	20.	20.1	19.8	19.8	20	20
Isentropic Efficiency %	66.5	67.2	67.7	68.1	67.7	67.4	68.1	67.5	68.2	68.8	66.3	67.1	67.8	68.1	67.1	65.9	66.6	66.6
Channel Length (m)	13	15.1	20	15.5	17.5	14.5	17.8	13	14.7	19	12	13.6	16	18.4	23	14	16	22
Compressor Power (MW)	66.1	66.1	66.1	66.1	71.6	72.3	72.3	72.3	72.3	72.3	78.3	78.3	78.3	78.3	78.3	81	81	81
Net Power (MW)	200.9	203.9	205.9	208.9	222.2	223.7	226.7	225.7	229	231.7	235.7	239.7	243.	244.4	239.7	237	240	240.1
Heat Loss (MW)	32.6	37	46.6	38.4	45.1	37	44.5	34.2	37.4	47.4	34.4	38.3	43.9	49.1	59	39	44	57.3

*Based on Diffuser Recovery Factor 0.5

As indicated earlier our system analysis of "first" baseload MHD power plants designs have been limited. Therefore our calculated baseload channel performance data should not be considered optimum for the various potential baseload power cycles considered. Nevertheless they provide valuable information for guidance and comparison.

2.3 ETF CHANNEL PERFORMANCE

2.3.1 Introduction

In order to determine a channel design for the ETF, performance calculations were conducted for various combinations of mass flow rate, inlet Mach number, stagnation pressure, and electrical load parameter. The ranges of ETF power plant and channel parameters are described in Tables 2-11 and 2-12. Based on these prescribed conditions, many channel calculations were performed. Major decisions involved the selection of the channel inlet Mach number (i.e., subsonic or supersonic), the selection of the mass flow rate, and the selection of the stagnation pressure.

2.3.1.1 Effects of Inlet Mach Number

Qualitative comparisons between subsonic and supersonic channel operation were made in the previous subsection. These favored, in general, subsonic channel operation. Table 2-13 shows calculated operating characteristics of channel designs operating at subsonic and supersonic Mach numbers. For the supersonic cases, the magnetic field was restricted to 4 Tesla in order to keep the maximum Hall parameter and axial electrical field within known limits. It is seen that the major performance parameters (enthalpy extraction, isentropic efficiency) are lower than for subsonic channels.

Another important consideration favoring subsonic operation is part-load performance. As is shown in Section 2.3.2, part-load operation is achieved at reduced mass flow, i.e., at reduced burner pressure. For supersonic operation, the burner pressure reduction is directly proportional to the mass flow reduction, while for subsonic operation the pressure reduction is less. Hence, higher isentropic efficiencies can be achieved at larger mass flow reductions in subsonic operation than in supersonic operation. On the basis of these considerations and the qualitative comparisons made in Section 2.2, subsonic operation was chosen for the ETF channel.

2.3.1.2 Effects of Mass Flow Rate

Tables 2-14 and 2-15 show comparative initial calculated ETF channel performance for a range of mass flows at preheat temperatures of 2500° and 3000°F, respectively. The superscripted numbers for the maximum axial field and Hall parameter signify the axial location at which these values occur in the channel. The assumed magnetic field distribution in

TABLE 2-11
INITIAL ETF MHD PLANT DESIGN PARAMETERS

<u>Coal</u>		<u>ETF</u>
Type		Montana Sub-bituminous
Moisture Content		22.7% raw/5% Dry
Thermal Input to MHD Burner	MW	112.225
<u>Oxidizer</u>		
Type	°F	Air
Preheat Temperature		2500/3000
<u>Prime MHD-Cycle</u>		
MHD Channel Thermal Input	MW	150-330
Tital Mass Flow Rate	kg/s	40-80
A/F Equivalence Ratio		0.95
Potassium Seeding		1% K
Magnetic Field		6T (peak)/5T (average)

TABLE 2-12
ETF MHD CHANNEL DESIGN PARAMETERS

Preheat Temp.	^o F	2500	3000
Pressure	atm	3.5 - 4.7	4.5 - 7.0
El. Load Parameter		0.65 - 0.70	
Mach # - Subsonic		0.75 - 0.90	
*Supersonic		1.10 - 1.40	
*Magnetic Field 4 Tesla			

TABLE 2-13

SUPERSONIC VS SUBSONIC ETF CHANNELS

	Peak Magnetic Field : 4T (Super sonic)					Peak Magnetic : 6T Field (Subsonic)	
	50.9	50.9	50.9	50.9	50.9	50.9	50.9
Mass Flow Rate (kg/s)							
Preheat Temperature (3000°F)	260	260	260	260	260	260	260
Electrical Load Parameter	0.675	0.675	0.675	0.675	0.675	0.675	0.675
Inlet Mach Number	1.2	1.3	1.4	1.3	1.4	0.85	0.85
Stagnation Pressure (atm)	5.0	5.0	5.0	5.38	5.38	5.38	5.0
Maximum Axial Field (kV/m)	2.9 ^(9 m)	4.1 ^(7.5 m)	5.6 ^(7 m)	3.4 ^(9 m)	4.7 ^(8 m)	2.3 ^(3 m)	2.8 ^(4 m)
Maximum Hall Parameter ($\omega\tau$)	4.7 ^(9 m)	5.4 ^(8 m)	5.9 ^(7 m)	5.0 ^(9.5 m)	5.5 ^(8.5 m)	3.1 ^(7 m)	4.1 ^(7 m)
Electric Power Output (MW)	27.1	27.1	26.2	27.1	26.2	29.8	30.8
Enthalphy Extraction %	12.8	12.8	12.4	12.8	12.4	14.1	14.6
Isentropic Efficiency%	45.9	45.9	44.3	44	42.6	48.5	52
Channel Length (m)	9.5	8	7	9.5	8.5	10	7

TABLE 2-14

MASS FLOW VARIATIONS FOR ETF CHANNELS AT PREHEAT TEMPERATURE 2500°F

Magnetic Field Distribution: As in Fig. 2-1							
Mass Flow Rate (kg/s)	45	52	60	65	70	75	80
Stagnation Enthalpy (cal/g)	173	179	182	185	187	189	190
Electrical Load Parameter	0.675	0.675	0.675	0.675	0.675	0.675	0.675
Inlet Mach Number	0.85	0.85	0.85	0.85	0.85	0.85	0.85
Stagnation Pressure (atm)	3.7	3.85	3.95	4.0	4.1	4.2	4.3
Maximum Axial Field (kV/m)	3.0 ^(3 m)	3.1 ^(3 m)	3.1 ^(3 m)	3.1 ^(3 m)	3.1 ^(3 m)	3.0 ^(3 m)	3.0 ^(3 m)
Maximum Hall Parameter ($\omega\tau$)	3.8 ^(7 m)	3.8 ^(7 m)	3.8 ^(7 m)	3.9 ^(7 m)	3.8 ^(7 m)	3.7 ^(7 m)	3.6 ^(7 m)
*Electric Power Output (MW)	17.5	23.1	29	32	35.5	38.5	41
*Enthalpy Extraction %	10.5	11.6	12.5	12.8	13.0	13.2	13.3
*Isentropic Efficiency %	43	47.6	50.7	51.1	51.5	51.6	51.7
Channel Length (m)	7.6	7.5	7.5	7.5	7.6	8	8.6

TABLE 2-15

MASS FLOW VARIATIONS FOR ETF CHANNELS AT PREHEAT TEMPERATURE 3000°F

Magnetic Field Distribution: as in Figure 2-1							
Mass Flow Rate (kg/s)	45	52	60	65	70	75	80
Stagnation Enthalpy (cal/g)	247	253	257	260	262	264	265
Electrical Load Parameter	0.675	0.675	0.675	0.675	0.675	0.675	0.675
Inlet Mach Number	0.85	0.85	0.85	0.85	0.85	0.85	0.85
Stagnation Pressure (atm)	4.8	5.0	5.3	5.4	5.5	5.6	5.7
Maximum Axial Field (kV/m)	2.6 ^(4 m)	2.6 ^(4 m)	2.5 ^(4 m)	2.5 ^(4 m)	2.5 ^(4 m)	2.5 ^(4 m)	2.5 ^(4 m)
Maximum Hall Parameter ($\omega\tau$)	3.8 ^(7 m)	3.8 ^(7 m)	3.5 ^(8 m)	3.5 ^(8 m)	3.5 ^(8 m)	3.4 ^(8 m)	3.3 ^(8 m)
*Electric Power Output (MW)	24.0	30.0	37	41	44.3	48.0	52.1
*Enthalpy Extraction %	12.9	13.9	14.9	15.2	15.2	15.3	15.6
*Isentropic Efficiency %	47.4	49.8	51.4	52.0	51.5	9.4	52
Channel Length (m)	7.6	7.8	8.7	9	9.1	15.8	9.8

these comparative channel calculations was shown in Figure 2-1. It was a simplified representation of the actual design field which was developed later.

The calculated power output, enthalpy extraction and isentropic efficiency are listed in Tables 2-14 and 2-15. These calculated results were plotted in Figures 2-2 and 2-3 as a function of mass flow rate. All three parameters (power output, enthalpy extraction and isentropic efficiency) increase with mass flow rate, but the incremental gain becomes markedly less beyond a mass flow rate of ~ 65 kg/s.

As earlier discussed, the larger the MHD channel is the better its performance will be because of reduced surface to volume ratio. However, beyond a certain size, at which the heat and friction losses are a small fraction of the power output, a further increase of the channel size will not improve its performance significantly. The enthalpy extraction approaches a constant thermodynamic limit determined mainly by channel pressure ratio and loading parameter. Since total plant cost increases with total mass flow rate, one seeks the minimum flow rate that satisfies the selected channel performance goals with inclusion of a certain margin for achieving those goals. A mass flow of 65 kg/sec was judged to fulfill the channel performance goals with the desired margin. Hence, a mass flow rate of 65 kg/sec was selected for the ETF reference channel.

2.3.1.3 Burner Pressure

In general, as the burner pressure is increased, higher power output and higher enthalpy extraction can be obtained up to a certain pressure. Offsetting these gains is the increased compressor power required. Also, the channel is longer and requires a longer and more expensive magnet. The most important factor was judged to be the increased compressor power. Figure 2-32 shows a plot of net MHD power (channel electrical power less compressor power) vs burner pressure, with load parameter and preheat temperature as parameters. The load parameter was here selected as 0.675, prior to selecting 0.69 as the final design value for the ETF channel. Figure 2-32 shows that the optimum burner pressures span a range around 5.5 atm for the case of 3000°F preheat, and around 4 atm for the 2500°F preheat. Tables 2-16 and 2-17 show operating parameters for possible ETF channels operating at stagnation pressures within these ranges. For 2500°F preheat

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Figure 2-32 Burner Pressures

TABLE 2-16

ETF CHANNELS FOR 65 kg/s PREHEAT TEMPERATURE 2500°F

Mass Flow Rate (kg/s)	65	65	65	65	65
Stagnation Pressure (atm)	3.8	3.9	4.0	4.1	4.2
Stagnation Enthalpy (cal/g)	185	185	185	185	185
Electrical Load Parameter	0.675	0.675	0.675	0.675	0.675
Inlet Mach Number	0.85	0.85	0.85	0.85	0.85
Maximum Axial Field (kV/m)	3.6 ^(4 m)	3.3 ^(4 m)	3.1 ^(3 m)	3.0 ^(3 m)	2.8 ^(3 m)
Maximum Hall Parameter ($\omega\tau$)	4.4 ^(6 m)	4.2 ^(7 m)	3.9 ^(7 m)	3.6 ^(7 m)	3.4 ^(7 m)
Electric Power Output (MW)	30.6	31.6	32	32.2	32.5
Enthalpy Extraction %	12.3	12.6	12.8	12.9	13.0
Isentropic Efficiency %	50.8	51.3	51.1	50.6	50.2
Channel Length (m)	6	7	7.5	8.5	0
Compressor Power (MW)			11.6	11.8	12.1
Net Power (MW)			20.4	20.4	20.4

TABLE 2-17

ETF CHANNELS FOR 65 kg/s PREHEAT TEMPERATURE 3000°F

Magnetic Field Distribution: As in Figure 2-1

Mass Flow Rate (kg/s)	65	65	65	65	65	65	65
Stagnation Pressure (atm)	5.2	5.3	5.4	5.5	5.6	6.5	7.0
Stagnation Enthalpy (cal/g)	260	260	260	260	260	260	260
Electrical Load Parameter	0.675	0.675	0.675	0.675	0.675	0.675	0.675
Inlet Mach Number	0.85	0.85	0.85	0.85	0.85	0.85	0.86
Maximum Axial Field (kV/m)	2.8 ^(4 m)	2.7 ^(4 m)	2.5 ^(4 m)	2.4 ^(4 m)	2.3 ^(4 m)	1.7 ^(2 m)	1.5 ^(2 m)
Maximum Hall Parameter ($\omega\tau$)	4.0 ^(8 m)	3.7 ^(7 m)	3.5 ^(8 m)	3.3 ^(8 m)	3.1 ^(7 m)	2.0 ^(6 m)	1.7 ^(6 m)
Electric Power Output (MW)	39.8	40.6	41	41	39.5	(28)	(23.5)
Enthalpy Extraction %	14.7	15	15.2	15.2	14.7	(10.4)	(8.7)
Isentropic Efficiency %	51.6	52	52	51.5	49.3	(32.4)	(26.2)
Channel Length (m)	8	8	9	12	12	(> 15)	(> 15)
Compressor Power (MW)	14.1	14.3	14.5				
Net Power (MW)	25.7	26.3	26.5				

(Table 2-16), the stagnation pressure of 4 atm was chosen, and for 3000°F preheat (Table 2-17) a stagnation pressure of 5.4 atm was chosen.

2.3.2 Design Load Performance

ETF channel performance characteristics at full load are given in Table 2-18. Detailed design data for the two ETF channels in the Faraday operating mode are plotted in Figures 2-33 through 2-37. The magnetic field profile used in these channel design calculations is shown in Figure 2-35. It is the actual design field developed in the magnet design, which has a higher average value than the simplified field distribution used in earlier channel calculations. This higher magnetic field results in somewhat higher power output and hence higher enthalpy extraction and efficiency than the earlier results. Operating characteristics for 2-terminal diagonally connected channels are shown in Figures 2-38 and 2-39 for the original simplified magnetic field distribution.

The principal figures of merit for the two channels are summarized below:

2500°F Preheat	35.5 MW power output 14.2% enthalpy extraction 57% isentropic efficiency
3000°F Preheat	45.6 MW power output 16.9% enthalpy extraction 57.8% isentropic efficiency

The channel designs have the following characteristics:

1. The L/D (channel length to width ratio) of all cases is around 10.
2. For the 3000°F preheat case, the maximum axial field is about 2.4 kV/m, while for the 2500°F preheat, it is about 2.5 kV/m. The maximum Hall parameter $\omega\tau$ is about 4.4 for both cases.
3. The pressure recovery of the diffuser is assumed to be 46%, corresponding to the maximum achieved experimentally to date.

2.3.3 Part Load Performance

Part load performance of the MHD generator is accomplished by reducing mass flow, and hence thermal input to the generator. During part loading, it is important that the net enthalpy extraction of the channel, or MHD power minus required compressor power, is maintained as high as

TABLE 2-18

CHARACTERISTICS OF ETF REFERENCE CHANNELS
AT THE FULL LOAD OPERATING POINTS

Air Preheat	2500°F	3000°F
Fuel	Montana Sub-Bitum	Montana Sub-Bitum
Thermal Input	250 MW	270 MW
Mass Flow (m)	65 kg/s	65 kg/s
Inlet Stagnation Pressure (P_{oi})	4.0 atm	5.4 atm
Inlet Stagnation Temp (T_{oi})	2675°K	2794°K
Inlet Mach No. (M_i)	0.85	0.85
Loading Parameter (Faraday)	0.69	0.69
Channel Design Mode	Constant Velocity	Constant Velocity
Power Output	35.5 MW	45.6 MW
Enthalpy Extraction	14.2%	16.9%
Isentropic Efficiency	57.0%	57.8%
Exit Stagnation Pressure (p_{of})	1.2 atm	1.2 atm
Exit Mach No.	0.90	0.92
Diffuser Efficiency	46%	46%
Length	9 m	9 m
Inlet Cross Section	0.48 x 0.48 m	0.42 x 0.42 m
Exit Cross Section	0.90 x 0.90 m	0.90 x 0.90 m
Max. Transverse Current Density(J_y)	0.65 A/cm ²	1.1 A/cm ²
Max. Hall Field (E_x)	2.5 kV/m	2.4 kV/m
Max. Hall Parameter ($\omega\tau$)	4.4	4.4
Average Power Density	8.4 MW/m ³	11.6 MW/m ³

Figure 2-33 ETF Channel Loft

3000 °F PREHEAT

Figure 2-34 Channel Operating Characteristics

Figure 2-35 Channel Operating Characteristics

2500 °F PREHEAT

Figure 2-36 Channel Operating Characteristics

Figure 2-37 Channel Operating Characteristics

Figure 2-38 Channel Operating Characteristics

Figure 2-39 Channel Operating Characteristics

possible in order to be able to maintain optimum MHD efficiency (η_{MHD}) and hence optimum plant efficiency. It is also important that the axial electric field and the Hall parameter remain reasonable

Part load performance calculations provide information on the voltage and current loadings on the channel, burner pressure variations and associated compressor operating conditions, as well as electrical performance data. It is noted that little information has so far been developed on part load operation of MHD channels. Although the part load channel performance analyses conducted here are preliminary, they provide valuable information on channel part load operating characteristics. However, much further design work is required in this important area to establish the optimum channel and plant operating conditions at part load.

Part load performance of the subsonic Faraday channels was studied for both the 2500°F and 3000°F preheat cases. The part load calculations are summarized in Tables 2-19 and 2-20. During part load operation, the geometry of the channel remains fixed, having been determined by the design calculations at nominal (full) load.

The part load calculations were made using the PG79 computer program discussed earlier. The diffuser pressure recovery factor was assumed to remain constant at 0.46 for all mass flow conditions. The channel exit pressure was selected to obtain a diffuser exit pressure of 1 atm. In these preliminary analyses, gas stoichiometry, seed fraction and the channel electrical load factor were fixed at their nominal values. It is recognized that by varying these parameters, the generator performance could be optimized. Such optimization require further design analyses.

Calculations made with a quasi-one-dimensional computer program confirm that adjustment of the electrical load factor during part-loading can increase channel performance. However, because of the operational nature of the PG79 program, this improvement in performance has yet to be quantified. The performance curves presented below may be therefore considered to be on the conservative side.

The variation in the burner pressure as the mass flow is reduced is shown in Figures 2-40 and 2-41 for the two preheat cases. The reduction in burner pressure, which accompanies part-loading, results in a lower

TABLE 2-19
OFF-DESIGN CHANNEL OPERATING CHARACTERISTICS
WITH 3000°F PREHEAT

Fraction	1.00	0.90	0.70	0.50	0.40
$\dot{m}$ (kg/s)	65	58.5	45.5	32.5	26.0
$p_{o_{inlet}}$ (atm)	5.4	4.99	4.19	3.40	2.98
Power (MW)	45.6	40.7	29.8	19.0	13.9
Comp. Power (MW)	13.5	11.5	8.4	5.7	4.65
Net Power (MW)	32.1	29.2	21.4	13.3	9.25
η_{enth} (%)	16.9	16.7	15.8	14.1	12.8
η_{MHD} (%)	11.9	12.0	11.3	9.8	8.6
η_{isen} (%)	57.8	59.9	62.8	65.2	66.1
$M_{c_{inlet}}$	0.85	0.78	0.66	0.54	0.47
$M_{c_{exit}}$	0.92	0.82	0.64	0.47	0.39
$p_{o_{exit}}$ (atm)	1.20	1.16	1.09	1.05	1.03
$J_y(4\text{ m})$ (A/cm ²)	0.52	0.51	0.44	0.35	0.28
$E_y(4\text{ m})$ (kV/m)	3.78	3.57	3.08	2.52	2.21
$T_{o_{burner}}$ (°K)	2794	2789	2778	2764	2756
$E_{x_{max}}$ (kV/m)	2.69	2.31	1.91	1.58	1.41
$\omega\tau_{max}$	4.4	4.3	4.2	4.2	4.2
Load Factor	0.69	0.69	0.69	0.69	0.69

TABLE 2-20
OFF-DESIGN CHANNEL OPERATING CHARACTERISTICS
WITH 2500°F PREHEAT

Fraction	1. 00	1. 90	1. 70	0. 50	0. 40
$\dot{m}$ (kg/s)	65	58. 5	45. 5	32. 5	26. 0
$p_{o_{inlet}}$ (atm)	4. 0	3. 70	3. 14	2. 60	2. 31
Power (MW)	35. 5	31. 5	22. 9	14. 3	10. 3
Comp. Power (MW)	12. 0	10. 5	7. 5	3. 9	2. 7
New Power (MW)	23. 5	21. 0	15. 4	10. 4	9. 6
η_{enth} (%)	14. 2	14. 1	13. 1	11. 5	10. 4
η_{MHD} (%)	9. 3	9. 4	8. 8	8. 4	7. 6
η_{isen} (%)	56. 8	59. 4	62. 7	65. 2	66. 1
$M_{c_{inlet}}$	0. 85	0. 78	0. 65	0. 52	0. 45
$M_{c_{exit}}$	0. 90	0. 78	0. 60	0. 44	0. 36
$p_{o_{exit}}$ (atm)	1. 20	1. 15	1. 09	1. 04	1. 03
J_y (4 m) (A/cm ²)	0. 37	0. 36	0. 32	0. 25	0. 21
E_y (4 m) (kV/m)	3. 78	3. 51	2. 93	2. 32	1. 99
$T_{o_{burner}}$ (°K)	2675	2670	2661	2650	2644
$E_{x_{max}}$ (kV/m)	2. 51	2. 20	1. 79	1. 32	1. 25
$\omega\tau_{max}$	4. 4	4. 3	4. 2	4. 2	4. 2
Load Factor	0. 69	0. 69	0. 69	0. 69	0. 69

Figure 2-40 Burner Pressure as a Function of Mass Flow

Figure 2-41 Burner Pressure as a Function of Mass Flow

pressure ratio for the channel. It is important to note that the reduction in burner pressure is less than the reduction in mass flow rate. For supersonic operation the burner pressures would be reduced in direct proportion to the reduction in mass flow rate. Because of this subsonic operation results in less reduction of the enthalpy extraction at part load compared to supersonic operation. Since the isentropic efficiency can be maintained for subsonic operation, it is also likely that subsonic operation results in higher net enthalpy extraction or MHD efficiency (η_{MHD}) and hence higher plant efficiency at part load than supersonic operation. A more detailed analysis of subsonic and supersonic channel operation should be carried out for a quantitative comparison of part load performance between these two different modes of channel operation.

The enthalpy extraction based on the total electrical power produced by the channel, shows a moderate decrease at part load for both preheat cases (Figures 2-42 and 2-43). The variation of the net enthalpy extraction or MHD efficiency, the most important measure of performance, is illustrated in Figures 2-44 and 2-45 for variations in mass flow rate. The variations in net MHD power (MHD generator power minus compressor power) are also shown in the figures. Note that the net enthalpy extraction remains almost constant down to about 70% mass flow rate. For the 3000°F preheat case, the net enthalpy extraction decreases rather rapidly below 70%, whereas at 2500°F preheat, the decrease is not so pronounced. This behavior is due mainly to the operating characteristics of the particular compressor selected. The pressure characteristics established for part load operation at 70% or less of mass flow rate at 3000°F preheat results in significant reductions of the efficiency of this compressor. However, for similar part load operating conditions at 2500°F preheat the compressor efficiency remains high. This points out the importance of establishing part load operating pressures for the compressor before its final selection. The compressor was originally selected before the channel part load operating pressures were determined. Therefore, it is recommended that the final selection of the compressor be based both on normal load and part load operating pressure.

Figure 2-42 Enthalpy Extraction and MHD Power as Functions of Mass Flow

Figure 2-43 Enthalpy Extraction and MHD Power as Functions of Mass Flow

Figure 2-44 η_{MHD} and Net Power as Functions of Mass Flow

Figure 2-45 η_{MHD} and Net Power as Functions of Mass Flow

The part load dependence of the Faraday current and voltage near the mid-channel location ($x = 4 \text{ m}$) is shown in Figures 2-46 and 2-47. The curves in these figures are typical of the voltage and current characteristics at other points in the channel. They also define the part load operating conditions for the inverter system. Note from Tables 2-19 and 2-20 that the maximum values of the Hall parameter $\omega\tau$ and the axial electric field E_X remain reasonable.

The part load performance curves shown in Figures 2-40 through 2-45 for a Faraday-loaded channel can also at present be considered representative for a multiterminal diagonal channel.

Figure 2-46 Faraday Current and Voltage as Functions of Mass Flow Near Mid-Channel

Figure 2-47 Faraday Current and Voltage as Functions of Mass Flow Near Mid-Channel

2.4 MECHANICAL DESIGN AND CONSTRUCTION

2.4.1 Overall Description

The channel cross section is shown in Figure 2-48. The channel consists of a four-wall assembly. Each wall consists of the gas-side surface elements mounted on an electrically insulating board. The boards are the major structural elements of the channel, and the pressure vessel is the box formed by the assembly of the four walls.

Figure 2-49 is a photograph of a section of the AERL Mk VI-C channel, which is constructed as described above, and shows clearly the construction technique used. Figure 2-50 is a photograph of a complete channel, showing the four wall construction and the assembly sequence.

2.4.2 Structure

The primary structure of the MHD channel is a continuous filament glass cloth reinforced laminated board. Resins are either epoxy (G-10, G-11) or silicone (G-7). The latter has greater residual strength at temperatures above 300°F. G-11 has the best flexural strength and highest modulus of elasticity in the temperature range of interest, 120 - 250°F. The epoxy bonded material has in Avco's experience proven to have superior resin impregnation and quality control. G-11 is the tentative selection for the ETF channel.

Figure 2-51 shows pressure distribution in the channel vs axial distance downstream from the beginning of the electrodes. Subsonic values were computed for operation under power generation conditions (see Section 2.3). Supersonic values were computed for zero magnetic interaction at design mass flow rate and channel inlet conditions. Maximum positive pressure loads on the channel occur at the inlet for subsonic operation. Maximum negative pressure load occurs at the exit for supersonic flow. G-11 wall thickness has been tentatively set at 2-1/2 inches.

The channel walls are edge bolted and RTV sealed with RTV silicon rubber to form a square cross section box. RTV sealing has been very successful at Avco. It is much easier to control than "O" ring seals which have severe groove leakage and sealing problems in laminated fiber reinforced plastic (frp's).

Figure 2-48 Installation Layout and Cross Sections, ETF MHD Channel

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Figure 2-49 Avco Mk VI-C Channel Construction

Figure 2-50 Mk VI Channel Assembly

Figure 2-51 Pressure Distribution, ETF MHD Channel

The channel is divided into 3 axial lengths. This is done primarily to limit frp board lengths to about 10 ft to simplify procurement. It reduces machining, assembly, and pressure checkout cost by making the individual panel assemblies more manageable in size. The channel modules are joined by bolted flanges. The flanges are bolted and epoxy bonded into external grooves in the frp walls. Ends of the frp walls are simple bolt joints sealed with RTV.

Channel supports and lifting points are located at the flanges. Supports are electrically insulated from the roller assemblies which bear on the channel magnet rails. A separate strongback will be provided for transportation and hoisting of the complete channel assembly when it is not resting on the diffuser or channel magnet rails.

2.4.3 Electrodes

An important tradeoff made in mechanical design of the channel involves the electrode pitch. Pitch, p , is defined here as the axial distance from a point on one electrode to the corresponding point on a neighboring electrode. A fine pitch minimizes the interelectrode voltage difference which is the product of the pitch and the local hall electric field, E_x . On the other hand, a fine pitch increases the cooling pressure drop, which can be shown to vary as the reciprocal of the square of the pitch, and also increase mechanical complexity, and the complexity of the cooling and wiring circuits. (The minimum cooling passage dimension occurs in the side bars which are inclined at an average angle of 45° to the axis.)

The conceptual design effort is evaluating the consequences of a pitch which varies from 10 mm (0.39") to 15.9 mm (5/8"). The larger pitch is that of the current Avco Mk VI generator. The interelectrode gap is on the order of 1.6 mm (1/16"). Emphasis is on getting as small a pitch as is feasible in order to minimize both electrical stress (voltage) between electrodes, and power available to a fault (short or arc between electrodes).

Electrodes will be 1" thick in the direction normal to the insulator board. The electrodes will be tentatively segmented transversely as shown in Figure 2-52. The longest electrode will not exceed 12". This is only 2" longer than those with which Avco has experimental experience.

Figure 2-53 is a photograph of recent Avco electrodes which are being evaluated for CDIF use. The cooling passages are drilled. Higher

Figure 2-52 Transverse Segmentation, Anode and Cathode Walls

Figure 2-53 Typical Electrodes

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Figure 2-52 Transverse Segmentation, Anode and Cathode Walls

L/D (longer) electrodes can be cut from extrusions. The longitudinal passage ends are plugged, and the cooling tubes brazed in. Material is OFHC copper. Protective caps of platinum or palladium are provided to protect upstream edges of anodic surfaces against oxygen ion attack. Anode caps wrap around the leading edge. Cathode caps need protect only from the side facing the neighboring electrode. The full length groove downstream of the protective cap is filled with castable zirconia to retain a protective slag layer.

Figure 2-54 shows two reference sidewall designs for the CDIF. Number 1 shows an 8.9 mm (0.35") pitch anode and cathode with side bars which span two electrodes. Number 2 shows a similar anode with a peg sidewall and a nonslagging, 17.8 mm (0.70") pitch cathode.

Sidewalls are segmented to minimize current draw between the opposite electrodes.

The ~ 1.69 mm (1/16") gap between electrodes is filled with a boron nitride insulator. The backside of the electrode facing the frp wall is insulated with RTV and possibly a mica sheet.

Electrode walls constructed as described above and as shown in Reference Design No. 2 of Figure 2-54 have operated successfully for 250 hours, under power generating conditions (current densities, axial electric fields, power density and heat flux) typical of those predicted for the ETF channel.

2.4.4 Cooling

The MHD channel is cooled by boiler feedwater. Two schemes were evaluated. One had the generator in a parallel water circuit receiving and returning cooling water to the deareator for control of copper entrainment. Pressure was maintained by a circulating pump, high enough to overcome loop pressure drop and to suppress boiling in the exit stream. Water conditions were as follows for a loop water flow rate of twice the boiler feedwater rate:

	<u>Inlet</u>	<u>Outlet</u>
p, psig	150	75
T, °F	227	280

The maximum water temperature would have caused problems with insulation materials and hoses currently used in MHD channel construction, plumbing,

Figure 2-54 Two Reference Electrode Designs

and wiring. (Note that metal reinforced hoses are not compatible with electrode-manifold voltage differences which will be developed.)

Manifolds and electrodes are connected by fabric reinforced rubber hose secured by clamps. Tube ends bulge to retain hose clamps. This scheme has been proven by Avco to be reliable both with respect to leakage, and to voltage standoff between electrodes and manifolds. Other manifolding schemes have been proposed, but would require development. Manifolds can be graded with respect to ground potential by lengthwise segmentation.

In order to retain materials and configurations which have proven to be successful, it was decided to revert to the other scheme which placed the channel in series with the boiler feedwater circuit. Condensate at 118 - 120°F is boosted through the channel at a pressure (similar to above) sufficient to overcome pressure drops and suppress boiling. The ~100°F temperature rise results in a maximum water temperature of 220°F which is within current practical materials limitations.

Boiler feed water is deaerated then boosted to 1500 psig for transmission on its way to the boiler through the economizer, burner, and diffuser.

Figure 2-55 shows the estimated heat flux distribution to the channel walls. The required water distribution can be achieved at a common supply pressure by orificing the hose nipples on the supply manifolds. This has the added advantage of stabilizing parallel flow paths against local self-aggravated flow starvation caused by hot spots (with consequent vapor generation). Similar practice has been used to stabilize flow through boiler water walls. Preliminary calculations indicate that manifolding is possible which feeds sets of 50 electrodes in parallel, connecting this set in series with 2 more sets of 50 parallel electrodes, then returning the cooling water to the deareator. Thus, 150 electrodes can be fed in one pass by means of this series-parallel arrangement. Figure 2-56 shows a current version of this grouping. Cooling manifolds will tentatively be arranged outside of the wire bundles which are described below.

2.4.5 Cabling

Figure 2-57 shows a typical current distribution for design point operation, and for the lower bound of the electrode pitch range. Also shown is the hall voltage relative to ground potential. Max current is 26 amps and

Figure 2-55 Typical Channel Heat Flux Distribution

Figure 2-56 Electrode Grouping and Manifolding for Cooling

Figure 2-57 Current and Hall Voltage Distribution, ETF Channel

max voltage to ground at the front electrode is -18.3 kV. Figure 2-58 shows a contemplated grouping of electrode segments to form 16 bundles of 150 wires each.

Wire bundle specifications are tentatively as follows, noting that amparage requirement is reduced by the transverse electrode sementation:

Current: 26 A/wire

Wire Temperature: 200°C

Ambient Temperature: 100°C

Max Voltage Difference within Bundle: 4000 V

An ITT 12 AWG wire with 0.156" OD, teflon insulated, and derated to 20 A at 150°C ambient with 19 conductors (tightly bundled) appears feasible.

Wires can be graded within bundles to reduce the maximum wire-to-wire voltage difference to a much lower value than 4000 V wire-to-wire.

Theoretical diameter for a tightly packed 150 wire cable is given by

$$\left(\frac{D}{d}\right)_{th} = \sqrt{1 + \frac{4}{3} (N - 1)}$$

where

D = bundle diameter

d = wire diameter

N = number of wires in bundle

For

$$N = 150, (D/d)_{th} = 14.13$$

$$d = 0.156''$$

$$D = 2.2''$$

In order to achieve the 20 A rating with a 150 wire bundle, we allow the wires to be loosely tied for heat dissipation, at say, 40% of theoretical packing density. Then

$$\frac{D}{d} = \left(\frac{D}{d}\right)_{th} / \sqrt{0.4} = 22.3$$

$$D = 3.5''$$

It appears feasible to get 32 such bundles out of the magnet bore, using both ends as shown in Figure 2-57.

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Figure 2-58 Electrode Grouping for Cabling

2.4.6 Installation

Figure 2-48 shows a conceptual layout of the MHD channel installed in the magnet bore. Internal channel dimensions were calculated as described in Section 2.3.2. Components shown are the burner extension (take-off duct), nozzle, MHD channel, and the inlet end of the diffuser. The channel rides in and out of the tapered magnet bore on 2 horizontal rails.

Assembly proceeds as follows: The diffuser rails are raised by a hydraulic lift to the level of the channel/magnet rails. The channel/nozzle assembly is placed on the diffuser rails by an overhead crane. The channel assembly is rolled far enough into the magnet to place the rearmost support rollers on the rear end of the channel/magnet rails. The channel/diffuser flange remains accessible outside the magnet.

The diffuser rails are lowered to their down position. The overhead crane places the diffuser on the diffuser rails. The channel/diffuser flange is bolted together, and the gas seal cover is fastened in place. Channel and diffuser are rolled together into their forward position. The connection between the nozzle and the burner extension piece is made inside the forward end of the magnet bore with the help of extension wrenches and capscrews. External wiring and cooling connections are made.

Either the transition section or the alternate exhaust quencher elbow are placed in position by overhead crane. Expansion joints are installed between the diffuser, alternate exhaust, or transition section, and boiler as applicable. Cooling (feedwater) connections are made.

Disassembly for access to the MHD channel proceeds in the reverse order, starting with removal of the alternate exhaust quencher, or transition section as applicable.

2.5 CHANNEL INSTRUMENTATION AND CONTROLS

2.5.1 Introduction

The channel instrumentation must detect and measure the operating characteristics to permit critical evaluation of performance, and also, the instrumentation must detect normal and abnormal operating characteristics and supply this information to a control system for use in a predetermined manner. The control system must assure reliable operation during steady state operation, operate the channel at optimum efficiency under all load conditions, and protect equipment and personnel in the event of channel failure.

2.5.2 Instrumentation

Measurements are needed to determine channel power output, channel efficiency, inter-element voltages and currents, static pressure and heat transfer. Table 2-21 is a list of channel instrumentation. Three main categories are included: electrical, gas static pressure and heat transfer.

Typical electrical measurements are shown in Figure 2-59. The list of instrumentation in Table 2-21 assumes 900 frames (i.e., a channel which is 9 m long, with electrodes at 1 cm pitch). Thus, 900 interelectrode voltage measurements are required, on both the top and bottom walls. Nine hundred transverse voltage and 900 transverse current measurements are also required. Each measurement requires a channel on the data acquisition system, and a transducer. In addition to the data acquisition system which will be used for data recording and analysis, on-line realtime monitoring of important variables will be provided by a visual readout of the appropriate data channel. Analog meters and chart recorders as listed in Table 2-21 are the least expensive visual display devices available.

If bar-wall construction is used for the sidewalls, as described in a previous section, monitoring of voltages between bar elements and between the sidewall elements and the electrodes is desirable, since these voltages can reach large values. Figure 2-59(b) shows a typical scheme, for a 3-element bar wall. Four measurements are required per frame (bar elements plus corresponding electrodes on one side only). Every fifth frame will be so instrumented for a total of 720 measurements for the channel.

TYPICAL SPLIT FRAME
CONTROLS

TYPICAL FRAME TO FRAME
CONTROLS

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Figure 2-59 Typical Electrical Measurements

TABLE 2-21

TYPICAL INSTRUMENTATION FOR ETF CHANNEL

<u>Function</u>	<u>Device</u>	<u>Number</u>	<u>Measuring</u>	<u>Range</u>
Generator Frame Current	Transducer	900	Current	0-50 Adc
	Meter	900	Current	0-50 Adc
	DAS	900	Current	0-50 Adc
Generator Frame Voltage	Transducer	900	Voltage	0-5 kVdc
	Meter	900	Voltage	0-5 kVdc
	DAS	900	Voltage	0-5 kVdc
Interelectrode Voltages	Transducer	1800	Voltage	0-100 Vdc
	Meter	1800	Voltage	0-100 Vdc
	DAS	1800	Voltage	0-100 Vdc
Side Wall Element Voltage	Transducer	720	Voltage	0-100 Vdc
	Meter	720	Voltage	0-100 Vdc
	DAS	720	Voltage	0-100 Vdc
Diagonal Load Voltage	Transducer	5	Voltage	0-50 kVdc
	Chart Recorder	5	Voltage	0-50 kVdc
	Meter	5	Voltage	0-50 kVdc
	DAS	5	Voltage	0-50 kVdc
Diagonal Load Current	Transducer	5	Current	0-5 kAdc
	Chart Recorder	5	Current	0-5 kAdc
	Meter	5	Current	0-5 kAdc
	DAS	5	Current	0-5 kAdc
Generator Power	Transducer		Power	0-100 MW
	Chart Recorder		Power	0-100 MW
	Meter		Power	0-100 MW
	DAS		Power	0-100 MW

TABLE 2-21

TYPICAL INSTRUMENTATION FOR ETF CHANNEL (Continued)

<u>Function</u>	<u>Device</u>	<u>Number</u>	<u>Measuring</u>	<u>Range</u>
Consolidation Current	Transducer	45	Current	0-50 Adc
	Chart Recorder	45	Current	0-50 Adc
	Meter	45	Current	0-50 Adc
	DAS	45	Current	0-50 Adc
Consolidation Power	Transducer	45	Power	0-2 MW
	Chart Recorder	45	Power	0-2 MW
	Meter	45	Power	0-2 MW
	DAS	45	Power	0-2 MW
Channel Static Pressure	Transducer	19	Pressure	0-100 psia
	Indicator	19	Pressure	0-100 psia
	DAS	19	Pressure	0-100 psia
Channel Cooling Water Heat Transfer	Flowmeter	24	Flow Rate	0-100 gpm
	DAS	24	Flow Rate	0-100 gpm
	Sensor	24	Temp.	0-300°F
	Chart Recorder	24	Temp.	0-300°F
	DAS	24	Temp.	0-300°F
	Transducer	24	Pressure	0-250 psig
	Gauge	24	Pressure	0-250 psig
	DAS	24	Pressure	0-250 psig
Safety Interlock	Flow Switch	24	Flow	0-100 gpm
	Temp. Switch	24	Temp.	
Channel Water	Indicator	1	Flow	0-1500 gpm
	DAS	1	Flow	0-1500 gpm
Cooling Water	Indicator	20	Flow	0-100 gpm
	DAS	20	Flow	0-100 gpm

WATER COOLING HEADERS

Channel	Transducer	5	Pressure	0-250 psig
	DAS	5	Pressure	0-250 psig
	Sensor	5	Temp.	0-300°F
	DAS	5	Temp.	0-300°F

A diagonal-loaded channel will require an axial voltage and current (load voltage and current) measurement for each section from which power is drawn. Five such measurements are shown since the maximum number of loads considered for a multiloaded diagonal channel is five.

The number of static pressure measurements required is that necessary to obtain a pressure distribution along the channel with enough resolution to locate possible shockwaves or other anomalous behavior. Pressure transducers located at 0.5 m intervals are judged adequate. Thus, 19 will be required for a 9 m long channel.

The number of measurements required to obtain channel heat transfer will depend on the number of cooling water circuits. As indicated in Section 2.4.4 it is expected that 150 elements can be cooled in one hydraulic pass. Thus, 6 circuits are required for each of the electrode walls, and a like number for the sidewalls for a total of 24 water circuits. To obtain heat transfer data, a differential temperature measurement (outlet temperature minus inlet temperature) and a water flow rate measurement is required for each circuit. Cooling water flow switches and temperature switches are used as safety interlocks.

Additional instrumentation for measurement of vibration, thermal expansion, etc. may also be required.

2.5.3 Controls

Major inputs to the channel such as mass flow rate, gas stoichiometry, seed traction etc. are controlled as part of the overall system and burner controls. Furthermore, aerodynamic and electrical operating characteristics are governed by the channel geometry and magnetic field distribution, which are fixed. The channel power output is controlled by the channel electrical load system, described in Section 2.6. This establishes also the exact channel operating conditions.

For both design (full-load) and off-design (part-load) operating conditions, the electrical load parameter of the channel can be adjusted by varying the firing angle of the load inverters. The desired load parameter for full-load and part-load operation can be predetermined by calculations. These calculated values can be verified by testing, in which the channel is operated under full-load and part-load conditions over a range of inverter firing angles.

around the calculated values. The MHD efficiency can be established from knowledge of the channel power output and compressor power and optimum firing angles for each load condition can be determined. Implementation of this procedure will require the development of an algorithm for the adjustment of the channel loading so as to optimize plant efficiency under all load conditions.

2.6 ETF LOADING AND CONSOLIDATION CIRCUITRY CHANNEL

2.6.1 Introduction

For a large-scale MHD generator, circuits interfacing with both the channel and the load inverters are contemplated. These circuits serve the dual function of controlling individual electrode currents, and consolidating power from several electrodes, each at a different potential, for conversion by a single inverter. The ultimate configuration of these control and consolidation networks will be determined by considerations of cost, efficiency, flexibility and effectiveness.

Consolidation and control circuits are applicable to both Faraday and diagonal generators. Their method of application for both types of generators is shown in Figure 2-60. It is important to note that in addition to applications in regions of power take-off, the consolidation and control circuits can be used to regulate the cross-connection currents in the diagonal channel, if this is necessary. (Experience on the Avco Mark VI diagonal generator indicates that severe electrode current nonuniformities will develop if no control is used.)⁽¹⁾

A successful control circuit must satisfy the following criteria: (1) the circuit must be essentially nondissipative; (2) it must not short the Hall field in the generator; (3) it must not induce destructive arcing along the channel wall; and (4) it must provide control over individual electrode currents so that the channel can be trimmed (i. e., the process of adjusting the electrical loading of the channel to achieve a desired electrical and gas-dynamic operating condition).

2.6.1.1 Description of Three Consolidation Circuits

Three consolidation schemes which satisfy the above criteria are shown in Figure 2-61. The first circuit is an adaptation of rectifier and inverter circuits to the problem of control and consolidation in the MHD generator. The second circuit uses the mutual inductance of an autotransformer to achieve consolidation, and the third circuit uses a combination of inductors and capacitors to provide isolation between the two electrodes while the currents are combined at a common potential.

The operation of circuit I is similar to that of conventional rectifiers and inverters. The operation of circuits II and III will be described in following sections.

Figure 2-60 Applications of Consolidation Networks to Both Faraday and Diagonally Loaded Channels

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Figure 2-61 Three Circuits for the Control and Consolidation of Current from Two Electrodes Which are at Different Potentials

The three consolidation circuits shown in Figure 2-61 are most easily applied to pairs of electrodes. Larger numbers of electrodes can be consolidated with any of the three circuits by using either of the two cascaded networks shown in Figure 2-62. Network 1 in this figure has the advantage of handling considerably less power than network 2. This is demonstrated in Figure 2-63 where the power capacities of the two consolidation networks (expressed as a fraction of the total electrical output of the generator) are plotted as a function of the number of electrodes in a single consolidated grouping for a Faraday loaded channel. However, network 2 affords greater protection against a serious accidental shorting of electrodes which are at widely differing potentials (i. e., electrodes at either end of the group being consolidated). Although at this stage of development, it is premature to decide which of these two circuit configurations is the optimum, network 2 will be used in this design since it does afford greater protection from serious shorts.

2. 6. 1. 2 Operation of the Mutual-Inductance Consolidation Circuit (Circuit II)

As described by Rosa,⁽²⁾ the mutual inductance between two coils of an autotransformer can be used to maintain an approximately constant ratio between two electrode currents. Circuit II operates on this principle. By equating the sum of the voltages around a loop composed of the gap and the two autotransformer coils shown in Figure 2-61 to zero, the following equation for the divergence of currents I_1 and I_2 is derived (assuming a turns - ratio of one).

$$v_g = (L + M) \frac{d(I_2 - I_1)}{dt} \quad (2-14)$$

Although the rate of divergence of currents I_1 and I_2 can be kept small by designing the autotransformer with large inductances, an unacceptably large difference will eventually occur with time. In addition, as the volt-seconds accumulate across the windings of the coil, the core will saturate. This will result in dramatically lower values of L and M , and a corresponding increase in the divergence between the two currents.

To prevent both the core from saturating, and the difference in the two currents from becoming unreasonable, it is necessary to continually

CONFIGURATION 1

H2268

CONFIGURATION 2

Figure 2-62 Two Consolidation Arrangements for a Multiple of Electrodes

Figure 2-63 Power Capacity of Consolidation Networks for Two Arrangements Shown in Figure 2-62

switch the electrode currents between two or more autotransformers connected in parallel. If three parallel autotransformers are used, then it is possible to employ the three legs of the load inverter bridge for switching the current into each of the autotransformers. However, this arrangement would limit the operating frequency of the consolidation circuit to 60 Hz. If the consolidation circuit operates at a higher frequency, then the size of the inductive components can be reduced, and a savings in cost can be achieved. It therefore appears likely that the most economical arrangement will consist of two parallel autotransformers with a switching circuit independent of the line, operating at a frequency considerably higher than the line frequency.

2. 6. 1. 3 Operation of the Capacitive Consolidation Circuit (Circuit III)

The third consolidation circuit shown in Figure 2-61 operates as follows. The inductors (L) maintain an approximately constant current flow from the electrodes. The thyristors are switched such that only one is conducting at any particular moment. The conducting periods of the thyristors may not be equal. During the period when thyristor 1 is conducting, current I_1 flows directly to the load and current I_2 flows into the capacitor, charging it positive as shown in Figure 2-61. When thyristor 2 is gated on, the capacitor voltage appears across thyristor 1 with the proper polarity to turn it off. Current I_1 now flows into the capacitor reversing its polarity. The switching sequence is then repeated.

In general, increasing the switching frequency allows for a decrease in the size of the capacitor and inductors. At a given switching frequency, the ripple in the DC current is decreased as the size of the capacitors and inductors increases. The relation between switching frequency f , capacitance C , inductance L , and current ripple $\Delta I/I$, is

$$\sqrt{LC} \frac{\Delta I}{I} = \frac{1}{f} \quad (2-15)$$

Switching frequencies are limited by the turn-off time of thyristors, and will be on the order of a kilohertz. If current fluctuations are to be limited to 10%, then the LC time constant for the circuit should be,

$$\sqrt{LC} = 0.1 \text{ ms} \quad (2-16)$$

A trade-off can be made between L and C to minimize cost subject to the constraint that the capacitance not be reduced below a value which produces voltages greater than the thyristors can withstand.

Development of consolidation circuitry for MHD generators is still in its beginning stage. Research has concentrated on identifying potential circuits and understanding their operating characteristics. Since questions of cost, efficiency, reliability and fault protection have not yet been addressed, it would be premature to decide on an "optimum" consolidation network for the ETF.

For this stage of the ETF study, the Mutual Inductance Consolidation Circuit (Circuit II) has been chosen. This decision reflects the fact that at present this consolidation circuit has received more extensive experimental testing than other candidates at AVCO (and probably at other research facilities). This testing has included both bench-top operation of the circuit and operation of a four-electrode network on the AVCO Mark VI for a period of two hours. All test results have indicated that Circuit II will operate satisfactorily.

2.6.2 Channel Loading and Consolidating

In accordance with the ETF design criteria, the consolidation network has been designed such that it is representative of a baseload system. For the ETF network, this involved designing networks that consolidated a number of electrodes that would be typical of a baseload channel.

Although there exists considerable uncertainty in the optimum number of electrodes per consolidation network applicable to a baseload channel, it is reasonable to assume that a limit on the maximum number of electrodes will be set by the power capacity of the consolidation network. As shown in Figure 2-63, if the power capacity of the consolidation networks is limited to 10% of the load power, then for a channel characterized by a gap-to-Faraday voltage ratio of 0.01, the consolidation should be done in groups of five to ten electrodes. The size of the consolidation network will scale inversely with the gap-to-Faraday voltage ratio. Since a baseload channel will be characterized by a voltage ratio which is two to four times smaller than that assumed above, typical sizes for the networks will be roughly between ten and forty electrodes.

A second criteria in designing the consolidation networks is that the power capacity of each individual circuit be comparable to the others. This will to some degree standardize the load inverters thus reducing their cost.

Employing the above two design criteria, the consolidation systems described in Table 2-22 for the 3000°F preheat channel and in Table 2-23 for the 2500°F preheat channel were developed for Faraday loading. These systems are characterized by 44 load outputs for the 3000°F preheat channel and 40 load outputs for the 2500°F channel. Both systems employ loads of approximately one megawatt.

For diagonal loading of the channel, consolidation circuitry would be required in the power take-off region of the channel. In the Faraday channel, the consolidation network would utilize the mutual inductance circuit. The number of electrodes in a consolidation network is determined by the load current at a particular output point (shown in Table 2-71 and 2-72 for 3 and 5 terminal diagonal channels), and the average electrode output current in that region. For example, in the parallel-connected, three-terminal diagonal channel shown in Table 2-71, 3500 amps are delivered from the upstream power take-off to three individual loads. If the 1500 amp load connected to this point consists of two parallel bridges of equal current, then the consolidation network at the upstream network should consist of two circuits, each delivering 750 amps at a common potential. If the average electrode current, as determined by the performance calculations discussed in Section 2.3.2, is 46 amps at the upstream power take-off, then these two consolidation networks would each consist of 16 electrodes. The remaining consolidation circuitry at the upstream power take-off would be determined from similar consideration involving the other two loads.

A detailed design of a twelve-electrode consolidation network, based on the mutual inductance circuit which is proposed for the ETF, is shown in Figure 2-64. A photograph of the assembled hardware for this circuit is shown in Figure 2-65. Circuits consolidating more electrodes would be extensions of this network.

2.6.2.1 Instrumentation

Considerable information on the operation of the consolidation network is provided by the channel instrumentation, i. e., interelectrode voltage and

TABLE 2-22
SPECIFICATIONS FOR CONSOLIDATION NETWORK
3000°F PREHEAT

<u>Load</u>	<u>Electrode Number</u>	<u>Electrodes Per Circuit</u>	<u>Nominal Voltage</u>	<u>Nominal Current</u>	<u>R=V/I</u>
1.	1-24	24	970	1072	0. 90
2.	25-44	20	1070	971	1. 10
3.	45-62	18	1160	896	1. 29
4.	63-79	17	1240	838	1. 48
5.	80-95	16	1310	793	1. 65
6.	96-111	16	1380	753	1. 83
7.	112-127	16	1430	727	1. 97
8.	128-142	15	1480	702	2. 11
9.	143-157	15	1520	684	2. 22
10.	158-172	15	1560	666	2. 34
11.	173-187	15	1600	650	2. 46
12.	188-202	15	1630	640	2. 55
13.	203-217	15	1660	625	2. 65
14.	218-232	15	1690	615	2. 75
15.	233-247	15	1720	605	2. 84
16.	248-262	15	1750	595	2. 94
17.	263-278	16	1780	585	3. 05
18.	279-295	17	1810	575	3. 15
19.	296-312	17	1840	565	3. 26
20.	313-329	17	1860	560	3. 33
21.	330-346	17	1890	550	3. 43
22.	347-364	18	1920	540	3. 54
23.	365-382	18	1950	533	3. 66
24.	383-400	18	1980	525	3. 77
25.	401-419	19	2010	517	3. 88
26.	420-438	19	2040	509	4. 00
27.	439-457	19	2080	500	4. 16
28.	458-477	20	2110	492	4. 28
29.	478-498	21	2140	485	4. 40
30.	499-520	22	2170	479	4. 53
31.	521-542	22	2210	470	4. 70
32.	543-565	23	2250	462	4. 87
33.	566-588	23	2290	454	5. 04
34.	589-612	24	2330	446	5. 22
35.	613-636	24	2370	438	5. 40
36.	637-660	24	2410	431	5. 58
37.	661-685	25	2450	424	5. 77
38.	686-711	26	2480	419	5. 91
39.	712-738	27	2520	412	6. 11
40.	739-766	28	2550	407	6. 25
41.	767-796	30	2550	407	6. 25
42.	797-828	32	2500	416	6. 01
43.	829-863	35	2420	429	5. 63
44.	864-900	37	2320	448	5. 18

TABLE 2-23
SPECIFICATIONS FOR CONSOLIDATION NETWORK
2500°F PREHEAT

<u>Load</u>	<u>Electrode Number</u>	<u>Electrodes Per Circuit</u>	<u>Nominal Voltage</u>	<u>Nominal Current</u>	<u>R=V/I</u>
1.	0-29	29	1200	740	1.62
2.	30-52	23	1340	662	2.02
3.	53-73	21	1470	604	2.43
4.	74-91	18	1600	555	2.88
5.	92-109	18	1660	534	3.10
6.	110-127	18	1720	516	3.33
7.	128-144	17	1770	501	3.53
8.	145-161	17	1810	490	3.69
9.	162-178	17	1860	477	3.90
10.	179-195	17	1900	467	4.07
11.	196-212	17	1940	457	4.24
12.	213-229	17	1970	450	4.37
13.	230-246	17	2000	444	4.50
14.	247-263	17	2030	437	4.64
15.	264-281	18	2050	433	4.73
16.	282-299	18	2080	426	4.87
17.	300-317	18	2100	422	4.97
18.	318-336	19	2130	416	5.11
19.	337-355	19	2160	411	5.25
20.	356-374	19	2180	407	5.35
21.	375-394	20	2200	403	5.45
22.	395-414	20	2230	398	5.60
23.	415-435	21	2260	392	5.75
24.	436-456	21	2290	387	5.91
25.	457-478	22	2320	382	6.06
26.	479-500	22	2350	377	6.22
27.	501-523	23	2400	370	6.49
28.	524-546	23	2430	365	6.65
29.	547-570	24	2460	360	6.81
30.	571-595	25	2480	358	6.93
31.	596-620	25	2510	353	7.09
32.	621-646	26	2550	348	7.32
33.	647-673	27	2590	342	7.55
34.	674-700	27	2610	340	7.67
35.	701-729	29	2640	336	7.85
36.	730-758	29	2660	333	7.97
37.	759-789	31	2665	333	8.00
38.	790-822	33	2640	336	7.85
39.	823-859	37	2570	345	7.44
40.	860-900	41	2450	362	6.76

Figure 2-64 Circuit Diagram of 12 Electrode Consolidation Network

Figure 2-65 Front-View of Consolidation Network Showing Diodes and Heatsinks

electrode currents. In addition, transducers monitoring the current through the SCRs and the commutation capacitor voltage are required to detect abnormal operation. Table 2-24 summarizes the additional instrumentation required by the consolidation network.

2. 6. 2. 2 Fault Protection

It will be necessary to protect the consolidation circuitry from current overloads which may result from faults occurring in either the channel, inverters or the consolidation network themselves. The protective devices for the channel electrodes can also be used to limit current through the consolidation circuitry provided that this does not result in a sharp decrease in current through the series choke of the inverter. To insure that this does not occur, it will be necessary to activate a shunt across the load inverter (see Figure 2-66), when the consolidation circuitry is being disconnected from the channel.

2. 6. 2. 3 Efficiency

The power capacity of the consolidation circuits is a function of the number of electrodes in an individual network and the gap-to-Faraday voltage ratio of the generator. The gap-to-Faraday voltage ratio can be calculated from Figures 2-34 and 2-35 to be 0. 01 for the ETF channel. The consolidation is done in groups typically of 20 electrodes. According to Figure 2-63 for this and a 0. 01 voltage ratio, the consolidation circuits will have a power capacity equivalent to 20% of the total generator output power. Assuming that the networks themselves are 97% efficient this results in a net power loss for the consolidation network of 0. 6% of the output power. This loss will decrease with decreasing gap-to-Faraday voltage ratio. Therefore, for a larger machine the circuitry loss can be made very small, because the gap voltage remains the same and the Faraday voltage increases with increasing channel size. It is mentioned that the magnitude of the consolidation losses (0. 6%) is well within the accuracy of the calculations of channel performance. Hence the calculated channel power outputs are considered inverted with a total inversion loss of 2. 0%.

TABLE 2-24
INSTRUMENTATION FOR CONSOLIDATION CIRCUITRY

	<u>Voltage Transducers</u>	<u>Current Transducers</u>
Range	± 500 V	0-1000 A
Frequency Response	20 kHz	20 kHz
Isolation	10 kV	10 kV
Number	2 per load	4 per load

Figure 2-66 Protective Circuitry for Consolidation Network

2.7 MHD GENERATOR MAGNET

2.7.1 Basic Requirements and Design Approach

The superconducting magnet is a very large and critical component of an MHD system. While a number of superconducting magnets have been built and successfully operated with MHD generators in mind, none have approached the size that will be required, first, for the ETF plant and later, for the demonstration plant. The basic materials and technology needed for superconducting magnets are currently available, however, the MHD system requires large dipole magnets with long, straight sections of conductor, which, in turn, create special structural problems that are different from those encountered in other types of large magnets that have successfully been constructed. The conceptual design described here represents a first step in the preparation of the detail design and construction. In carrying this out, the primary guiding principle has been that the magnet should provide a specified magnetic field in a specified working volume reliably, for long periods of time, with minimal maintenance and with adequate safety. Small size and light weight are not, in themselves, primary objectives, but capital cost, and the need to conserve helium are very important objectives because the magnet cost is a significant part of the overall MHD system cost.

For MHD generators such as the ETF the magnet must produce a field perpendicular to the flow axis of the channel. In this study, the field direction is assumed to be horizontal.

The general shape for the winding is a combination of a saddle coil and a race track coil. The saddle coil geometry is considered to be the most suitable and economical configuration for linear generators based on past experience and investigation.⁽³⁾ However, the maximum field at any point in the winding can be significantly reduced by providing part of the required ampere turns as a race track winding.

The peak magnetic field considered on the center line of the ETF channel is 6 T and the active required length is 9 m. The warm bore of the magnet, dimensioned to accommodate the MHD channel with associated coolant piping and wiring, diverges along the active length from inlet to exit with an area ratio of about 2.5. It has been tentatively established by the channel designers that the desired field profile along the channel axis is one

which has a peak of 6 T - slightly downstream from the inlet and decreases toward the exit with the slope that results naturally from the divergent winding that conforms approximately to the divergence of the warm bore. This is discussed in greater detail in Section 2.2.

With the above requirements as a starting point, a number of different magnet configurations were investigated for the ETF design. These included the circular saddle and the 90° rectangular saddle design which was presented in an earlier study of large MHD magnets⁽⁴⁾ and the 45° rectangular saddle coil placed around a rectangular warm bore. Since this appears to be a departure from the results of the previous study in which circular saddle coils were found to be most suitable, a discussion of this point is appropriate.

In a large size magnet, the windings by themselves cannot support the forces created by the interaction of the winding current with its own magnetic field. These magnetic forces, amounting to tens of millions of pounds in a magnet of the ETF size, must be contained by a structure directly supporting the windings and operating at the same temperature as the windings. This containment structure is one of the major items of weight and cost. Therefore, in the conceptual design reported here, the materials have been chosen and attention has been given to the ease of fabrication, cost and cryogenic properties. A special effort has been made to providing modular components and to allow critical components such as the windings and structural items to be factory-fabricated as subassemblies. Wherever possible, the design has avoided industrial processes that require unusually large fabrication facilities or special equipment that must be brought to the magnet site for winding and assembly. It has been possible to design with structural shapes that are within fabrication capability which may be found in the plants of a number of structural manufacturers. Not only does this modular design permit the magnet to be assembled at the site but in event of a repair being necessary to either the windings, the structure, the insulation, etc., the magnet may also be disassembled in part or in toto. It is hoped that this design, therefore, will be considered as a model for the construction of even larger magnets.

The earlier study referred to above was based on a specified magnetic field profile sloping much more steeply from inlet to exit than that being considered in the current study (0.29 T/m vs 0.14 T/m). The steeper profile of the earlier study dictated the use of a winding with intermediate crossovers, and such crossovers are readily accomplished with the circular design described in that study. In the present design, if it is determined that a specific field taper is required other than the one that naturally results from the expansion of the winding from inlet to outlet, this will be accomplished by modifying the coils in the race track portion of the winding. In fact, it is possible through the use of a technique to be described in the next section to change the magnetic field taper after the magnet has been constructed.

Second, the earlier study determined that a rectangular saddle coil with 90° ends was less economic than a circular saddle coil. This result does not apply to a comparison between a 45° rectangular and a circular saddle coil. Based on current ETF requirements, it seems clear to us as will be shown in the next section that the 45° rectangular saddle magnet has a lighter, less expensive containment structure than either an equivalent circular saddle coil or a 90° rectangular saddle coil of the type described in the earlier study.

The vacuum tank is another component that has been designed as modular. The tank consists of four parts bolted together with vacuum seals at the joints. This is made possible because the vacuum tank has a single shell construction and all joints are at room temperature. The central part is fixed and supports the magnet weight on legs which pass through the vacuum tank and rest on a secure foundation. Two large, removable covers are bolted to the central part. When the covers are removed, the opening is large enough so that the magnet may be completely assembled or disassembled within the central part of the tank. The warm bore tube is the fourth part. The central part of the tank is deeply notched in the vicinity of the inlet. This permits the combustor to be located closer to the MHD channel than with a tank of conventional design, and thereby reduces the length of the duct between the combustor and the inlet nozzle.

In preparing this design, we have kept in mind that the cryogenic aspects of the magnet are as important as the electromechanical. The

vacuum tank, the power leads, the instrumentation leads, the structural feedthrough, etc., have all been proportioned so as to minimize the heat leakage. The support members that hold the winding and cold structure in place were designed to withstand not only gravity loads, but also loads resulting from the interaction of the windings with the channel and with magnetic materials located near the magnet. Large liquid helium filled volumes have been avoided in the interest of economy.

The equipment for refrigeration includes a recovery system for liquid and gaseous helium released when the magnet is warmed. The overall cryogenic and refrigeration systems appear to be well within the state-of-the-art and present no problems to the MHD magnet system.

The design of the magnet winding itself must take into account a number of special considerations. The peak magnetic field within the windings of a large air-core MHD magnet tends to exceed the central field by a significant amount and one of the factors that was considered in the choice of a winding design was to obtain the smallest peak field in the windings consistent with 6 T on the bore centerline.

A matter of particular importance in the design of large magnets is the destabilizing effect of mechanical motion in the winding during charging. Even with very careful design and construction, it is to be expected that some relative motion will occur. Such motion will result in friction which will tend to initiate a quench. Local quenching is also caused by flux jumps. Even though the multifilament superconductor may be designed to avoid flux jumps resulting from self magnetization, a twisted monolithic superconductor, will still exhibit flux jumps due to the self field instability.⁽⁵⁾ The method of dealing with this is to make the windings completely cryostatically stable so that they will recover reliably from any localized quench condition or flux jump.

As an additional precaution, the internal support system (substructure) will be made of stainless steel which matches the expansion coefficient of the copper winding quite closely. As a result, differential expansion between the conductors and the conductor support is very small. Motion under magnet forces is primarily between structure-to-structure interfaces and not directly at conductor-insulation interfaces.

A great deal of thought and calculation has gone into the design of each part of the proposed magnet. Although major problems are not believed to impede the design, it must be recognized that the construction of a magnet of ETF size and larger does present engineering challenges.

We believe that this conceptual design phase has produced a magnet design that is both suitable for the ETF and represents a model for a larger future power plant. There is additional work to be done in defining the forces, and demonstrating the suitability of the engineering solutions to the containment of these forces. The winding design on which this report is based employs a single current grade of conductor. It is well known that a multiply graded winding is more economical. It is therefore quite reasonable to conclude that the cost developed in this conceptual design may represent an upper limit to the magnet cost and that these costs may be reduced as a result of additional work that has been suggested in Section 1.6, Support Development Work.

2.7.2 Magnet Configurations

A number of variations of the saddle coil magnet were investigated to determine which type would give the best design. The basic dimensional information, which is common to all designs, is described in Figure 2-67, which is an axial cut through the central bore region of the magnet. The warm bore cross section is tapered from 1.35 m square at the inlet to 2.13 m square at the outlet. The windings have a straight section, ~ 9 m long and extend an additional 2 m on each side for end turns.

Two alternative types of winding designs were considered. They are:

1. A 45° rectangular saddle winding very similar to the concept of the CDIF winding. This has the advantage that the magnetic forces in the end turns of the winding are largely tension forces and therefore, the structural constraints needed in the end turn region are simpler and smaller than for other winding geometries.
2. A circular saddle winding, similar to designs that have formerly been extensively explored for baseload size magnets.⁽⁴⁾

Figures 2-68 through 2-71 are cross sectional views of the rectangular saddle design of the ETF magnet.

Figure 2-67 ETF Magnet Warm Duct Dimensions

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Figure 2-68 Cross Section View of ETF Magnet

H4952

Figure 2-69 Plan View of ETF Magnet. The combustor fits inside notch in central part of vacuum tank.

H4949

Figure 2-70 Cross Section Through ETF Magnet in Vicinity of Inlet

H4950

Figure 2-71 Cross Section Through ETF Magnet in Vicinity of Outlet

The magnet surrounds a square core tube. The windings are divided in two halves, each contained in its own helium container. Lorentz forces developed with the windings are transferred to the outer walls of the helium containers which are, in turn, supported by a truss and tension plate structural system. This will be described in greater detail below. The alternate magnet design uses a circular warm bore with magnet coils cylindrically arranged on both sides of the bore as shown in Figure 2-72. The Lorentz forces are again transferred to the outside wall of the helium container and constrained by a truss and tension plate system. The truss conforms to the circular shape of the windings.

A computer program was developed to determine the field distribution and forces for each of the two winding types. This work was done under subcontract by Magnetic Engineering Associates. The field plots and other data cited here are taken from their reports.

Figure 2-73 shows the axial field plot for the 45° rectangular saddle. Figure 2-74 shows the same data for the circular saddle winding. It is seen that in the central part of the channel both designs reach 6 T. The 45° saddle does not contain special provisions for a tapered field other than the fact that the central core tube is tapered. In circular saddle, on the other hand some of the turns are shorter than others, causing a steeper taper in the central region. However, the axial span between 6 T and 4 T which is the length of the magnetic field region, is the same in both designs. In the 45° saddle winding, the field drops at the outlet end far more rapidly than for the circular saddle. This characteristic is very desirable as it reduces the magnetic field at the diffuser and also at the boiler. Therefore, from this standpoint, the 45° saddle design is highly desirable. Additional field taper is not a problem if it should be needed. This can be provided most simply by reducing the length of the race track coil so that there are fewer ampere turns in the vicinity of the outlet and of the channel. We will see that the connections between sections of the magnet windings are made in a special splice box which is accessible for inspection or reconnection. Therefore, it is feasible that the race track winding be provided with several coils which may be reconnected via the splice box so that the magnet field taper can be adjusted after the magnet is built.

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Figure 2-72 Case Ib Illustrating Truss/Tension Plate Design Applied To Round Bore Magnet

Figure 2-73 Axial Field Plot for 45° Saddle Winding

Figure 2-74 Axial Field Plot for Circular Saddle Type of Magnet Winding

Figures 2-68 through 2-71 contain various numbered components which are now described. The vacuum tank houses the magnet winding and structure. It consist of a central main section (1) and two large covers, (1A). The covers are held in place by bolts but the vacuum seal, being approximately room temperature is made by gaskets or by inflatable seals. The tank design has been approved by a qualified tank manufacturer (Pittsburgh-DesMoines Steel Company) who have also submitted a budgetary estimate for its construction. The forces due to atmospheric pressure on the covers are transferred to the central structure by large pressure pads. The vacuum seal is independent of the structural support.

When the covers are removed, the magnet is exposed for assembly, inspection, or repair. All components may be inserted or removed. With the covers in place, the tank may be evacuated.

The superconducting windings are contained in two helium containers (2) that fit the windings closely and conform to the saddle shape. The conductors may have anys shape that is suitable. They may be rectangular bars with monolithic superconductors soldered to a copper strip, braids, cables or it may be a rectangular copper bar with superconducting wire or cables soldered in a spiral on the exterior surface. Whatever conductor is chosen, the general principles of fitting the windings to the helium container are the same. When the windings are finally inserted, they will be placed in structural submodules or channels which transfer the loads from the windings through insulation to suitable structural supports. A number of conductors are located within a single channel, separated by insulation. A number of channels are placed in rows and columns within the helium container. The number of conductors and the number of channel submodules are determined by comparing the forces on the winding and the allowed material stress. The submodules have a channel or U-shaped cross section. The walls are perforated by holes to permit liquid and gaseous helium to pass through the windings. The channels are produced in convenient lengths by welding flat plates or by rolling. The lengths are then joined together to form loops that are fitted to the helium container and conform to the desired winding shape. Then the loops are removed from the helium container and the windings and the insulation inserted. These channels have the correct perimeter to fit the helium

container. Even if they become bent during the conductor insertion, they will contain the correct length of conductor and when they are returned to the helium container, the windings will fit.

The insulation carries the conductor Lorentz forces in compression. It is proposed to mold this of a semirigid, appropriately filled thermoplastic material produced in convenient sheets. Grooves are molded into the opposite sides of the sheet in perpendicular directions so that there are both holes and helium distribution ducts automatically produced by the molding process. This is similar to the subplate construction of the CDIF magnet. When cooled, the material will become hard and rigid and be able to support the windings, however during the insertion of the windings the insulation will yield so that the windings may be driven into place by mechanical pressure. Splices between the ends of the conductor to create a continuous winding are made in a separate splice box. Within the helium container, in addition to the windings, there are filler blocks (6) whose purpose is to conserve helium.

After the windings have been placed in the channels, the channels placed in the helium container, all necessary instrumentation such as voltage taps, strain gauges, etc., connected, the container cover is then fitted and closed. The cover is located on that side of the container where there are only minor forces. Thus, it may be held in place by bolts prior to final sealing. It is planned that the seam and the bolts will be bridged over by thin metal sheets continuously welded to the helium container and the cover. These sheets are brought together and edge welded producing a hermetic helium seal. If the container has to be reopened, the weld may be cut away and a new weld created when the container is resealed. In this way, completely sealed winding modules are assembled at the winding factory and shipped to the magnet site. The windings may have two modules or more. In the latter case, the race track coils may be individually sealed in modules. In general, the windings will be too large to be shipped by conventional rail or truck. It is anticipated that for the ETF the winding assembly can be shipped by barge until it is close to the power plant site. After that, it will be transported by rail with "special handling" for oversize loads. When even larger magnets are constructed as for example for baseload power plants, it is possible that the assembly and sealing of the helium container may have to

be done at the magnet site. However, the construction and fitting of the individual submodules may still be done at the winding factory.

The splice box (3) is an integral part of each helium container. It is located at the low field end of the magnet in the end turn region. Conductor lengths will be chosen so that the splices come at the splice box. This box has its own cover so that the splices may be exposed without disassembly of any major portion of the magnet. Voltage taps and some of the instrumentation are located in this region. This box also collects gaseous helium liberated at the splices which is ducted directly to the gas vent without interfering with the cooling of the remainder of the magnet.

In the illustration of the helium container, five rectangular sections of windings are shown. One is a race track winding (4) and the other four are saddle windings. These five regions represent the division of the winding for computer simulation and are not intended to represent that there are five support submodules. The actual conductor dimensions and the size of the submodules will be discussed below. Ducts (5) introduce power, and other services into the windings. Two access ducts are shown although more may be required depending upon such factors as winding grades and liquid coolant distribution.

The choice of the rectangular cross section for the winding has produced a number of important simplifications in the magnet design. The helium container, fashioned of flat plates requires only a small amount of rolling for bends in the region of the end turns. The flat plates give large contact surfaces for transmitting forces between two containers and also between the helium containers and the force containment structure. The use of a rectangular winding permits a similar shaped bore for the MHD hot duct. This makes more room available for the MHD channel than a cylindrical winding and cylindrical bore.

The force containment structure surrounds the winding so that the only force transmitted outside of the structure is the magnet weight. The structural components are fabricated from separate modules which can be shipped to the magnet site, and assembled without special machinery. They consist of trusses held together by tension plates. These are made of flat aluminum alloy plates purchased from the mill and assembled with only a

small amount of machining required. Teeth cut in the end of the trusses mate with matching teeth in the tension plates and are assembled via bolts.

In the ETF power plant, the magnetic field is horizontal. There is a large vertical force tending to pull the straight sides of the winding apart. This is transmitted through the helium container walls to horizontal trusses (8) above and below the windings. These trusses are held together by tension plates (10) so that the vertical forces are contained. Between the trusses and the helium container, wedges (9) are employed which conform to the tapered shape of the container. Another large force is one of compression in the direction of the magnetic field. This causes the two winding halves to press against each other at the center line (7). These forces are internally contained and no external force results. Axial forces arise from two causes. The principle cause is the interaction of the current in the end turn region with the magnetic field. This force has a different direction depending on the location of the end conductor, however, the resultant force tends to unbend the end turns and separate them from the straight part of the windings. This force is taken in part by the helium container wall. It is supplemented by horizontal tension plates (14) again via teeth and bolts. Another source of horizontal force is the gradient of field in the axial direction that results from the field taper. This causes an imbalance between the force at the inlet and outlet end of the magnet during the charging process and during interaction with the MHD current. These forces are taken out by the diagonal stability rods (20) which are part of the magnet support system. At the inlet end of the magnet, trusses (12) and (12a) fit between the 45° end turns and the straight section of the windings. At the outlet end, truss (13) is similarly located. These trusses are connected by tension plates (14). Additional stiffening is required to react against the torque of the end turns. This is provided by fitting trusses (12) (12a) and (14) firmly to the helium container. There are also forces in the vertical direction which arise from the magnetic field in the end turn region. These are contained by tension plates (15) and (16).

Two teeth are shown on each side of the tension plate, however, it is clear that more than two may be used. The angle of the teeth which is $\approx 10^\circ$, is less than the angle of repose for unlubricated metals. By using

this angle, the tensile forces are contained in the tension plates and trusses and do not cause substantial loads on the bolts holding the plates and trusses together. During cool down of the magnet, differential expansion between the various materials cause the tension plates to shrink more than the winding. In order to avoid a thermal stress preload, the teeth are made with a predetermined amount of slack. The bolts holding the tension plates and trusses together have cupped washers and the teeth are able to slide for ~90% of the differential expansion range. The very last part of the contraction is picked up by final engagement of the teeth. The width of the tension plates is the same as that of the truss, however, it is displaced one-half a modular width in the axial direction. By so doing, the tension plates straddle two trusses and lend bending stiffness to the structure.

The magnet weight is taken out through the vacuum tank to the magnet foundation (21). There are four pedestals (17) that support the vacuum tank. The magnet weight is transferred to these pedestals by vertical rods (18). There is a section of thermal insulating material made of reinforced epoxy. This allows only a small transfer of heat between the vacuum container wall and the support rods. The magnet weight rests on support plates (19) attached to the structure. Because of temperature differences between the magnet and the vacuum tank, displacement occurs between the upper and lower ends of the support rods during cool down. All of the rods except one terminate in roller bearings which adjust to thermal displacement. Stability rods (20) are attached between the support pedestals (17) and the structural support system. These react against externally produced axial and lateral forces such as magnet-channel interactions or forces between the magnet and exterior ferromagnetic materials.

The surface of the tank is near room temperature whereas the windings and thermal support are at low temperature. Superinsulation is used as a liner on the interior walls of the vacuum tank. Two layers of superinsulation are used separated by a thermal shield at about 80°K which is maintained by the main refrigerator. During installation or service of the magnet, a temporary floor is placed over the superinsulation to protect it from damage.

The magnet warm duct (25) is shown square although a round or rectangular duct cross section would just as easily be incorporated. This contains

fittings and rails for positioning and supporting the MHD channel. There is also an ablative water cooled liner on the inside surface to protect the magnet from heat released from the channel if the channel walls become accidentally penetrated. On the vacuum side of the warm duct, there is a double layer of insulation and the intermediate temperature shield. The bore tube is bolted to the vacuum tank at the outlet end shown by (28). At the inlet end of the bore tube there is a sliding seal denoted by (26) which may have a bronze or teflon bearing surface capable of supporting the weight of the channel and the bore tube but not restricting axial motion. The vacuum seal at the inlet end is made by means of a diaphragm or flexible boot contained in a housing (27). Thus the vacuum integrity of the magnet is maintained while thermal expansion is permitted.

During the initial assembly, before the support rods are in place, the weight of the windings, helium container, and some of the structure rest directly on the bore tube. In this case, the insulation is compressed. However, when the support rods are fitted in place and the weight of the magnet removed from the bore tube, the resilient nature of the insulation allows it to expand to its full thickness and full insulating quality. During the initial construction period, it will be necessary to support a portion of the weight by a fixture so that the bore tube does not become over stressed.

This brief description will be amplified and supplemented in the material that follows.

The design of the truss and tension plate structure which contains the magnetic forces is now discussed. Figure 2-68 shows a longitudinal section through the magnet. Illustrated here is the 45° saddle winding with square bore. A rectangular truss is shown at the top and bottom of the coil container. The trusses are subject F_y forces, acting vertically. They are held by tension plates on each side which are bolted to the truss to form a modular structural element. To facilitate shipping and assembly, the truss/plate modules are one meter long in the axial direction, which gives a compromise between the number of modules and the weight. This provision, however, could easily be changed to be shorter or longer.

The magnetic forces acting on the windings and thus on the supporting structure are calculated by a three dimensional computer program.

The forces F_y (vertical), F_x (horizontal) and F_z (axial) are listed in Table 2-25 shown diagrammatically in Figure 2-75. They apply to Cut 3, which is approximately the mid-plane of the magnet and have been used for preliminary structural analysis. Forces F_x oppose each other in two adjacent winding containers. Thus they are in equilibrium. The F_z forces listed per quadrant are taken up by the winding containers and by the structure connected to the end turns. The F_y forces are contained by the truss and tension plate system.

TABLE 2-25
FORCE SUMMARY

	Cut	Force lbs/m		
		F_y (2 Quadrants)	F_x (1 Quadrant)	F_z (1 Quadrant)
Round Bore	3	10.5×10^6	$- 5.4 \times 10^6$	$- 0.18 \times 10^6$
Square Duct	3	9.6×10^6	$- 6.3 \times 10^6$	$- 0.15 \times 10^6$

Materials and construction techniques are discussed in the following paragraphs. Carbon steel is brittle at cryogenic temperatures, therefore unsuited to low temperature structures. Of various cryogenic materials which may be used, two candidates, austenitic stainless steel and aluminum alloy are particularly suitable and available commercially in many forms and gauges. The preferred material and the one used in the present design analysis of the containment structure is aluminum alloy 2014-T651 because of its light-weight, ease of machining, and suitable cryogenic behavior. The yield strength of this material increases in both longitudinal and transverse directions from about 60,000 psi at 297°K to about 85,000 psi at 4°K, for 1" to 2" thick plates. The elongation is about 12% at 297°K and is somewhat higher at 4°K. Supplier's data indicate that the material is available in plate thicknesses up to 6 inches, with some lowering of properties as the thickness increases. The structure is assembled by bolting. Where welding is required, as for the helium container and the winding submodules, the preferred material

a) SQUARE DUCT

b) ROUND BORE

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Figure 2-75 Force Diagram for Case Ia and Ib Truss/Tension Plate Design

is type 310 stainless steel because of its high strength after welding. This material in the annealed condition has a yield stress of 104,000 psi at 4°K. The tank manufacturer has a preference for type 304 stainless steel.

A modular approach has been adopted for the construction of the magnet. It offers advantages in shipping and final assembly. Two categories of modules have been considered: one is completely prefabricated, and can be assembled at the site using bolts or other fastening methods. In this category are structures supporting the winding container, such as trusses and tension plates, the winding module and the thermal insulation. The winding container assemblies are best fabricated in a shop where they may be instrumented, inspected and then shipped as assemblies. However, disassembly and reassembly is possible at the power plant site because of the modular nature of the windings. The vacuum tank belongs to the second category; it is prefabricated in sections and requires welding, in addition to bolting at the site.

Three concepts of structure to support the windings were evolved and evaluated before selecting the design. They are shown in Figure 2-76. Each of three concepts may be applied to either round or square bore designs. However, Case 1, consisting of two straight trusses and two tension plates is particularly well suited to the square duct design and was the only case studied in detail. Case 2, consisting of a ring support for the round bore containers was evaluated for weight. Similarly, the weight of Case 3 was determined, consisting of two C-Clamp shaped truss supports.

The structural design, Case 1 consists of two trusses, and two tension plates, tying the trusses together. The trusses handle bending and shear loads generated by the magnet forces, while tension plates provide reactions for the trusses, forming a closed frame. The square duct configuration simplifies the construction of the truss. The round bore magnet, having a conductor container of curved shape, is preferably fitted to a curved truss.

With the forces for 6 T design central field as determined by computer simulation, the maximum bending moment of the truss in Case 1a was found to be 1.3×10^8 lb in/m. The truss cross section is taken as uniform throughout its span to simplify design and manufacturing. It is

CASE 1 TRUSS/TENSION PLATES

CASE 2 RING

CASE 3 C-CLAMP TRUSS

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Figure 2-76 Three Structural Support Concepts Applied to ETF Magnet Construction

made of flat plates bolted together. Teeth are cut into the ends for mating with the tension plates. The maximum design stress is 46,000 psi. This gives a minimum factor of safety of 1.85 compared to the yield stress. This is the factor recommended by the aluminum association for aluminum structures used in bridges and similar type structures. Similar factors of safety were used in comparing the other truss designs. It is noted that the truss design is easily adapted to changes in the load carrying requirements that may arise because only straight sections such as plates are used in its makeup. The moment of inertia of a truss is about 79,000 in⁽⁴⁾ and the weight is about 2230 lb/ft.

The tension plates in Case 1a are required to carry a tension load of about 9.6×10^6 lb/m of length along the channel, due to magnet forces (4.8×10^6 lb on each side). The tension members consist of four inch thick plates with one inch of the thickness formed into teeth. In these plates, the tensile stress is 33,000 psi giving a factor of safety of 2.6 based on the yield stress for this aluminum alloy at 4°K. The teeth are subject to triaxial stress due to shear, bending, and bearing loads. Evaluation gives a combined stress of 20,000 psi in the teeth area. These stresses appear to be reasonable.

The axial forces created by the interaction between the end turns and the magnetic field have been estimated to be 32×10^6 lbs for each coil. The helium container is constructed of 3-inch thick Type 310 stainless steel and has sufficient cross section to support this in tension with an estimated tensile stress of 27,000 psi. (The yield stress is 104,000 psi for annealed Type 310 stainless steel at 4°K.) Notwithstanding this high margin of safety, it was decided to add horizontal tension plates on either side of the winding. This is because through the vicissitudes of construction the welds by which the helium container is fabricated may not have full penetration. Also, the bending forces in the end turn region should be partly reacted by the helium container and partly by the horizontal tension plates attached to the container at the extreme edge. The design also includes a separate set of horizontal trusses and vertical tension plates in the end turn region.

Two alternate structures were also considered in addition to Case 1. Case 2 uses a ring girder in place of the truss and tension plate. Case 3

uses a bolted C-Clamp truss. These both resulted in larger weight for the same loading. In the Case 2 structure, the two winding containers are held in place by circular rings having an axial length of 1 m each, similar to Case 1. The section required for the ring is 39 inches by 40 inches. For assembly purposes it is necessary to split the ring top and bottom and equip it with bolted flanges. The installation of the ring over the superconductor containers would be performed by moving the two halves horizontally until they come in contact with the containers and bolting them together. The ring concept was considered for the round bore magnet only, since it is not well suited to the square duct.

The concept of the C-Clamp truss, developed for a round bore magnet, involves 2 vertical structures, shaped like a letter "C", enveloping the winding container and meeting at the Y-axis. The trusses are calculated as though a pivot point existed at the Y-axis, top and bottom. For the actual configuration, a bolted flange would be used instead of a pivot. The flange is attached to the truss via a tension link (plate) capable of handling tension loads. The bending moment resulting from magnetic forces of the winding is opposed by the bending of the truss. The flanges also perform an assembly function, allowing a light preloading of the truss to the containers. The required section is 3 inches. Its construction is similar to Case 1 and 2 in other respects.

Comparative weights of the 3 concepts were determined. They include only the support structures. The container itself does not change in this comparison. Also, the effect of the container shape between the round bore design and the square duct design was not included. The weights were determined for 2014-T651 aluminum and are listed below:

<u>Support Structure</u>	<u>Weight (One Module)</u>	<u>lbs</u>
Case 1	Case 2	Case 3
15,000 lbs	24,650 lbs	20,750 lbs

Case 1 design is the best from the weight standpoint and from the overall design effectiveness of the magnet.

The estimated weight of the components and materials are calculated for the Case 1 square duct design are listed in Table 2-26.

TABLE 2-26
ESTIMATED WEIGHT OF ETF SUPERCONDUCTING
MAGNET COMPONENTS

<u>Item</u>	<u>Quantity</u>	<u>Total Weight kg x 10⁻³</u>
Conductor - Copper and NbTi	2.66 x 10 ⁷ AT	215
Insulation - Plastic	Various	15
Winding Submodules - Stainless Steel	30	200
Helium Container - Stainless Steel	2	330
Force Containment Structure Aluminum Alloy	Various	309
Legs - Aluminum Alloy	4	20
Vacuum Tank - Stainless Steel	1	276
Warm Bore - Stainless Steel	1	51
Intermediate Shield - Stainless Steel	1	13
Refrigerator/Compressor	1	10
Liquid He Storage Container	7500 gal	8
Liquid N ₂ Storage Container	2000 gal	2
Liquid He Charge	20,000 l	<u>1</u>
Total		1450

2.7.3 Refrigeration System

The conceptual design for the refrigeration system for the ETF magnet is similar to refrigeration for other large superconducting magnet installations. It will contain a refrigerator/liquefier capable of supplying helium gas at 4°K and also at about 80°K for intermediate shield purposes and heat stationing. JT valves located in the helium storage vessel and in the magnet vessel will convert a portion of the gas to liquid. The return gas flowing directly from dewar to the refrigerator/liquefier will be cold. However, for the power leads, the return gas is at room temperature or above and the refrigerator must be capable of accepting a portion of the return as warm gas. We have estimated that the system heat load will be 100 W plus 60 ℓ/hr of liquid helium. This includes a generous margin above the actual minimums. A commercial machine such as the TCF-200 is available which can supply 200 W of refrigeration plus 60 ℓ/hr of liquid helium. Our requirement would utilize approximately $3/4$ of the machine total capacity. No liquid nitrogen precooling is required. The machine capacity can be upgraded considerably if liquid nitrogen is supplied, with trucked-in liquid nitrogen, so that during cool-down an additional heat load can be accommodated. The machine also has sufficient capacity to supply the extra refrigeration required during charging and discharging of the magnet.

In addition to the refrigerator/liquefier, a liquid helium storage container is needed with an estimated holdup of 7500 gal. This container will be able to store the full liquid charge of the coil container in event that liquid helium is transferred out of the magnet prior to warm-up. It has sufficient space above the helium level for baffling to collect liquid thrown off during a rapid quench. A standard 2000 gal liquid nitrogen storage tank is also provided for initial cool-down. During normal operation of the coil it is not required.

Helium gas, generated during normal operation of the magnet, is returned via a compressor to the refrigerator liquefier, then to the liquid helium storage vessel. There is no need for gas storage under these conditions. However, it is conceivable that an emergency will occur in which large quantities of gas are liberated (such as produced by a quench) which exceed the

capacity of the compressor. Under these conditions, gaseous helium storage is required in the form of a gas holder. After a quench, it is expected that the coil and structure will have an average temperature of 60°K.

The other items included in the refrigeration system are all standard to this type of installation. The total refrigeration system cost has been estimated as shown in the section under cost estimation.

2.7.4 Charge and Discharge

The maximum rate of charge and controlled discharge of the magnet is determined by the eddy current loss in the conductor and magnet structure. This creates a heat load on the refrigeration system in addition to the normal heat load. There are three locations where loss due to eddy currents may be calculated. The first was in the conductor itself. This is driven by the gradient of the magnetic field in the axial direction. Calculations show that it is negligible compared to the other two eddy current loss terms. The others are the eddy current loss in the structural submodules and helium containers.

Consider a primary winding containing N_1 turns with an inductance, L_1 . The emf appearing across this during charging is given by

$$e_1 = L_1 (dI_1/dt)$$

Consider a secondary winding which is unity coupled to the primary. Then

$$e_2 = e_1 (N_2/N_1)$$

The power dissipated in the secondary is $P_2 = e_2^2/R_2$ where R_2 is the secondary resistance, $\rho_2 \ell_2/A_2$. In this way, we find

$$P_2 = \left(\frac{L_1 N_2}{N_1} \right)^2 \left(\frac{dI_1}{dt} \right)^2 \frac{A_2}{\rho_2 \ell_2}$$

This inductance, L_1 is 36.7 h based on an estimate that the stored energy is 2.5 GJ, $N_1 = 2100$ turns. There are two sources of eddy current loss. The principle one is the eddy current induced in the helium container. Based on the helium container dimensions, and a charging rate of 1 A/sec, the energy dissipated is 13.0 W. The second source of dissipation is in the winding

submodules. Here, there are 30 submodules in parallel and each is assumed to be a short circuited turn. Based on the submodule dimensions as calculated in a succeeding portion of this report, the energy dissipated is estimated to be 8.7 W with the same charging rate.

For a charging rate of 1 A/sec, the magnet may be charged in 3.6 hr and the dissipation is 21.7 W. The power supply required for this is 36 V. A 50 V power supply has been specified for the system. The same rate of discharge can also be accommodated by inserting a resistor in series with the magnet and reducing the power supply voltage to 0. The charging circuit and the protective resistance is shown in Figure 2-77.

It is believed that the ETF magnet will be inherently self protected. That is, if a quench should originate, the energy dissipated in the helium container and the structural submodules will be sufficient to drive the helium from the container into the storage tank and quickly spread the normal region throughout the entire magnet. In this case, there will be no local hot spots, most of the helium will be recovered, and the magnet can be re-cooled from about 60°K. However, protection can be furnished in the event of an interruption in the cables by connecting a resistor in parallel with the magnet. In order to limit the peak magnet voltage to $\pm 1,000$ V the resistors should be 0.153 ohms, as shown in Figure 2-77.

2.7.5 Instrumentation and Controls

The instrumentation and controls described here pertain to the following subsystems: Magnet, magnet power supply, refrigeration system, vacuum system and helium recovery system.

The measurements required within the magnet are magnetic field, current, temperature, stress (strain) and EMF across voltage taps. Controls needed are for magnet charging voltage and the value of the resistance in series with magnet, that determines discharge rate. The emergency dump circuitry should be automatic and requires no actual control, but an over-riding panic button is desirable.

In the magnet power supply, instruments and controls are usually furnished by the manufacturer. Refrigeration controls are more difficult to specify because they will only supplement the controls furnished by the refrigeration subcontractor. In addition to the basic refrigerator, there

CHARGING

STEADY STATE

CONTROLLED DISCHARGE

BROKEN POWER LEAD

H5038

Figure 2-77 Charging, Discharging and Protection of ETF Magnet

will be tanks for liquid helium and liquid nitrogen, vacuum insulated transfer lines, a power and instrument lead in system entering the magnet, vacuum pumps and valves, a helium recovery system including low pressure gas storage and an emergency cryogenic fluid dump system to be used if the normal storage system is overloaded. It is assumed that the basic instrumentation and controls needed to operate the various components are all supplied by the component manufactureres. Therefore the instruments listed below are only those necessary in ensure proper system integration. In the liquid storage tanks, we will need level and pressure measurement. Many valves will be remotely operated. We will need to know valve position. For vacuum systems, we will need vacuum gauges on each side of valves. For the transfer lines, we will need a differential pressure gauge and orifice plate calibrated in terms of flow or some other type of flow measuring device. For the helium recovery system, we will need storage tank pressure and temperature. For the emergency dump, we will need pressure and temperature. Controls will be determined in more detail by an analysis of the operating and failure modes.

The following information pertains to the type of instruments that are suitable. Magnetic field measurement is required in the warm bore of the magnet and at a number of locations within the windings, used for system checkout. In the warm bore, a rotating gaussmeter is most appropriate. In the magnet windings, both Hall-effect probes and copper magneto resistance probes may be used. Current is measured by ammeters and recorded in the "data acquisition and recording system." For diagnostic purposes; about fifty voltage taps are useful. Voltage levels will be between a few millivolts and several volts. Because of the large number of readings and the need for early warning, a scanning and recording system is useful. For control and for determination of an abnormal operating condition requiring a dump, four voltage taps out of the large group should be sufficient. Temperature measurement is required. Down to about 20° absolute, copper-constantan thermocouples may be used. About 20 of them are required throughout the magnet to check on the temperature gradients during cooldown. For the final stages of cooldown, germanium resistance thermometers are used. The germanium thermometers and copper resistance thermometers

are affected by the magnetic field. Therefore, to determine the temperature gradients during a quench, before the magnet is fully discharged, carbon resistance thermometers are used. Three or four of these placed strategically within the magnet will suffice. They are read out by resistance bridges and recorded along with the other data. Pressure is measured by conventional pressure transducers which are usually supplied by the manufacturers of the tanks or other equipment. Gas flow is measured by pressure drop across an orifice in a line or by a "rotameter." Vacuum measurements are by ion gauges or equivalent. Liquid level is determined in a variety of ways depending on the type of liquid. For liquid nitrogen tanks, float-type level gauges can be used. Also the tank weight can be read out. The latter is particularly useful when there is fluctuating level change due to boiling or other turbulence. Strain (and therefore stress) is read by conventional strain gauges. Adhesives are available for use at liquid helium temperatures. Between 50 and 100 strain gauges will be required at various points in the structure and winding and they should be read by a scanning system.

There is always danger that the MHD channel may spring a leak and the extremely hot gases are liberated within the magnet bore. Puncture of the magnet warm bore tube may be delayed but only for a short time, by the use of a water-cooled teflon ablative liner as warm bore insulation. However, it is very important to detect a flame or arc condition within the bore by using a CO₂ or ion type flame detector. This must initiate a shutdown of the MHD channel before damage occurs to the magnet cryostat.

There are also interaction forces between the channel and the magnet itself which are caused by pressure variations within the channel. These forces can be quite large and results in some vibration of the channel walls. Vibration pickups and force gauges are useful at attachment points to the magnet to monitor these forces.

2.7.6 Parametric Design of Superconducting Magnets for the ETF

It is possible to design a superconducting magnet for an MHD power plant by parametric analysis that exhibits the tradeoffs involved in design choices. This discussion is based on a simple model which gives an insight into the design procedure and preliminary design data. It is backed up by computer simulation of the ETF magnet.

The model magnet that was chosen is illustrated in Figure 2-78. It is a two-dimensional model. The coils are approximated by large area conductors with area $Y \times 2Y$ spaced on two sides of the channel region. The magnetic field is B_x in the center of the channel. The coil area is related to the dimensions of the magnet and magnet bore, b_c , according to the following geometrical relations: See supplementary sheet for an explanation of the symbols.

$$\begin{aligned} Y &= (A/2)^{1/2} \\ A &= NI/J \\ r_o &= (b_c + Y)/2 \\ r_c &= [r_o^2 + (Y^2/4)]^{1/2} \end{aligned}$$

A single conductor containing $NI/2$ ampere turns produces a magnetic field at a distance r_c from its center.

$$B_1 = -\mu_o NI/4\pi r_c \quad (2-17)$$

If the field due to four conductors are added vectorially, the central field is

$$B_o = -\mu_o NI/\pi r_o [1 + (Y/2 r_o)^2] \quad (2-18)$$

The relations are only slightly sensitive to the precise shape of the windings. It is convenient to replace B_o by a new variable.

$$B'_x = B_o [1 + (Y/2r_o)^2] \quad (2-19)$$

The MHD magnet is estimated to have a cold bore b_c of 2.06 m for a 45° saddle magnet. Also, the central magnetic field B_o will be 6 T in the highest field region. With these data, we can determine the total number of ampere turns required and the overall current density from Eq. (2-18). These are shown plotted against the coil build, Y , in Figure 2-79.

Another equation is derived from the fact that the conductor is a stable superconductor. If J is the overall current density, and J_c is the current density in the conductor, it follows that

$$J = NI/A = J_c p_c \quad (2-20)$$

where p_c is the winding packing factor. The current density within the superconductor is $J_s = (R + 1) J_c$ where J_s is the maximum allowed current density at the temperature and in the maximum magnetic field region of the coil.

H1728

Figure 2-78 Diagram of Model and Notations for Calculation of Optimum MHD Magnet Dimensions

H1729

Figure 2-79 Curve Showing Dependence of Current and Current Density on Build

The superconductor will be a niobium titanium, multifilament, monolithic superconductor with the superconductor area occupying $1/R$ of the substrate area. The coil will probably operate at a temperature slightly in excess of 4.2° to provide a pressure slightly above atmospheric within the helium container. Let J'_s be the minimum superconductor current density guaranteed by superconductor manufacturers at 4.2°K . In terms of J'_s , we can write $J_s = c_j c_s J'_s$. Here, c_j is a correction factor that converts the published current density at 4.2°K to that for the operating temperature and c_s is a safety factor to ensure that there is a satisfactory operating margin between the critical current designed into the magnet and the minimum specified conductor current. The well-known, full cryo-static stability condition for the conductor may now be written

$$q'' = \frac{\rho J_c^2 d w}{2dC_d + 2wC_w} \frac{R+1}{R} \text{ W/m}^2 \quad (2-21)$$

In the above, q'' is the allowed surface heat flux if all of the current flows in the substrate at the maximum magnetic field of the conductor. ρ is the resistivity of the substrate after correction for the magneto resistance at the maximum magnetic field, d and w are the dimensions of the conductor, assumed rectangular, with d signifying the y dimension of the conductor. d is perpendicular to the central magnetic field. C_d is the fraction of the d side that is exposed to coolant and C_w is the fraction of the w side that is similarly exposed. For the resistivity of copper, the relationship used is $\rho = \rho_0 (1 + 0.4B)/150$.

Still another relationship comes from the pressure exerted on the conductor in the maximum field region. Identify B'_x as the approximate maximum magnetic field seen by the conductor. The pressure on the conductor or the insulation placed on the w side is given by

$$P_w = \frac{B'_x J_c d}{1 - C_w} \quad (2-22)$$

Consider that there are a number of conductors separated by insulation all contained within a single slot in a structural channel or subplate. Each conductor transmits its Lorentz pressure to its neighbor until a value, P_m is

reached, which is judged to be the largest pressure that the design will permit. Then $P_m = n P_w$.

Equations (2-19), (2-21) and (2-22) may be combined into a single equation that is useful for parametric design.

$$\left(\frac{P_m}{B_x} \right)^2 \frac{\rho}{2q} = Z = \frac{dC_d + wC_w}{(1 - C_w)^2} n^2 \frac{d}{w} \quad (2-23)$$

The quantity Z is a magnet design constant with the dimensions of length that can be evaluated without reference to a particular winding. It contains the desired magnetic field, the allowed maximum pressure on conductor or insulation, the copper resistivity at the maximum magnetic field and the allowed surface heat flux for the conductor. These are quantities that are predetermined for the particular application. On the right side, the quantities relate only to the conductor dimensions the amount of surface that is exposed to liquid helium and the number of conductors in a slot. The magnet size does not appear. The winding buildup, Y appears only very weakly in the function B_x . The forces that react on the structure of the magnet are now calculated. These are evaluated in the maximum field region near the inlet. A single grade of winding is assumed with uniform current density. In the axial plane of the coil, ($x = 0$) the magnetic field, B_x is estimated within the winding. It has two components, B_{x1} due to the nearby conductors and B_{x2} due to conductors on the remote side of the channel. The first goes linearly with the distance from the center of the conductor because it is in the interior of a winding with uniform current density. The second component goes as $1/\text{distance}$ from the center of the conductor because it is external to the winding. The total magnetic field is the result of four currents, two nearby and two remote carrying opposite directed current. Therefore, $B_x = 2B_{x1} - 2B_{x2}$.

From elementary considerations, it can be shown that

$$B_{x1} = \mu_o J [Y + b_c - 2y]/4 \quad (2-24)$$

and

$$B_{x2} = - \frac{\mu_o J Y^2}{2\pi (y + b_c + Y/2)} \quad (2-25)$$

$$B_x = \frac{\mu_o J}{2} \left(Y + b_c - 2y + \frac{Y^2}{\pi(y + b_c + Y/2)} \right) \quad (2-26)$$

To derive an expression for the pressure exerted by the conductor on the external structure consider a thin slice of conductor of area dy by $2Y$. The pressure on this current element is $dP_y = J B_x dy$. The total pressure is the integral over the conductor.

$$P_y = \frac{\mu_o J^2 Y^2}{\pi} \ln \frac{3Y + 3b_c}{Y + 3b_c} \quad (2-29)$$

The total force is $F_y = 2YP_y$. Figure 2-80 is a plot of the force and pressure for various values of coil build factor Y with the current density J taken from Figure 2-79. This is an interesting result because it shows that the total force is actually reduced by using a small coil build and therefore, a high current density. Since the cost of a magnet depends on the dimensions both of the conductor and of the structure, a small winding build-up results in a more economical magnet.

Equation (2-27) actually overestimates the pressure because the magnetic field lines within the coil are assumed to be parallel to the X axis. Comparing the data of an actual computer simulation of an ETF magnet having a build of 0.8 M shows that Figure 2-80 overestimates the pressure and forces by about 25%.

It is now reasonable to examine the type of superconductor that may be used to wind the ETF magnet. Two types are considered. The first is a square conductor with the sides d and w equal. The exterior surface of the conductor is wound with a spiral of monolithic superconductor. This gives a complete transposition of the current. Also, the exterior surface is larger than the conductor outline because of the increased surface of the individual superconductors. An example of this conductor is shown in the Figure 2-81. Another example of this type of conductor is one of the two chosen for the Winding Model Magnet program recently issued by the MIT Francis Bitter National Magnet Laboratory.

Another conductor that is suitable for winding into magnets of large size is a flat strip containing one or more monolithic superconductors

Figure 2-80 Curve Showing Dependence of Pressure and Force on Magnet Structure on the Coil Build Up

Figure 2-81 Cross-Section Sketch of Rectangular Cable Type Conductor

soldered into a groove or grooves in the strip. A group n of these placed side by side are contained in a slot in a winding module. Clearly the limitation on the number of conductors, n , is given by the maximum pressure that one allows on the insulation or the conductor.

We will consider first, the implications of the use of a rectangular conductor having superconductors soldered in a spiral on its exterior surface. Figure 2-82 is a plot of the magnetic field vectors in the midplane of a typical MHD saddle wound. It also represents the magnetic field vectors in the ETF magnet. The forces on the conductors are perpendicular to the vectors. In the conductors near the channel, the forces are nearly vertical. In the conductors at the top of the winding, the forces are principally horizontal and toward the axis. Unless one wishes to have a different insulation system in different parts of the winding, a conductor must be capable of withstanding magnetic forces in both the X and Y direction.

Conceptually, consider that the magnet windings are enclosed in a helium container box. It is rectangular in cross section for the 45° saddle type winding which is now used as a model; the packing factor p_c is the ratio of the total conductor cross section to interior area of this box. Consider that before the conductors are wound, a set of channels are constructed and fitted inside the box so as to form winding modules. Each module contains a number of conductor turns, the number being determined by the maximum pressure that one allows on the insulation or the conductor. These winding modules are fitted within the box, shaped, welded and shimmed, if necessary, until they fit the winding space accurately. They are then removed, the windings placed in them and returned to the helium container. Since these channels have the correct developed length, if some distortion occurs during the winding, they can be returned to the correct shape and the windings will again fit. The winding modules then take the force developed within the conductors into their own structure and transfer them to the container walls.

In choosing an allowed compressive stress for insulation, we note that unfilled nylon at -170°C has a yield stress of 35000 psi. At 20°K , glass cloth reinforced teflon has a yield of 85,000 psi. Therefore, a low temperature working stress of 10,000 psi ($6.9 \times 10^7 \text{ n/m}^2$) is considered

Figure 2-82 Magnetic Field Vectors in Warm Bore Region and Winding Region of CDIF Magnet Midplane

to be very conservative and in keeping with the novel nature of this design concept. Nevertheless, it does result in an acceptable design.

The conductor is assumed to be square with dimensions $d \times d$. The extended surface of this type of conductor permits roughly twice the heat flux that one would allow for a smooth surface conductor, however, because the pressure is taken on the small rounded surfaces of the conductor wires, this conductor will permit a maximum pressure that is approximately half of what one would use with a smooth conductor. The values chosen for Eq. (2-23) are therefore:

$$\begin{aligned} P_m &= 3.45 \times 10^7 \text{ n/m}^2 \text{ (5000 psi)} \\ B'_x &= 6.5 \text{ T} \\ \rho &= 3.74 \times 10^{-10} \text{ ohm m} \\ q'' &= 8000 \text{ W/m}^2 \end{aligned}$$

With these values, we find that $Z = 0.658 = 4 d n^2$. This is plotted in Figure 2-83. Unless one wishes to reduce the conductor current density or use larger values of C_d , this relation gives the largest size of conductor that can be accommodated in the slots of the winding module.

We now calculate what value of packing factor, p_c , is allowed by particular winding design. The stability relationship, Eq. (2-21), may be rewritten

$$d = 4 p_c^2 C_d q'' / \rho J^2 \quad (2-28)$$

This is combined with Eqs. (2-22) and (2-19) to give

$$p_c = \frac{\pi P_m}{16 \mu_o} \frac{b_c + Y}{Y^2} \frac{\rho}{q''} \left(\frac{1}{2 n^2 C_d} \right)^{1/2} \quad (2-29)$$

For the square conductor,

$$p_c = 0.252 \frac{2.06 + Y}{n Y^2} \quad (2-30)$$

This is shown plotted in Figure 2-84 against the coil build, Y as the curve labelled "required $n p_c$."

Another relation that can be used to fix the design parameters is to determine what value of p_c is reasonable for a particular magnet winding. The number of turns of conductor that may be placed within the channel is

Figure 2-83 Curve Showing Dependence of Square Conductor Size On the Number of Conductors per Winding Module

Figure 2-84 Curve Showing Comparison of Required and Available Packing Factor for the Square Conductor Winding Design vs Winding Build

n^2 . Between conductors, there is insulation. According to the present module design, the insulation may be sheets of a molded thermo-plastic material. Grooving on the opposite sides run in perpendicular directions so that there are both perforations and helium distribution channels produced by the insulation. It is similar to the sub-plate construction of the CDIF magnet. However, because the insulation carries the winding stresses in pure compression, they may be made of a semi-rigid thermo-plastic material, such as teflon or one of the vinyl compounds appropriately filled. When cooled below liquid nitrogen temperature, the material will become hard and sufficiently rigid to support the windings. However, during the placement of the windings, it is expected that it will yield so that the windings may be driven into place by mechanical pressure. With this model, we may now estimate the packing factor. Let v_c be the conductor volume $= n^2 d^2$. The insulation volume is v_i . Assume that the insulation thickness is 0.004 m between turns and 0.012 m thick adjacent to the channel wall. This additional thickness assists in the distribution of helium from channel to channel. We have then

$$v_i + v_c = [nd + 0.004 (n-1) + (2) (0.012)]^2 = [n(d + 0.004) + 0.02]^2 \quad (2-31)$$

The winding submodule is stainless steel, type 310. This material in the annealed state has a yield stress of 104,000 psi. The allowable stress is two-thirds the yield stress for this material. In the interest of conservative design, we are choosing one-half the yield stress or 52,000 psi for S_s ($3.6 \times 10^8 \text{ n/m}^2$). The winding channel is metal with a compressive stress $S_s = 25,000 \text{ psi} = 1.724 \times 10^8 \text{ n/m}^2$. The number of conductors contained in one-half of the helium container is

$$\frac{N}{2} = \frac{NI}{2J_c d^2}$$

The total force transmitted to the end walls of the helium container is

$$F_s = IB'_x \sqrt{N/2}.$$

The Thickness of the module wall is therefore $t_s = F_s / 2S_s$

$$t_s = \frac{IB'_x}{2d} \frac{Y \sqrt{p_c}}{S_s} \quad (2-32)$$

The volume of metal required for the module is

$$v_s = 3 t_s \times \sqrt{v_i + v_c} \times 1.4$$

The factor 1.4 is an estimated factor that allows for holes that perforate the metal, plus some allowance for fastenings, shims, etc. With these equations, it is now possible to write the packing factor

$$p_c = \frac{v_c}{v_i + v_c + v_s} = \frac{n^2 d^2}{(v_i + v_c) + 4.2 t_s \sqrt{v_i v_c}} \quad (2-33)$$

For an assumed build, Y and an assumed n , a trial value of thickness t_s may be calculated from Eq. (2-32). The conductor size d is taken from Figure 2-83 which in turn is derived from the constant Z . The various volume terms are then calculated and a value of p_c obtained in accordance with Eq. (2-33). The procedure is iterated until the calculated and trial values agree. The values of $n p_c$ required to fit the conductors into the helium box are plotted on Figure 2-84 for $n = 2$ and $n = 3$. Where they cross the required $n p_c$ line, we have fixed the coil build. A conceptual diagram showing conductors installed in a channel (module) and modules installed in a helium container is shown in Figure 2-85.

To summarize, the constant Z can be evaluated without reference to the details of the conductor design. This leads to a required packing factor which depends on the assumed coil build up. A winding construction is assumed and a packing factor is derived. Comparison of the required value and the realized value of the packing factor fixes the build up.

A second type of conductor that was evaluated is constructed of a monolithic, multifilament niobium titanium superconductor soldered to two copper conductors to form a rectangular strip. With this conductor, there are no external wires to raise local stress in the insulation. Therefore, P_m may be chosen as $6.9 \times 10^7 \text{ n/m}^2$ (10^4 psi) compared to half that value for the conductor previously considered. The surface heat flux $q'' = 4000 \text{ W/cm}^2$ and the other values are the same as for the square conductor. Based on evaluation of Eq. (2-23), $Z = 5.27$. With so large a value of Z the allowed packing factor may be larger than for the square conductor. The insulation thickness between turns can be increased to 0.007 m. With these values Eq. (2-23) becomes

HI960

Figure 2-85 Detail of One Concept of Modular Winding. The Square Spiral Wrapped Conductor is Illustrated in this Winding Design

$$n^2 d = \frac{A}{2 + 2 d/w} \quad (2-34)$$

Corresponding to Eq. (2-28),

$$p_c = \frac{\pi}{8 n \mu_o} \frac{P_m \rho}{q''} \frac{b_c + 7}{Y^2 + Y^2 d/w} \quad (2-35)$$

Since the support module is U shaped, there are twice as many walls in one direction compared to the perpendicular one. Therefore, the winding submodule is able to transmit twice as much force in one direction compared to the other. It is, therefore, appropriate to stack approximately twice as many conductors in the strong direction compared to the weak. In the high field region, the magnetic field is nearly horizontal, and therefore in this region, the two walls should be vertical. The winding should be inserted with the w dimension (the short dimension) vertical and the d dimension horizontal. A large number of trials of Eq. (2-35) were done to determine how the ratio of d to w and the number of conductors in a submodule affect the resulting packing factor. The details of the calculations apply only to the particular model chosen. This model has a single conductor grade and a single value of current density. The conclusion of this work is that a balanced winding design would contain 70 conductors per submodule, 14 per row by 5 rows. There would be 30 submodules in the two helium containers. The conductor dimensions would be 0.033 x 0.011 m. This conductor would carry 13,100 A at 4.5° and 7 T, conservatively and the winding would have 2.66×10^7 A turns. The overall packing factor that applies to this design is $p_c = 0.40$.

2.7.7 Estimation of Cost for the ETF Superconducting Magnet

The cost of the magnet is given in Section 1.6.1 and are based on June 1977 prices.

- Winding cost is taken from a quotation by a major conductor manufacturer. This applies to a conductor conservatively rated at 13,100 A at 4.5° and 7 T. The conductor will be supplied in single lengths for winding a single submodule. A total of 215,000 kg of conductor will be required at a cost of \$14.48/kg. This is \$0.0010/A ft.

- Winding submodules will be fabricated from three flat strips of Type 310 stainless steel, perforated and welded into U shaped channels. They probably can be purchased from a steel mill in convenient lengths. Cost are based on about 200,000 kg of material at \$1.75/lb. Manufacturing cost is an engineering estimate for machining time and welding cost. The latter was obtained after consultation with a welding shop where there is a submerged arc, automatic welding machine. An allowance was also made for jigs, straightening of the welded channels, assembly into submodule loops, fitting to the container and rework of the fitted pieces after the first trial assembly. The estimated cost is \$953,000.
- Helium container is to be fabricated from 3-inch thick, Type 310 stainless steel, submerged arc welded from plates to larger sections, shipped to the winding shop and welded into a complete winding module. An allowance has been made for drilling and tapping of the bolt holes for the helium container cover and attachment of the helium seal material by continuous spot welding. Material cost is 1.75/lb. Welding costs have been estimated by scaling the welding cost developed for the submodules by the square of the thickness. The helium containers are estimated to weigh 330,000 kg and to cost \$3.64 million.
- Other winding cost estimates were made for detail design, insulation, winding labor, winding tooling, and shipping. The total cost of windings delivered to the site, \$8.34 million, is entered into Table 1-23
- Vacuum tank was first designed to be fabricated of aluminum alloy. It was sent to a major tank manufacturer (Pittsburgh-Des Moines Steel Company) who approved the design with modifications, but expressed a preference for fabrication from Type 304 stainless steel. It included their method of sealing large covers by separating the load support structure from the sealing. They have estimated the weight at 276,000 kg including scrap and the cost at \$3.4 million. This includes the cost of design, drafting, procurement of materials, fabrication, site erection, and vacuum leak testing.
- Warm bore tube design was based on stainless steel, two inch thick. A cost estimate of \$380,000 was obtained from the same company

that estimated the vacuum tank, including design, fabrication, delivery to site and field assembly. The warm bore will weigh 51,000 kg.

- Miscellaneous vacuum tank cost, in addition to the main fabrication items, were estimated for producing and installing the intermediate temperature shield, superinsulation, power leads, internal fittings and the detail design of components other than the tank and the warm bore itself. The combined cost of the vacuum tank is estimated as \$4.19 million and is entered into Table 1-23.
- Force containment structure: This consists of the aluminum alloy trusses, tension plates and legs. The material cost, for 2014 T 6 alloy is \$1.29/lb. To obtain the machining cost, drawings of typical trusses and tension plates were sent to three shops for bidding of the machining cost. One shop bid \$0.08/lb. The remaining shops bid 0.32/lb and 0.40/lb. In our judgement, the lowest price is too low and we have chosen to use the geometric mean between the lowest and middle price, namely, 0.19/lb. To this we have added design tooling, inspection and shipping to arrive at a total manufactured, delivered price for aluminum alloy structure at \$1.79/lb. The structural material will weigh 329,000 kg, and will cost \$1.754 million.
- Refrigeration: The refrigerator compressor is based on an estimate made by C. T. I. for their TCF-200 machine with a compressor large enough to supply all of the cooling. It includes supervision of installation. In addition, we have estimated the cost of a 7500 gal liquid helium storage tank, a 2,000 gal liquid nitrogen storage tank, gaseous helium storage, 20,000 liters of an initial supply of liquid helium, detailed design installation and shipping. The system is estimated at 28,200 kg and will cost \$1.015 million.
- Shipping costs are based on shipment by truck for an average distance of 1500 mi. Two shipments per item have been estimated, from the mill to the fabrication shop and from the shop to Butte, Montana. Shipping cost is \$0.055/lb for each shipment.
- Other costs: In addition to the above mentioned items, engineering allowances were made for a 50 V, 15,000 A power supply with diode

protection, a series resistance for controlled discharge; a circuit breaker for short circuiting the resistor, a water cooled protective resistor to be placed across the magnet terminals; installation of the cables between the power supply and the magnet. Instrumentation and controls include design, materials, and hook up. The cost of the above is \$583,000.

- Installation cost at the site included labor, material and tooling. Tooling will be required to raise and install the helium containers. These same tools can later be used for handling of the two doors of the vacuum tank. Assembly labor, engineering supervision, also miscellaneous cryogenic piping and electrical assembly between the magnet and the control station was estimated. These items, combined, come to \$227,000 and are entered in Table 1-23.

It is interesting to examine the cost per pound of the stainless steel components. The raw material cost is about \$1.75/lb. We have two bids from PDM for the manufacture of large structures including field erection. These include a great deal of welding. For the tank, the installed cost is \$5.50/lb. For the warm duct, the installed cost is \$4.26/lb. This may be compared with our cost estimates for the helium container and the winding submodules which, in many ways, resemble the warm duct in that they are fabricated from many plates by welding. There is also fitting, drilling, sealing, and design and shipping. Our estimate, based on analysis of each step of manufacture, but not field assembly, comes to \$4.25/lb. This is very close to the estimate by PDM.

2.8 COAL COMBUSTOR

Prior to and during the conceptual phase of the ETF combustor an extensive survey was made of the literature related to coal combustors suitable for coal fired MHD systems. The survey included the basic principles of coal combustion, slag handling and chemistry, properties of ceramics and other high temperature materials, a number of design concepts and limited reports of actual experimental MHD coal combustor work and results. It is evident that no coal combustor has been tested which fulfills all the essential characteristics required of the ETF combustor. Therefore, considerable engineering judgment must be employed to utilize the pertinent information which does exist and integrate it into an overall concept which provides a reasonable probability for successful operation. Specific design features should be subjected to intensive evaluation testing to determine their viability before one is committed to undertake the final design and fabrication of a component so vital to the successful implementation of commercial MHD power systems as the coal combustor. It is expected that much of the necessary engineering design data will be acquired from testing at the CDIF and elsewhere within the time frame projected for the ETF.

2.8.1 Performance and Tradeoff Analysis

2.8.1.1 Performance Requirements

The basic performance requirements which the combustor must satisfy are those defined by the ETF project general guidelines and ground-rules together with the specific requirements which were established from the work conducted during Task I of the program. The requirements are:

1. Fuel: Primary - Montana sub-bituminous, 5% moisture as fired; secondary - Illinois no. 6 bituminous, 2% moisture as fired. (Both coals pulverized to 70% through 200 mesh.)
2. Oxidizer: Air preheated to 2500°F to 3000°F (max), O₂ + air preheated to 2000 - 2500°F for short term testing.
3. Oxidizer/Fuel: Equivalence Ratio ~ 0.93 nominal
4. Seed: 1% K nominal, injected as solid (K₂SO₄, K₂CO₃, KO₂CH)
5. Pressure: ~6 atm nominal
6. Pressure Drop: ~5%

7. Gas Outlet Temperature: $\sim 2800^{\circ}\text{K}$ (4580°F) nominal
 8. Residence Time: ~ 50 ms
 9. Thermal Input: 200 - 350 MW (total thermal input)
 10. Coolant: Boiler feed water
 11. Slag Removal: 80% to 90% of coal ash content
 12. Electrical Isolation: For 15 - 20 KV above ground potential
 13. Turndown ratio: $\sim 4/1$
- 2.8.1.2 Selection of Combustor Type

One of the major considerations in the design of the combustor was to determine whether all of the requirements could best be satisfied by a single stage or multiple stages. The principal advantage of the multiple stage combustor, cited by advocates of this concept, is a greater degree of control in the removal of slag. In theory, this is accomplished by operating the initial stage with an oxidizer deficient stoichiometry in order to limit the temperature attained in this stage. At this selected temperature level all the major ash constituents have been transformed into the liquid phase, but the temperature is sufficiently low to preclude substantial vaporization of the more volatile slag components. The liquid slag droplets are driven to the wall by imposed cyclonic forces and directed to an area where the liquid slag may be removed from the combustor. The fuel-rich gases and remaining char are burned to completion in a subsequent stage (or stages) where the remaining oxidizer is introduced. Uncertainties still exist related to the chemical and physical behavior of the ash constituents under actual burner operating conditions. In the actual combustor environment, where the particle heating rate is extremely high, with correspondingly high reaction rates, the chemical kinetics are not well known and the species produced cannot be accurately predicted. There is also evidence from experimental and analytical work that combustion in a fuel-rich or reducing atmosphere at temperature levels of $3300 - 3500^{\circ}\text{F}$ actually enhances the volatilization of silica and ferric oxide. Carry-over of these species into the generator channel where they can condense on the channel walls has been demonstrated to have an adverse effect upon channel performance. (6, 7)

On the other hand experimental work supports the concept that a single-stage combustor which is not specie selective in coal mineral matter rejection can provide a slag layer in the channel that enhances electrode and

overall channel performance. The single stage combustor recommended in this report is designed to provide for extremely rapid devolatilization of the coal and char burnout at high temperature in an oxidizing environment. Slag control is based upon short contact time as a two phase flow in high temperature gas so that equilibrium mineral matter vaporization does not occur.

It is again pointed out that very limited information is available of actual coal combustor operating conditions and its important interphasing with the MHD channel and other power system components. Therefore, design data and recommendations presented here must be recognized as preliminary and should be verified or revised as necessary as further information becomes available from experimental work with MHD coal combustion.

Another major advantage of the single stage combustor is minimal bulk heat loss to the combustor walls. The longer residence time implicit in a multi-stage combustor requires a larger total combustor volume and hence more wall surface area for wall heat loss. Rapid burnout of the char and early removal of the mineral matter from the combustion products also decreases the radiant heat transfer from the gas-borne particulates. While no definitive numbers can be applied to the radiant heat flux at this time, preliminary combustor test data recently obtained at AERL show a marked decrease in wall heat losses in that section of the burner where the particulate loading in the gas stream was low.⁽⁸⁾

A multistage combustor also requires that the high temperature oxidizer be carefully apportioned to each of the stages in accordance with the stoichiometry selected for each stage. Because no metering valves are available which can provide this function at the required temperature level, some type of fixed area flow restriction would probably have to be employed to divide the flow. This demands a precise knowledge of the expected pressure drop in the respective stages during operation. Combustion instabilities may also result from pressure feedback through the oxidizer lines to each of the stages. This could be prevented by operating the control orifices at sonic velocity so that downstream pressure disturbances would not be propagated into the supply manifold, but would require additional compressor power.

In summary, at present a single stage combustor is preferred over a two-stage or multi-stage combustor for the following reasons:

1. It is not specie-selective in rejection of slag.
2. Results in relatively low heat loss, because of high reaction rates and relatively low surface area.
3. Produces a higher volatile fraction resulting from higher attainable particle heating rates.
4. Less prone to exhibit combustion instability for a given inlet pressure drop.
5. Has a lower degree of mechanical complexity.

Consequently, the single stage combustor was selected as the preferred configuration for the conceptual design studies.

2.8.2 Combustor Design and Construction

The combustor is designed to operate at a pressure level slightly less than 6 atm with air preheated to 2500 - 3000°F as the oxidant or with a mixture of air plus gaseous oxygen at lower preheat temperatures. Montana Rosebud sub-bituminous coal is the primary fuel and Illinois no. 6 bituminous is the secondary fuel. The analysis of both of these coals is given in Table 2-27. The as-received coal will be crushed and pulverized to 70% through 200 mesh, the standard utility grind. The Montana coal will be dried to a 5% residual moisture while the Illinois coal will have 2% moisture as fired. Adiabatic flame temperature for the Montana coal at 2500°F and 3000°F air preheat temperature is shown in Figure 2-86 as a function of stoichiometric air/fuel ratio. For NO_x emission control, it was decided to operate the combustor at a stoichiometric air/fuel ratio of 0.93.⁽⁶⁾ For the selected ETF design operating conditions the inputs to the combustor are: fuel, 16.13 lb/sec; air, 126.1 lb/sec; seed (as K₂SO₄), 3.19 lb/sec.

In order to assess part load operating characteristics of the ETF, the combustor will have a turn-down capability of ~ 4/1. This is estimated to correspond to ~ 25 - 30% of nominal load which should cover the load range generally encountered by base load power plants.

The principal areas of investigation related to the performance and physical characteristics of the combustor which governed its design are:

TABLE 2-27

COAL AND ASH ANALYSES OF SELECTED COALS

	Montana (Rosebud)	Illinois (No. 6)
<u>Proximate Analysis, Coal as Rec'd, Percent</u>		
Moisture	22.7	8.9
Volatile Matter	29.4	38.0
Fixed Carbon	39.2	41.7
Ash	8.7	11.4
<u>Ultimate Analysis, Percent</u>		
Hydrogen	6.0	5.4
Carbon	52.1	62.4
Nitrogen	.79	1.2
Oxygen	31.5	16.3
Sulfur	0.85	3.3
Heating Value, Wet, Btu/lb	8920	11265
Heating Value, Dry, Btu/lb	11560	12370
Coal Rank	Subbit B	HVCB
<u>Ash Analysis, Percent</u>		
SiO ₂	37.6	41.4 ± 5.4
Al ₂ O ₃	17.3	19.3 ± 6.8
Fe ₂ O ₃	5.1	22.3 ± 6.8
TiO ₂	0.7	0.9
P ₂ O ₅	0.4	0.12
CaO	11.0	5.4 ± 3.3
MgO	4.0	1.7 ± 1.3
Na ₂ O	3.1	0.6 ± 0.2
K ₂ O	0.5	2.1 ± 0.4
SO ₃	17.5	7.5 ± 0.6
Initial Deformation Temp °F	2190 ± 230	1960 ± 70
Softening Temp °F	2230 ± 240	2030 ± 70
Fluid Temp °F	2280 ± 200	2260 ± 200

Figure 2-86 Adiabatic Flame Temperature vs Air/Fuel Equivalence Ratio for Montana Sub-Bituminous Coal

1. Fuel injection and ignition - to provide for extremely high fuel particle heating rate resulting in maximum yield of volatiles optimum ignition and flame stability characteristics and minimum char production.
2. Residence time - sufficient to liberate all of the energy from the fuel and consistent with minimum wall heat losses.
3. Gas uniformity - provide a homogeneous plasma to the generator channel that is spatially and temporally uniform.
4. Slag separation - removal of 80 - 90% of the fuel mineral matter from the combustion products before entry to the MHD channel.
5. Seed injection - to provide uniform plasma electrical conductivity with minimum loss to the slag removed from the gas.
6. Water cooling - use of boiler feed water to cool combustor components to maintain structural integrity of overall assembly, but with minimum heat loss.
7. Electrical isolation - provide isolation of all water, fuel oxidizer and seed feed lines and structural members against the total axial Hall potential developed by the MHD generator.
8. Magnetic forces - selection of materials and mechanical design to avoid high stresses in combustor resulting from close proximity to the superconducting magnet.

Each of these individual subjects will be discussed in greater detail in subsequent sections.

A drawing of the ETF combustor showing its principal design features is presented in Figures 2-87 and 2-88. It consists of a downward firing cylindrical chamber where the combustion and slag separation processes are performed and a side-mounted, square, horizontal duct where the seed is introduced and through which the gases are conveyed to the channel.

The pulverized coal and high temperature primary air are introduced through a number of equally spaced coaxial injectors located near the top of the combustion chamber. A single inlet torroidal shaped manifold with equi-spaced ports supplies the primary air to the coaxial injectors. The molten slag, which is driven to the chamber wall by inertial forces created by the torroidal vortex flow field, flows down the combustor walls

Figure 2-87 Cross-Section Drawing of ETF Coal Combustor

Figure 2-88 Piping Drawing of ETF Coal Combustor

due to gravitational and gas shear forces into the slag port at the bottom of the chamber and is discharged to the slag collection system. The entire combustor and discharge duct are cooled by high pressure boiler feed water through an arrangement of closely spaced welded tube bundles. Because the combustor is operating at the full axial Hall potential developed by the MHD generator, the entire assembly including all the feed lines (primary air, fuel, seed, cooling water) is electrically isolated.

2. 8. 2. 1 Fuel Injection and Ignition

Coaxial fuel/air injectors were selected in order to provide for extremely high heating rates of the pulverized fuel particles to enhance devolatilization, ignition, flame stability and char burnout. Testing conducted at AERL ^(8, 9) has shown that a coaxial injector consisting of a central coal-carrier gas stream surrounded by a high velocity annular flow of high temperature primary air is a very effective means for achieving high particle heating rates and homogeneous fuel/air mixtures. Recently obtained data ^(10, 11, 12) have shown that volatile yields observed from high particle heating rates can be as much as a factor of two greater than that obtained from the usual proximate analysis. This minimizes char formation which reduces the residence time required for complete combustion as well as the radiative heat transport. Tests conducted on a development burner employing this injection scheme have also demonstrated that the preheated air/coal mixture is hypergolic, ⁽⁸⁾ thus eliminating the need for a separate ignition system. Fears that the hypergolic reaction might lead to plugging of the coal injector tip by a build-up of deposits were not realized because the annular flow field was tailored to avoid local separation at the wall injector tip. Location and orientation of the fuel/air injectors in the burner were determined on the basis of providing the optimum flow field with regard to utilization of injection momentum, homogeneity of combustion products and separation of slag. This pointed toward placement of the injectors in the periphery of the combustor directed toward the axial centerline. The small turbulence scale associated with the flow from the individual injectors is superposed on a large scale recirculation field creating a torroidal vortex flow pattern. Radial acceleration of the recirculating flow provides the mechanism for removal of

the slag droplets by driving them to the water cooled combustor wall. Placement of the injectors at an angle of 30° above the horizontal plane was done in the interest of achieving uniform residence time by avoiding a split between the flow from the individual injectors and the main axial core flow. A series of tests to evaluate these design concepts were recently conducted on a model of the combustor in a water flow tunnel. Test conditions were chosen to maintain the same flow Reynolds number as would exist in the combustor. A photo of the flow field within the model is shown in Figure 2-89. This clearly shows the existence of a highly turbulent mixing region created by the injectors and the formation of a torroidal vortex which decays into a reasonably uniform main core flow with little residual swirl as had been predicted by earlier analytical studies.

The number of injectors chosen for the ETF combustor was based on a tradeoff between complexity, flow uniformity, and efficient coal-air mixing. Results from a computer program to determine the volatile yield from a representative Eastern bituminous coal, ground to 75% through 200 mesh, interacting with air preheated to 2000°K , is shown in Figure 2-90 as a function of the number of injectors. Recent tests at AERL have shown that the volatile release at high heating rates is significantly greater for Montana Rosebud coal than Eastern bituminous coal. The computer results are therefore conservative for this application and are used only in a qualitative sense to indicate the direction one should take. The curve shows that multiple injectors produce a significantly higher volatile yield than a single injector. For this design, eight injectors were selected on the basis of minimizing complexity while still providing adequate volatilization and flow uniformity.

Sizing of the injectors was based upon minimizing combustor pressure drop yet providing a sufficiently high velocity differential between the primary air and coal-carrier gas streams to produce effective jet mixing. In order to limit the combustor pressure drop to $\sim 5\%$, which was established as one of the design criteria, the maximum primary air Mach No. should not exceed 0.28. For air at 3000°F this corresponds to a velocity of 778 ft/sec. The coal-carrier gas stream velocity was selected to be 50 ft/sec. This was considered to be sufficiently high to prevent

Figure 2-89 Model of Fuel/Air Injector Configuration in Water Tunnel Test

Figure 2-90 Coal Volatile Yield vs Number of Injectors

Figure 2-90 Coal Volatile Yield vs Number of Injectors

particle fall-out, yet low enough to limit erosion in the coal feed lines. A number of references^(13, 14, 15) giving analytical and experimental results on particle-gas mixing for coaxial turbulent gas streams were reviewed and it appeared that satisfactory mixing would occur with the conditions selected. This was substantiated by experimental combustor tests⁽⁸⁾ where a single coaxial injector operating at a comparable R_N based on the coal injector diameter and jet velocity differential was used. Figure 2-91 shows a sketch of the injector configuration used in the reference combustor tests at AERL. The results from these tests indicate that very rapid particle - gas mixing and particle heating rates were obtained as evidenced by measured particle temperature and gas composition.

The mass ratio of carrier air/coal was selected to be 0.1 based upon available information on the Petrocarb system which was the prescribed coal feed device. With the known volume flow rate and injector velocity the required cross section area for the injectors could be readily calculated. Similarly, the area of the primary air injectors could be determined by taking into account the effective area of the fuel injectors. Based upon these calculations the required diameter of the coal delivery tube was found to be 1.75 in. The entire injector assembly as shown in Figure 2-92 consists of three concentric stainless steel tubes to provide annular cooling water passages in order to maintain a temperature level in the coal injector which will preclude softening or sticking of the coal particles. The injectors are supported and positioned centrally in the air injector passages by a dense, high strength, ceramic spider. The diameter of the primary air injector passage which is an integral part of the ceramic liner surrounding the air inlet manifold is 8.12 inches.

2. 8. 2. 2 Slag Removal

The basic mechanism for the separation of slag from the combustion products is the inertial forces created by the intense recirculating flow field generated near the head end of the combustor by the high velocity primary air injectors. The slag particles or droplets which are formed from the micro scale inclusions of mineral matter in the fuel, are subjected to inertial forces resulting from radial acceleration and aerodynamic drag. An analysis was made to determine the magnitude of these forces for various size droplets in order to find the time required for a droplet to

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Figure 2-91 Single Coaxial Fuel Air Injector Assembly Used in Test Combustor

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Figure 2-92 Drawing of Injector Assembly

impact the combustor wall. In these calculations a constant angular velocity based upon conservation of the injector flow momentum was assumed, and that droplet drag was a function of Reynolds No. The results of this analysis showed that for the range of particle size expected, ash capture level would be of the order of 70%. It is expected that additional ash rejection will occur in the bulk of the combustor from weaker recirculation coupled with turbulent transport to yield a total ash rejection rate in excess of 80%. These results are consistent with data obtained at AERL from tests to determine slagging characteristics in the Mk VI combustor and the Replenishment Test Facility.⁽¹⁶⁾ It is recommended, however, that additional investigations be conducted in this area to provide reliable engineering design information.

The thickness of the slag layer formed on the combustor wall is a function of several parameters. These include the backing wall temperature, local heat flux, slag thermal conductivity, slag viscosity and the gas shear and gravitational forces acting upon the slag. A substantial amount of analytical and experimental work has been performed at AERL to characterize an equilibrium slag layer formed under these conditions.⁽⁷⁾ Reasonable agreement between predicted and measured results has been attained. For the calculated conditions of the ETF combustor it is anticipated that the total thickness of the slag layer will be slightly under 0.5 inch. The temperature of the slag in intimate contact with the combustion gas will be $\sim 2800^{\circ}\text{F}$. At this temperature the slag viscosity is estimated to be 20 poise which will permit it to flow freely under the combined gravitational and gas shear forces.

The build-up of a stable slag layer implies some mechanism for initial adhesion of the slag to the backing wall structure. It has been found that slag will bond very well to ceramic materials but quite poorly to bare metal surfaces. This indicates the desirability of covering the cooling tubes with some type of high temperature refractory material. This would also be beneficial in reducing the combustor heat loss. Unfortunately, however, there appears to be no refractory material available at this time which would stand up reliably to the erosive and corrosive environment which will exist in the combustor. Therefore, in the interest of long term reliability, it was decided not to use refractory materials in the main section

of the combustor. Recent operating experience⁽⁷⁾ has shown, however, that a thin cermet flame spray coating over bare metal provides an excellent surface for slag adhesion. The bond between the cermet and slag has been found to be very tenacious and has stood up well even during severe transient conditions encountered during combustor start-up and shut-down. This proven technology will, therefore, be employed to treat the interior surface of the combustor during the fabrication process to ensure the formation of a stable slag layer during operation.

The slag collected on the walls of the combustor flows into the conical section at the bottom of the burner which interfaces with a pressurized slag collection and disposal system. The conical section will be uncooled but will be lined with 8 inches of insulating refractory to minimize heat loss. It is believed that the refractory material will be suitable for this application because of the low velocity in this region.

Details of the slag collection and disposal system are presented elsewhere in this report (Section 2.10.3). Temperature in the slag collection chamber will be controlled to maintain the partial pressure of the water at a slightly lower value than the partial pressure of water in the burner. This will preclude the flow of water vapor formed by the slag quenching into the combustor with its attendant deleterious effect upon plasma temperature and conductivity.

2.8.2.3 Residence Time

The combustor residence time must be of sufficient duration to permit complete burnout of the fuel yet as short as possible to minimize heat loss from the high temperature combustion products. In an oxidizing atmosphere the burnout rate of coal particles depends predominantly on the rate of devolatilization and the subsequent reaction rate of the volatiles and residual char with reactive species in the gas phase. Because the gas phase reaction of the volatiles is normally much faster than the heterogeneous reaction of the char, the burnout time and consequently the required residence time is minimized if the char fraction is small. As stated earlier, the fraction of volatiles produced is significantly increased by the high particle heating rate resulting from the coaxial fuel/air injection selected for the combustor. In order to determine the devolatilization rate and hence, the required residence time, an AERL developed computer program was employed. This program treats the coal particle size

distribution as a histogram consisting of discrete mass fractions for each size group of a specified radius. It is assumed that the mixing of the coal-carrier stream with the preheated air proceeds in an exponential manner with a time constant equal to the characteristic time.* The rapid devolatilization of coal is modeled by two competing first-order reactions, each describing the thermal degradation of coal to residual char and volatiles. The high temperature reaction is assumed to produce less char (or more volatiles) than the low temperature reaction. In addition, it is assumed that during devolatilization the particles retain their original size and reduce their density. The reaction between the volatiles and the surrounding hot air is assumed to be fast and to occur in a locally uniform manner resulting in instantaneous local gas phase thermochemical equilibrium. Heat transfer to the particles is described by the well documented formulations for small spheres. The output of the computer program gives the volatile yield as a function of elapsed time for each characteristic mixing time.

The coal particle burnout is modeled according to the following assumptions and approximations:

1. All particles are spherical. The diameter of each particle is estimated by approximating the log normal size distribution by a histogram.
2. Each particle reacts with a local nonconvective gaseous environment solely by the mechanism of diffusion of gaseous species towards or away from the surface.
3. Volatiles and gaseous products formed by surface reactions diffuse outwards. The reactive gaseous species diffusing inwards are O_2 , O , CO_2 , H_2O and OH . The heterogeneous surface reactions are

*The mixing region of the preheated air jet and the coal carrier gas stream is assumed to be fan-like with an angle of 5° . The characteristic mixing time is defined as the time taken by the higher velocity primary air to traverse the length of the inner cone.

where T is the temperature of the particle surface

4. Devolatilization occurs throughout the particle; this reduces the density of the particle while maintaining the size. Surface reactions reduce the size while maintaining the density.
5. The reaction products and the volatiles equilibrate with the bulk gases changing the bulk temperature and composition.
6. Each particle has uniform temperature at any instant of time and this temperature is determined by heat transport and reaction heat produced or absorbed at the surface.

The equations for species conservation and energy are solved for the local diffusion model around each particle undergoing simultaneous devolatilization and surface reactions. The solution, in the form of mass removal rate from each particle, is integrated into the rapid devolatilization computer code described earlier. This computer code gives the following quantities as a function of elapsed time: (a) bulk gas phase composition and temperature, (b) chemical and sensible enthalpies of the gas phase, (c) fraction of coal devolatilized, (d) fraction of coal gasified by surface reaction, (e) particle temperatures, (f) particle radii, and (g) particle densities.

Results from these computations for HVA bituminous coal ground to 70% thru 200 mesh and at the conditions indicated is shown in Figure 2-93. Of particular interest are the curves showing the fraction of total fuel burned and the fraction burned of various particle sizes as a function of residence time. This indicates that ~ 95% burnout of the fuel has occurred in 40 ms. The unburned particles are primarily the large diameter particles present in the original feedstock. The effect of coal size distribution was further investigated using a 70% minus 200 mesh histogram as reference G_A and a finer grind, G_B . Figure 2-94 shows the variation of required residence time

Figure 2-93 Predicted Time History of Representative Physical Processes in Coal Combustion Model

Figure 2-94 Effect of Fuel Size Distribution and Stoichiometry on Residence Time Required for Specified Fuel Utilization. (HVA bituminous, 4 atm, 1600°K preheat)

with input stoichiometry for lines of constant fuel utilization, C_R/C_F , for the reference grind, G_A , and a finer grind, G_B . At the fuel/air equivalence ratio selected for the ETF combustor, 1.07, the plot shows that a marked reduction in residence time for 95% fuel utilization is indicated for the finer grind. It also shows that a 3% improvement in fuel utilization can be realized for the finer grind at a residence time of some 5 ms less than the standard grind.

Preliminary data on reaction behavior of Rosebud sub-bituminous coal suggest that pyrolysis may proceed at a higher rate and to higher yield than is the case for a swelling bituminous. Also, the residual char is expected to have a much higher reactive surface area per unit mass. Consequently it is expected to combust more rapidly than an equivalent size distribution of a bituminous coal.

On the basis of these results, 50 ms residence time was selected for the ETF combustor. This figure is conservative and can probably be reduced when more definitive data on the characteristics of Montana coal are obtained. Further analysis should also be conducted on the tradeoff between residence time and particle size distribution as they relate to heat loss and the energy consumed by finer grinding and particle classification. Using the 50 ms residence time, the required burner volume was determined by assuming plug flow and using a mean value of gas temperature and pressure from the thermo-chemical calculations. Actual combustor dimensions were obtained by assigning an L/D of 4.0.

2.8.2.4 Seed Injection

The seed is introduced into the hot combustion gases after the slag has been removed to minimize loss of seed to the slag. Seed in the form of powdered K_2SO_4 , KO_2CH , K_2CO_3 or mixtures thereof is injected through 8 watercooled probes located in the square, horizontal, burner take-off duct. The seed is delivered to the injectors from a Petrocarb feed system using dry air as the carrier in the ratio of ~10 lb seed per lb of air. Because the seed is hygroscopic it is imperative that the air be completely moisture free to prevent the seed from agglomerating and sticking to the feed lines. The design of the injectors is similar to that of the coal injectors described earlier. Proper selection of the seed/carrier injection momentum will be used to achieve uniform distribution of seed in the gas duct and to keep it away from the walls. Injection of the seed transverse

to the flow will provide adequate mixing to form a homogeneous plasma entering the nozzle at the duct exit. Gas residence time in the duct from the seed injection plane is slightly more than 8 ms. Operational experience from a number of MHD combustors indicates that complete seed vaporization and plasma homogeneity can be readily achieved in this period of time.⁽¹⁷⁾

Some loss of seed to the residual slag which will coat the water-cooled walls of the duct is inevitable due to vapor phase seed transport. The magnitude of this loss cannot be readily determined analytically and should be a subject for experimental investigation.

2.8.2.5 Combustor Cooling

The total thermal energy absorbed by the combustors walls from the high temperature combustion products must be transferred to the cooling water and removed from the combustor at essentially the same rate at which it is absorbed in order to maintain thermal equilibrium. The total heat transferred to the walls is the sum of the convective and radiative components. The relative magnitude of each of these components will vary widely throughout the combustor as a result of several rapidly changing parameters such as gas particle loading, gas composition, mean gas temperature, turbulence level, wall friction factor, and others. A number of analytical models may be used to calculate the thermal loading of the combustor but each requires the selection of several assumptions which significantly effects the solution of the problem. In view of this uncertainty we chose to use actual experimental results^(18, 19) obtained from a previous coal combustor development program to estimate the heat loss for the ETF combustor. In these tests, the measured heat flux for a cold wall, non-slagging burner was 2.2×10^5 Btu/ft²/hr, while for a ceramic lined burner, the heat flux was 10^5 Btu/ft²/hr. Because the gas enthalpy from these earlier tests was essentially the same as that for the ETF burner it is valid to assume that the convective heat flux will be virtually the same. The radiant heat flux, however, must be corrected for the differences in geometry between the burners to reflect effective beam length. After this correction was applied, the expected average total heat flux for the ETF burner was found to be $\sim 200,000$ Btu/ft²/hr. This value was compared to computed values obtained from conventional heat transfer calculations in which a range of conditions, representing what were considered to be "best"

and "worst" cases were assumed. These calculations indicated that the estimated heat flux of 200,000 Btu/ft²/hr was a reasonable value and this figure was used to calculate the combustor cooling requirements and heat loss.

For ease of fabrication a combustor configuration consisting of two concentric shells with cooling water flow between the shells was initially considered. This was found to be impractical because of the heavy wall thickness dictated by the 1500 psi cooling water pressure in the boiler feed water circuit.

A tube wall design consisting of relatively small diameter cooling tubes brazed or welded to the inner wall of the combustor outer shell was subsequently investigated. While copper tubes would be the preferred material from heat transfer considerations, it was determined that copper would be unacceptable because of corrosion problems in the steam generator. A 90-10 cupro-nickel material, Alloy 706, and Type 321 stainless steel, both of which were acceptable by the steam generator manufacturer, were analyzed for use in the combustor cooling system. Heat transfer and stress calculations showed that both materials could safely be used but that Alloy 706 provided a higher safety margin. Consequently this material was used in the subsequent more detailed study. This study resulted in a cooling wall configuration consisting of contiguous 1/2" O.D., .065" wall, tubes brazed or welded to the outer shell. Milled slots at the tube ends communicate with circumferential manifolds to permit water flow into and out from the tubes. A spiral tube arrangement was selected to provide cooling for the heat or top of the combustor. This consists of a central inlet manifold with a spiral tube array discharging to a circumferential manifold.

The various inlet and outlet manifolds are interconnected to give a balanced heat load to the entire cooling system and results in a single supply and discharge line which need to be electrically isolated from the high potential at which the combustor is operating.

The following diagram represents the boiler feed water cooled elements of the combustor.

- (1) Coal injection section, 288 tubes, $1/2$ O. D. x .065 wall
131.5 lb/sec, 46 in. length
- (2) Combustor shell, 288 tubes, $1/2$ O. D. x .065 wall, 131.5 lb/sec, 96 in. length
- (3) Seeding duct, 164 tubes, $19/32$ O. D. x .078 wall 146.5 lb/sec, 82 in. length
- (4) Water inlet pipe, spiral, 1 tube, $1/2$ O. D. x .065 wall
.46 lb/sec, 154 in. length
- (5) Dished head, multiple spiral, 32 tubes, $1/2$ O. D. x .065 wall,
14.6 lb/sec, 202 in. length
- (6) Side walls of combustor exit, 48 tubes each side, $1/2$ O. D. x
.065 wall, 14.6 lb/sec, 12 in. length

The cooling tubes in the seeding duct (3) are larger in order to take the total water flow rate with a moderate pressure drop. The lower heat flux in this section at the walls will cause a shift of the largest allowable water cooling tube diameter to a higher value and makes a $19/32$ in. diameter tube permissible.

The interconnecting piping is sized as follows:

The 6" boiler feed water line feeds a 4" x 4" ring manifold, welded on the top flange of the coal injector section with drilled connections to the annulus at the inlet of the vertical cooling tubes. The lower flange of the coal injector section is ducted directly to the top flange of the combustor shell by utilization of a double seal. A lower 4" x 4" ring manifold in the corner of the lower combustor shell and flange collects the water from the

lower annulus. A single 6" or two 4" pipes can connect to a 4" size manifold near the upstream flange of the seeding duct, from which 20 pipes of 1-1/2 in. size run to the annulus in the upstream flange. A similar 4" size manifold collects the cooling water from the downstream flange of the seeding duct with 20 pipes of 1-1/2 size, which are long enough to place this manifold outside of the magnet vessel. The exit of the boiler feed water is again a 6" pipe off the manifold.

Bypassing the cooling water for the coal injector and the combustor shell are:

- a. A 4-ft diameter spiral array (5) of 32 tubes that cool the top of the combustor.
- b. A small single tube spiral (4) for cooling the end cap of a 6-inch schedule XX pipe, which serves as the inlet for the large spiral array.

The large spiral by-pass also cools the two side walls (6) of the combustor exit in series.

A preliminary investigation of the manufacturing problems which might be encountered in fabricating the cooling system has not revealed any major technical problems. The success achieved with similar cooling schemes in modern high performance rocket engines would indicate that the developed technology in this area could profitably be employed in the detailed design and construction of the ETF combustor.

The outer shell of the combustor is essentially a pressure vessel and must be able to withstand the normal operating pressure of ~ 6 atm plus any transient pressure spikes. The combustor is considered a fired pressure vessel and the design requirements as set forth in the ASME Boiler and Pressure Vessel Code were adhered to. Because of its close proximity to the superconducting magnet and the resultant high magnetic forces to which it would be subjected, it was decided to make it from non-magnetic stainless steel, type 304 or 321. This will not add significantly to the cost of the combustor and might indeed be less costly than providing the additional structural supports required to withstand the magnetic forces if it were made from carbon steel.

Using the formulas from the ASME code it was found that 0.5 in. wall thickness for the outer shell provided a wide margin of safety for temperatures as high as 1300°F . Although it is expected that the maximum temperature of the wall will not exceed 400°F , it was decided to use the 1/2 in. thick material.

2. 8. 2. 6 Electrical Isolation

Because the combustor is directly connected to the upstream and of the MHD generator it is subjected to the full axial Hall potential developed by the generator. In the case of the ETF generator this may be as high as 15 - 20 kV. Consequently, it is necessary to provide electrical isolation for all combustor support members, the slag collection system, and the feed lines supplying high temperature combustion air, fuel, seed and cooling water.

The high temperature primary air supply line is electrically isolated from the combustor by a section of insulating material placed between the flanges of the primary air duct and the burner inlet transition section. A drawing of this insulating section is shown in Figure 2-95. As indicated by the drawing it consists of two 2" thick sections of G-10 fiberglass epoxy which has a dielectric strength rating of 400 V/mil. The stand-off capability of the two sections is, therefore, many times the anticipated potential. Between each section of insulator is a high voltage breakdown sensor to detect a fault in the insulator. The sensor is wired into the plant control system which will indicate a fault to the operator who will initiate the appropriate response. The threaded rods used for bolting the flanges together are also made from a dielectric material, G-10 or equivalent.

The four column supports for the combustor will be insulated from the floor by incorporating insulating flanges similar to that described above. The insulating material to be used here will again be a laminated glass epoxy, G-10. Placement of the insulating sections was chosen to be 3 ft above floor level to prevent possible contamination from water or other foreign material on the floor.

Cooling water supply and discharge lines will incorporate insulating sections such as shown in Figure 2-96. The insulation is provided by a section of Vespel, a molded polyamide resin which has excellent dielectric and mechanical properties at the operating temperature level ($\sim 400^{\circ}\text{F}$). The insulator assembly is made up as a flanged joint with the Vespel between the two flanges. Threaded G-10 epoxy rods are used to bolt the flanges together. A Teflon coating is used to shield the Vespel from direct contact with the water to prevent deterioration of the Vespel. The Teflon is subjected only to hydraulic compressive forces and is not required to

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Figure 2-95 Drawing of Electrical Isolation Section for Primary Air Duct

Figure 2-96 Drawing of Electrical Isolation Section for Water Lines.

contribute mechanical strength to the assembly. Tensile stress in the Vespel is calculated to be well below the limit recommended by the manufacturer in this temperature range. Because the coefficient of thermal expansion of both materials is essentially the same, there should be no problems resulting from transient thermal conditions.

Isolation of the combustor from the seed and coal feed systems will be accomplished by placing a 3-foot section of Teflon lined fabric reinforced rubber hose in each of the feed lines. These will be located between the metal seed and coal feed lines and the individual injectors mounted in the combustor. This type of insulator has operated successfully for several thousand hours of test operation in the Mk VI generator with no apparent problems from particle erosion of the hoses.

Some parties have expressed concern over the electrical conductivity of the stream of suspended coal particles entering the burner and the ability to effectively isolate this current path. AERL investigated this problem a number of years ago and conducted a series of experiments in which a high potential (70 kV) was impressed between the ends of an insulated tube conveying a pulverized coal/air stream; a micro-ammeter was used to measure current flow. The coal used in the tests was extracted from the mills of a local utility. The tests were run for a number of solids loadings and stream velocities and in no case was there any measurable current flow through the coal/air stream. A final static test was made with the tube filled with densely packed coal and as before no current was observed.

The slag collection and removal system discussed in Section 2.10.3 must also be electrically isolated. The problems associated with providing effective electrical isolation for the slag system are not underestimated and should be the subject of additional development effort. Electrical conductivity of molten slag as function of temperature and composition as well as slag contaminated water must be determined in order to provide adequate engineering data to permit design of a viable slag collection system.

2.8.3 Instrumentation and Controls

The combustor instrumentation and controls are used to regulate the output of the combustor, measure its performance, and monitor critical parameters to ensure that safe operation within predetermined limits is

maintained during all modes of plant operation. This includes normal start-up, normal operation with response to electrical load changes, normal shutdown, and emergency shutdown. The control system will incorporate all the sequencing, interlocks and monitoring functions with appropriate safety limits required for all operating modes. Design of the control will permit either automatic or manual operation and will provide digital readouts of the significant parameters on the control console. Some of the instrumentation will form an integral part of the overall plant control system while other will serve to provide the necessary data for performing engineering calculations. A description of the integrated plant control system is described in Section 3.0 of this report.

A principal component of the combustor control is a ratio controller which will regulate the mass flow rates of oxidizer (preheated air or oxygen enriched air) and pulverized coal to provide the desired stoichiometry to the combustor. The ratio controller is connected with the integrated plant control system which will vary the combustor firing rate as a function of plant load demand. Trim control of the fuel/air ratio controller is provided by a signal from the gas analyzer located in the diffuser which will adjust the stoichiometry as a function of the CO content of the generator exhaust gas. A schematic drawing of the ratio controller is shown in Figure 2-97.

The seed mass flow rate supplied to the combustor gas discharge duct will also be regulated by the ratio controller to maintain a prescribed quantity of seed flow relative to the oxidizer and fuel flow in order to provide the proper gas conductivity.

Because there will be some variation in seed composition dependent upon the efficiency of the seed reprocessing system, the amount and type of seed added for replenishment, and the amount of impurity in the seed, the potassium content will also vary. Hence, control of the seed flow rate on only a mass or volumetric basis will not be adequate for maintaining the proper gas conductivity level. Therefore the seed flow rate signal from the ratio controller will be biased by a gas conductivity measurement obtained in a section of the MHD generator channel.

At the present time, there is no reliable means available to directly measure gas conductivity on a continuous basis while the generator is delivering power. However, some preliminary work done on the Avco Mk VI

Figure 2-97 Schematic of Combustor Control

MHD generator shows promise that a method for making this measurement can be developed for peg-wall type channels. The conductivity can also be measured by superimposing a high frequency AC signal on the generator DC signal but the scheme has met with only limited success. A conductivity probe can also be employed to measure the conductivity, however, this will provide only an intermittent signal because the residence time of the probe in the plasma must be restricted to less than ~ 0.3 sec. It is reasonable to assume, however, that a reliable means for direct measurement of gas conductivity will be available for incorporation into the ETF control system.

A list of the combustor instrumentation in terms of the quantities to be measured and the type of device used for making the measurement is given in Table 2-28. A discussion of some of the measurements and their significance is given in the ff paragraphs.

A direct measurement of the combustor performance requires a measurement of the temperature of the gas discharged from the combustor. This temperature ($\sim 4750^{\circ}\text{F}$) is well beyond the range that can be measured by conventional high temperature thermocouples and can only be measured by optical techniques.

This type of measurements is satisfactory for experimental or development testing but would be impractical in an on-line operational situation. During the initial performance evaluation test period one of the optical temperature measurement methods can be used to develop a set of combustor performance data which can be used to determine combustor performance in terms of more readily obtainable data such as combustor pressure or heat loss.

Because the combustor is discharging thru a fixed area nozzle the gas temperature can also be inferred from the continuity equation, $\dot{m} = \rho uA$. The total mass flow rate is the sum of the reactants entering the combustor (less the slag withdrawn) and the velocity can be calculated from a total pressure measurement in the seed duct and the static pressure at the nozzle throat.* The major uncertainties in determining the temperature by this method are the lack of knowing the effective throat area to the degree of precision required and the true value of the ratio of specific heat of the

* Analogous to C^* parameter employed in rocket engine performance evaluation.

TABLE 2-28
COMBUSTOR INSTRUMENTATION

Quantity Measured	Instr't Type	Range
I. <u>Mass Flow Rate</u>		
1. Primary Air	Venturi or orifice meter	50 - 150 lb/sec 2000 - 3000°F
2. Oxygen	Venturi or orifice meter	0 - 15 lb/sec 0 - 1500°F
3. Fuel	Δp Sensors	0 - 20 lb/sec total
4. Seed	Δp Sensors	0 - 4 lb/sec total
5. Cooling Water In and Out	Orifice or Ultrasonic Flow Meter	50 - 150 lb/sec total
II. <u>Pressure</u>		
1. Oxidizer Inlet	Pressure Transducer	0 - 150 psia
2. Comb. Discharge	Pressure Transducer	0 - 150 psia
3. Fuel Carrier Gas	Pressure Transducer	0 - 200 psia
4. Seed Carrier Gas	Pressure Transducer	0 - 200 psia
5. Cooling Water In and Out	Pressure Transducer	0 - 1750 psia
III. <u>Temperature</u>		
1. Oxidizer Inlet	Thermocouples	1000 - 3000°F
2. Fuel Carrier Gas	Thermocouples	0 - 300°F
3. Seed Carrier Gas	Thermocouples	0 - 300°F
4. Cooling Water In and Out	Thermocouples	0 - 600°F
5. Component and Structure	Thermocouples	0 - 2500°F
IV. <u>Strain and Vibration</u>		
	Elect'l Strain Gages and Accelerometers	

gas, γ . A small change in the effective throat area caused by a greater or less thick layer of slag than that predicted or in the assumed boundary layer thickness will result in a significant variation in the calculated combustor temperature.

A qualitative measure of the combustor temperature and hence its performance can also be made from the measured heat loss. This involves uncertainties in the thickness of the slag layer deposited on the cooling tubes, the homogeneity of the slag, and the thermal conductivity of the slag. Consequently this method can only be used for a gross qualitative measurement of the combustor performance. Combustion gas analysis (CO , CO_2 , O_2) will be utilized in determining the combustor performance.

From the foregoing discussion it is obvious that no viable means is currently available for accurately measuring combustor performance in a routine operational setting at the temperature level required for the ETF. A qualitative measurement derived from other measured quantities in conjunction with a comprehensive pre-operation performance calibration test may be adequate; however, this is an area where additional analysis and experimentation are required.

Flow rates of the preheated air, oxygen (when used) fuel, and seed will need to be measured to control the combustor and determine its performance. The air and oxygen flow rate will be measured by a sharpedge orifice or venturi meter in a straight section of duct upstream of the combustor. A venturi meter is preferred over an orifice because it will operate with a lower pressure drop, however, a search for such a device to operate at this high temperature was unsuccessful. It is believed that there is no technical barrier to fabricating a venturi from a dense high temperature refractory material and this possibility will be explored. Another possibility is the use of an ultrasonic flow measuring device. This class of instruments which operates on the Doppler shift principle has recently been developed and should be investigated for use in this application.

An alternative to measuring the oxidizer flow rate in the high temperature state is to make this measurement before entry into the preheater. The difficulty here is that there is inevitably an undetermined leakage rate in the preheater system. This can perhaps be evaluated during calibration tests of

the preheater system but will probably not remain constant during the operation lifetime of the preheater.

Control of the combustor air flow rate will be achieved by controlling the rotational speed of the cycle compressor and the setting of the variable stator vanes. As described in Section 2.4.4, flow rates below 50% of design flow will be attained by control of a flow dump valve.

Flow rate of the fuel and seed will be measured by means of Δp readings in each of the individual injector lines. Previous calibration tests will have been conducted to establish flow rate as a function of differential pressure between the Petrocarb feed lines and the receiver. Control of the fuel and seed will be by automatic pressure controllers in each injection line. Measurement and control signals will be integrated with the ratio controller and the overall control system.

Cooling water flow rate thru the combustor cooling system will be measured in both the inlet and outlet water manifolds by means of flat plate orifices or ultrasonic flow meters. The latter are preferred because they present no obstruction to the flow and their output signal can be fed directly into the control system. Measurement of cooling water flow rate is important for calorimetric determination of combustor heat loss and to ensure hydraulic integrity of the coolant system. High and low flow limits will be established and tied into the control system to alert the plant operator of any abnormalities in the cooling circuit.

Pressure will be measured in the combustion air inlet manifold and at various points in the combustor by means of water cooled pressure transducers. The pressure sensing ports in the combustor will be kept free from slag by a high pressure air flow which will be directed thru the ports at all times except when pressure readings are taken. The pressure measurements will be used in the calculation of overall combustor performance.

Temperature will be measured with appropriate thermocouples of the preheated inlet air, cooling water inlet and discharge, and various structural points. The temperature at a number of different depths will also be measured in the refractory liner in the inlet manifold and slag collection cone. High temperature limits will be established for the cooling

water discharge and key structural elements to alert the operator of possible impending difficulties.

Strain and vibration instruments will be installed at critical locations on the combustor and supports to measure these quantities and provide a warning when established limits are reached.

2.9 MHD GENERATOR INLET NOZZLE

2.9.1 Performance and Tradeoff Analysis

The MHD channels under consideration in the present effort operate at subsonic entrance Mach numbers at design operating conditions. Therefore, inlet nozzle design involves only that portion of the plasma flow that is immediately upstream of the throat; namely, upstream of the point of minimum flow area which also coincides with the channel entrance.

The criteria for subsonic nozzle design in a non-slag-carry-over environment is to: (a) make the length short so as to minimize heat transfer losses and (b) provide an ample radius of curvature of the nozzle wall in the vicinity of the throat to avoid flow separation. Usual and successful practice at Avco has been to utilize a wall radius of curvature that is at least equal to half the throat height, as shown on Figure 2-98(a). When the plasma contains slag and one wishes to be certain to coat the entrance region of the MHD channel, as well as the remainder of the channel, with a uniform slag layer, additional requirements exist. A low pressure gradient nozzle (low dp/dx) is used and the nozzle walls are designed so that they themselves will be coated with slag. An example of a type of design used successfully in Avco MHD channel experiments is shown schematically in Figure 2-98(b). The grooves in the water-cooled copper walls are prefilled with castable zirconia ceramic and these provide attachment points for the slag. Even if the grooves were not prefilled, they would eventually be filled by residual slag from the coal combustor. The length and surface area of the slagging nozzle of Figure 2-98(b) is significantly greater than that of the non-slagging type. But, the heat transfer rate (W/cm^2) to the walls of the former is reduced relative to a cold wall nozzle by the presence of a hot slag layer. The factor of reduction is ~ 1.5 for the case of $3000^\circ F$ air preheat. In any event, the total amount of heat lost to the nozzle cooling water is a small fraction ($< 1\%$) of the total plasma thermal power emanating from the ETF coal combustor.

For the ETF, the nozzle has been considered as part of the MHD channel and utilizes the same low-pressure boiler feedwater. This allows an inexpensive nozzle design fabricated of copper.

Figure 2-98 Sketch of Channel Inlet Nozzle Configuration

2. 9. 2 Nozzle Design and Construction

The nozzle for the present effort is designed for slagging operation. A scale diagram of the inlet nozzle is shown in Figure 2-99. The nozzle is of square cross section and is fabricated of three segments of ordinary ETP (electrolytic tough pitch) copper. Each segment is a single piece of copper 4.0 inches thick, for a total nozzle length of 12.0 inches. The motive for using segments is two-fold: a) copper plates for forgings of appropriate size are readily available b) fabrication, assembly, and disassembly are facilitated. Water cooling passages are gun drilled holes of 3/16-inch diameter. These holes are manifolded external to the plates forming the nozzle.

The milled grooves which allow slag deposition were sized according to experimentally verified criteria. Namely, a depth to width ratio of at least one, and an absolute groove width that is greater than 1/16 inch. Too shallow a groove will not allow a groove to remain filled or be filled with slag, because of high shear forces. Simultaneously, the width of a groove must be many times larger than the expected slag-particle diameter of 30-40 μm , if the groove is to remain filled by slag replenishment. Groove dimensions are a width of 1/8-inch by a minimum depth of 1/8-inch. Nickel plating of the gas side surface is specified to enhance abrasion resistance. For the case of 3000^oF air preheat, heat loss and cooling water characteristics are as follows:

Q(total)	=	0.875 MW
Water Flow	=	60 GPM = 30,000 lbs/hr
T (inlet)	=	117 ^o F
T (outlet)	=	227 ^o F

H2143

Figure 2-99 Scale Diagram of Inlet Nozzle

2.10 MHD GENERATOR DIFFUSER AND TRANSITION SECTION

2.10.1 Summary

The primary function of the diffuser and transition section is to decelerate the high velocity gases exiting from the MHD generator with a minimum loss of stagnation pressure. The diffuser must also provide acceptable gas entry conditions (velocity and static pressure) to the steam generator section.

Figure 2-100 summarizes the results of the conceptual design study of the ETF MHD diffuser. All features are considered within present state-of-the-art in technology.

Geometry is simple. A square cross section diverging diffuser having plane walls and a 2.5° half angle is followed by a square cross section constant area transition section. Area ratio of the diffuser is 4.0 and the area ratio of the transition section is 2.0 resulting in an overall area ratio of 8. Overall length of the diffuser/transition section is 15.2 m.

State variables describing the averaged gas flow conditions are listed at the diffuser inlet and exit, and at the exit from the transition section. Gas enters the radiant boiler at 1 atm static pressure and a Mach number of 0.083, or a velocity of 73 m/sec (240 ft/sec).

The diffuser/transition section combination optimizes the pressure recovery factor obtainable for thick generator boundary layers. Table 2-29 summarizes key design features and their basis. Discussion of tradeoffs is presented in Section 2.10.2.

Diffuser walls are metallic and will have provision for slag retention. Cooling is by boiler feedwater. Construction is of steel tubing, similar to boiler tubes.

2.10.2 Performance and Tradeoff Analysis

2.10.2.1 Requirements

The following summarizes diffuser requirements and the rationale behind them:

<u>Item</u>	<u>Diffuser Design Requirement</u>	<u>Comments</u>
1.	Minimize stagnation pressure loss.	Maximize MHD generator performance.
2.	Subsonic inlet velocity.	MHD generator exhaust gas condition.

$p = 0.835 \text{ ATM}$
 $p_o = 1.193 \text{ ATM}$
 $T = 2296^\circ\text{K}$
 $T_o = 2422^\circ\text{K}$
 $h_o = 64 \text{ CAL/GM}$
 $M = 0.790$
 $V = 796 \text{ M/SEC}$
 $B = 0.26$

0.992 ATM
 1.008 ATM
 2364°K
 2370°K
 29.1 CAL/GM
 0.188
 110 M/SEC

1.0 ATM
 1.004 ATM
 2334°K
 2336°K
 10.7 CAL/GM
 0.063
 73 M/SEC
 (240 ft/SEC)

H2508

Figure 2-100 ETF Diffuser and Transition Section

TABLE 2 - 29

ETF DIFFUSER AND TRANSITION SECTION SUMMARY

Component	Variable	Symbol	Value	Units	Comments
Diffuser	Area ratio	A_3/A_2	4.0	--	Peak C_{pr} occurs at less than 4. 8.0 required for velocity reduction to boiler.
	Half angle	θ	2.5	deg.	Half angle. Slightly in excess of Doss's(22) separation limit for thick boundary layers.
	Square cross section, symmetric	--	--	--	Square MHD generator exit. Shorter than 2-D. Can be modified to round corners or octagonal for improved corner boundary layer behavior.
	$\frac{\text{Length}}{\text{Inlet height}} = \frac{\sqrt{A_2/A_1} - 1}{2 \tan \theta}$	$\frac{L}{D}$	11.45	--	Determined by A_2/A_1 , θ .
	Pressure recovery factor	C_{pr}	0.44	--	In accordance with data reported by Doss(22) and Sajben(20) considering thick boundary layer.
Transition Section	Area ratio	A_4/A_3	2.0	--	Maximizes(23) C_{pr} for a sudden enlargement. Achieves 8.0 overall area ratio required for boiler velocity reduction.
	Half angle	θ	0	deg.	Flow is separated up to exit. No advantage to divergence. Preferred for boiler design.
	Square cross section				
	Length/diameter difference	$\frac{L}{D_T - D_D}$	6	--	Maximizes(23) pressure recovery after dump.
	Pressure recovery factor	C_{pr}	0.46	--	Value for 2:1 sudden expansion.
Combined Diffuser and Transition Section	Area ratio	A_4/A_2	8	--	Required for velocity reduction to boiler.
	Pressure recovery factor	C_{pr}	0.46	--	Combined (diffuser and transition section).

<u>Item</u>	<u>Diffuser Design Requirement</u>	<u>Comments</u>
3.	Low exit velocity < 100 m/sec).	Radiant boiler requirement.
4.	1 atm exit static pressure, + 20" H ₂ O, -0" H ₂ O.	Boiler design requirement.
5.	High mechanical reliability.	Operational requirement.
6.	Cooled slag-covered metallic walls.	Capable of working with MHD exhaust gases containing seed and ash impurities.

It should be recognized that projections of the performance of subsonic MHD diffusers with relatively thick boundary layers as have been considered are uncertain because of boundary layer flow complexities. Therefore diffuser design and performance should be revised as necessary as more reliable design data from experimental work becomes available.

2.10.2.2 Aerodynamic Tradeoffs and Limitations

The thick boundary layers exiting from an MHD generator impose serious difficulties in obtaining a high degree of pressure recovery from a subsonic diffuser.

The blockage factor for the reference ETF MHD generator is calculated to be 0.25. The gas inlet velocity to the diffuser of close to 800 m/s is reduced to about 100 m/sec as required by the steam boiler design. This corresponds to an area ratio of about 8:1. Runstadler's data, ^(20,21) (Figures 2-101 and 2-102 show that with a high inlet blockage factor (even at an optimum wall divergence half-angle of, say, 3-4 degrees) an area ratio greater than 4:1 would reduce the diffuser pressure recovery. Any increase in area ratio above the optimum value, even at the optimum divergence angle results in a decrease of C_{pr} .

Considering the above, an area ratio of 4.0 and a half angle of 2.5° were selected for the ETF diffuser. In order to reduce the gas velocity further to the value required by the boiler design, the gas from the diffuser discharges into a constant cross section transition section of twice the area of the diffuser exit. The pressure recovery factor is estimated ⁽²²⁾ at 0.46 for this 2:1 sudden expansion. Length of the transition section was chosen to be six times the difference between the hydraulic diameters of the diffuser exit and the transition section. Reference 22 indicates that this is the plane of maximum pressure recovery.

E5086

Figure 2-101 Diffuser Performance Map, AS = 1.0, M = 1.0, B = 0.12

E5088

Figure 2-102 Maximum Pressure Recovery vs Blockage, $AS = 1.0$, $M = 1.0$

Overall pressure recovery of the combined diffuser/transition section, was calculated to be 0.44.

2.10.2.3 Aerodynamic Design

The overall pressure recovery C_{pr} is defined by

$$C_{pr} = \frac{P_3 - P_1}{P_{01} - P_1} \quad (2-36)$$

The diffuser pressure recovery factor, and the diffuser exit conditions were computed from

$$\frac{P_2}{P_1} = 1 + C_{pD} \left(\frac{P_{01}}{P_1} - 1 \right) \quad (2-37)$$

$$M_2^2 = \frac{-1 + \sqrt{1 + 2(\gamma - 1) M_1^2 \left(1 + \frac{\gamma - 1}{2} M^2 \right) \left(\frac{P_1 A_1}{P_2 A_2} \right)^2 \frac{T_{02}}{T_{01}}}}{\gamma - 1} \quad (2-38)$$

$$\frac{T_{02}}{T_2} = 1 + \frac{\gamma - 1}{2} M_2^2 \quad (2-39)$$

$$\frac{P_{02}}{P_2} = \left(\frac{T_{02}}{T_2} \right)^{\frac{\gamma}{\gamma - 1}} \quad (2-40)$$

T_{02} was obtained from the heat loss estimate described later in Section 2.10.2.4.

The calculated value of $C_{pr} = 0.44$ is reasonable compared to reported data by Idzorek, ⁽²³⁾ Doss ⁽²⁴⁾ and Sajben. ⁽²⁵⁾

2.10.2.4 Diffuser Wall Construction and Cooling

The diffuser walls are water cooled and of metal construction. The walls are considered coated with slag deposited from the gas stream. An initial thin layer of castable refractory is considered used to improve initial slag retention. By cooling the diffuser/transition section with boiler feed-water, the heat transferred is recovered. The diffuser/transition section becomes an extension of the boiler heat transfer surface.

Four different diffuser wall constructions were considered as listed below.

Wall Construction	Comments	
1. Gun drilled copper slabs	Extensive AERL experience. Slag retention with refractory filled grooves.	Several sections (5+) required to limit drill L/D. Copper may lead to boiler corrosion problems.
2. Tube walls	Practical construction for high pressure boiler feed-water.	
3. Platecoil liner inside pressure vessel	Cooled panels can be non-structural with easy removal for maintenance.	Water pressure limited to 400 psig. Not commercially available in copper (Repeat this only here)
4. Plate coil pressure vessel.		Same as above.

The tube wall construction was selected mainly because it is considered most suitable for high pressure boiler feed water cooling and its construction techniques are well known.

Trapezoidal walls of the diffuser present a problem in tube arrangement which does not occur with the rectangular walls of the transition section. The tubes can be installed tangent to each other parallel to the diffuser axis and diverted diagonally through the wall to the outside of the diffuser when a tube intercepts the converging corners of the diffuser.

An alternative scheme is to arrange the diffuser tubes as shown and discussed in Section 2.10.2 which follows. The tubes are tangent at the diffuser inlet, and diverge toward the outlet. A membrane wall connects the diverging tubes. The latter scheme has been adopted for the present conceptual design, principally because manifolding and gas sealing appears easier.

Figure 2-103 shows the estimated cooling heat flux and cooling heat loss for both diffuser and transition section. Diffuser heat flux decreases from 35 W/cm^2 ($1.11 \times 10^5 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$) at the inlet to 8.4 W/cm^2 ($2.66 \times 10^4 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$) at the exit. Transition section heat flux is approximately

Figure 2-103 Diffuser and Transition Section Heat Loss

10.3 W/cm² (3.27×10^4 Btu/hr-ft²-°F). Integrating numerically over the surface areas yield heat losses of 9.5 and 5 MW respectively. All surfaces are assumed to be covered by slag with a surface temperature of 1800°K. It is implied that the slag will attain an equilibrium thickness determined by the viscosity and dynamic forces.

2.10.3 Mechanical Design and Construction

Figure 2-104 shows a conceptual layout study of the ETF MHD diffuser and transition section. Construction is of welded steel tubes. Pending a structural analysis for conformance to the ASME Boiler and Pressure Vessel Code, Section 1, Rules for Construction of Power Boilers, schedule 40 pipe has tentatively been selected for the tube walls of 1 1/4" size pipe, and manifolds of 6" size pipe. The diffuser dimensions shown by Figure 2-100 require 22 tubes, and 60 tubes per wall for the diffuser and transition section respectively. The diffuser tubes are tangent to each other at the inlet and diverge to a spacing of 1 tube diameter at the exit. A 1/2-inch thick triangular membrane or fin spans the gap between tubes to form a so-called membrane wall. Transition section tubes are parallel and tangent to each other.

Tubes join the manifolds by means of loops which allow access for welding, and provide for relative thermal expansion. The diffuser plumbing provides for single pass parallel flow, while the transition section has 3-passes with the 1st and 3rd passes in parallel flow. Tube ends are configured accordingly.

The diffuser and transition section incorporates presently a 1/4" thick outer pressure shell structure. The possibility of eliminating this shell structure and letting the membrane walls serve as pressure vessel instead is planned to be determined in the remaining work. The transition section is stiffened externally by a 48" square waffle structure consisting of 5" deep x 1/2" thick ribs.

2.10.3.1 Loads and Stresses

The pressure load on the diffuser is maximum and directed inwards when the MHD generator operates supersonically in the unloaded condition. A thick boundary layer normal shock system will start approximately at the midpoint of the diffuser. Static pressure upstream of the shock is estimated to be the order of 0.1 atm absolute. To support the tube walls against the

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Figure 2-104 Conceptual Layout of Diffuser and Transition Section (Continued)

maximum 1 atm pressure differential, 8" I 18.4 beams are welded into "picture frames," and spaced at 48" intervals. Maximum beam stress is 17 kpsi, and deflection is 3.9×10^{-2} inch. Maximum tube wall stress in bending, with the picture frame supports at 48" intervals is 12 kpsi. The corners of the diffuser walls are welded. The diffuser as a whole forms a box beam with walls stabilized against buckling by the picture frame stiffness.

2.10.3.2 Thermal Considerations

Tubes are suspended from the stiffening frames by capscrews threaded into bosses welded to the backside of the tubes. Slots permit relative thermal expansion of tubes and frame. In the case the outer pressure shell is eliminated, an alternate suspension is being evaluated. Pin jointed rods will join the tubes to the picture frame. The rods will be designed to accommodate radial and axial thermal expansion of the tube walls by rotating about their point of attachment to the picture frames. Picture frames are outside the thermal insulation blanket, and will remain at room temperature.

Thermal insulation for the diffuser and transition section is applied. Insulation will be field-installed, to avoid damage during shipping and installation, and to facilitate initial assembly. The external mains carrying boiler feedwater will be suitably lagged. Outer surface temperature of the insulation will not exceed a temperature safe for human contact.

Thermal expansion of the MHD generator is estimated to be $1/4$ " and the diffuser $1\ 3/4$ ", measured axially from the generator inlet flange. To minimize thermal stresses, the exit end of the diffuser is allowed to expand freely 2" toward the transition section.

The forward end of the transition section is constrained by a pinned joint. The rear end is suspended from the radiant boiler. The radiant boiler is suspended from its top, and the suspension point for the transition section descends 5" when the radiant boiler warms up from room temperature to operating temperature. This causes the transition section to rotate 2.3° about the pinned joint.

To accommodate the relative motions, flexible corrugated metal expansion joints are provided between the diffuser and transition section, and

transition section and radiant boiler. Protection of the expansion joints from direct exposure to the hot combustion gases are incorporated. The expansion joint is cooled by natural convection of air.

2.10.3.3 Supports

The diffuser rests on two rails. The rails are supported by longitudinal beams, which in turn are supported by two T-shaped cross members which transfer loads on the rails to two hydraulic lifts.

Support structure near the magnet will be made of either non-magnetic material, or low carbon steel braced to withstand the magnetic loads. The 64" spacing between the rails matches the channel/magnet rail spacing and provides adequate lateral stability for the diffuser.

Two 12" O.D. schedule 40 columns support the transition section. T-shaped cross members on top of the columns prevent the transition section from rolling about its long axis. Two pinned joints join longitudinal stiffening beams under the transition section to the forward support. The rearmost column and support provide for safety during installation. The rear end of the transition section is normally suspended from the radiant boiler by a tension link. A 7" clearance between the keel beams of the transition section and the lateral member of the rear support allows the transition section to rotate about the forward pinned support.

2.10.3.4 Materials

Material for construction of the diffuser will be 304 S.S. Picture frames and skin aft of the midsection can be carbon steel. Reasons for selection are:

1. Non-magnetic. High magnetic field near the rear end of the MHD superconducting magnet.
2. Relatively good corrosion resistance.
3. Ability to withstand high temperatures.

The transition section is at some distance from the magnet and will be constructed of low normal carbon steel material used in boiler tubes. Cooling is by boiler feedwater.

Materials considered for the expansion joint are 316 S.S. and Inconel, depending on detailed further heat transfer analysis which will determine the final selection of material for this.

2.10.4 Controls and Instrumentation

Instrumentation is provided to monitor the heat exchange between the MHD gases and the boiler feedwater, and to monitor temperature of critical structural members. A tentative list of instrumentation is given by Table 2-30.

TABLE 2 - 30
LIST OF INSTRUMENTATION FOR DIFFUSER AND TRANSITION SECTION

Component	Device	No.	Measurement	Range
Diffuser and Transition Section	Transducer	6	Gas static pressure	0-20 psia
	Indicator	6	Gas static pressure	0-20 psia
	Data Acquisition Sys. (DAS) channel	6	Gas static pressure	0-20 psia
	Cooled window	2	Optical (sodium line reversal) particle temp measurement. Also wall (slag) surface temp by optical pyrometer.	1000°F-3000°F
Diffuser and Transition Cooling Water	Transducer	2	Water inlet manifold pressure	0-2000 psig
	Indicator	2	Water inlet manifold pressure	0-2000 psig
	DAS channel	2	Water inlet manifold pressure	0-2000 psig
	Sensor (TC)	2	Water inlet temp	40-350°F
		8	Water outlet temp	40-650°F
	Indicator	2	Water inlet temp	40-350°F
		8	Water outlet temp	40-650°F
	DAS channel	10	Water temp	40-650°F
	Flowmeter (ultrasonic, venturi, turbine, etc.)	4	Water inlet flow rate	0-527,000 lbm/hr total (60-300 gpm each)
	Indicators	4	Water inlet flow rate	60-300 gpm each
Structure	DAS channels	4	Water inlet flow rate	60-300 gpm each
	Sensor (TC)	24	Material temp	R. T. to 2000°F
	Indicator	4	Material temp	R. T. to 2000°F
	DAS channels	4	Material temp	R. T. to 2000°F

2.11 HIGH TEMPERATURE AIR PREHEATER SYSTEM

Conceptual designs of two regenerative air preheater systems, each consisting of four heat exchanger units with individual, full capacity external reheat burners have been performed. The two designs are very similar except in the first conceptual design referred to as Case I, the air enters the air preheater system directly from the air compressor at a temperature of about 500°F, whereas in the second conceptual design referred to as Case II, the cycle air is preheated to 1100°F prior to entering the regenerative preheaters.

Based upon the results of these design analyses, the preheater system referred to as Case I was selected as the reference design. The main reason for this is the consideration of reliability. A relatively low temperature at the inlet end of the regenerator is considered the most reliable because it simplifies the matrix support design and it is more in line with present design and operation of refractory type regenerators for steel industry applications (blast furnace stoves). Also, the necessary valving for cycle air inlet and combustion gas exhaust, and for pressurization/depressurization will not be subjected to very high fluid temperatures, which facilitates their design. Furthermore, for Case I, a pressurization/depressurization heat recovery heat exchanger is not necessary for preheating the pressurization air. As stated, although Case I has been selected as the reference design, the design analysis for the alternate design, Case II, is also presented for completeness.

2.11.1 General Design Requirements

The air preheater systems are of the regenerative type with stationary checker brick matrices. The preheaters are separately fired with oil or alternatively with a relatively clean, low Btu fuel gas produced from coal so as to avoid serious operational problems related to seed and ash impurities. The design of the air preheater system is such that the design of all major components can be scaled up to that required for a full scale plant.

The major operational requirements for the preheater system are:

1. Continuous delivery of high temperature preheated air with minimum fluctuation in pressure, temperature and mass flow rate at the inlet to the MHD combustor.
2. Continuous flow of reheat combustion gas from preheater system.

3. High thermal effectiveness including minimum heat loss through vessel, combustion chamber, high temperature ducting and valving.
4. Minimum pressure loss through matrix, valving and high temperature ducting.
5. Minimum leakage between fluids.
6. Low oxidizer contamination by adequate purge of reheat gas prior to switchover.
7. Capability of part load operation at reduced temperature, pressure and flow rates.
8. Controlled startup and shutdown.

2.11.2 Specific Design Parameters

In order to provide a basis for the conceptual design of the regenerative air preheater systems, the parameters listed on Table 2-31 below were specified. These parameters are based on the overall MHD power cycle requirements and practical heater design limits. Following Table 2-31 is a short discussion of these parameters.

TABLE 2-31

SPECIFIC DESIGN PARAMETERS

1. Preheat air flow rate - 126 lb/sec $\pm 1\frac{1}{2}\%$
2. Preheat air pressure - 6 atm $\pm 1\frac{1}{2}\%$
3. Preheat air temperature delivered to the MHD combustion - $3000 \pm 1\frac{1}{2}\%$
4. Inlet cycle air temperature to preheater - Case I - 500°F, Case II - 1100°F
5. Preheat temperature of reheat burner - air, and recycle gas - 1100°F
6. Maximum reheat inlet gas temperature - 3400°F
7. Maximum reheat exit gas temperature - Case I - 900°F, Case II - 1500°F
8. Maximum reheat gas pressure - 1.2 atm
9. Maximum preheat air pressure loss through preheater system - 5% stagnation
10. Maximum oxidizer contamination - 5% of air
11. Maximum leakage between fluids - 5% of air

The stated inlet and outlet preheat cycle air flow rate, pressure and temperatures are defined by the requirements for the basic ETF power cycle as outlined in Section 2.0. For control purposes, the maximum deviations of the air flow rate, pressure, and temperature at the MHD combustor have been set as a practical design limit of $\pm 1\text{-}1/2\%$. These deviations are based upon operating experience with the Avco regenerative air preheater facility, and considered practical for a larger full scale heater facility. During the delivery of the heated air, the exit air temperature from each regenerator will vary and decrease in time. To compensate for this temperature decrease during each cycle, cooler bypass air will be mixed and continuously adjusted to control a constant air stream temperature to the MHD combustor. The refractory insulated ducting from the air preheater to the MHD combustor will also serve to minimize temperature fluctuations of the preheated air. It should also be recognized that the actual variation at the MHD combustion temperature ($\pm 45^{\circ}\text{F}$) will only be about $1/3$ of the variation in the air preheat temperature (or $\pm 15^{\circ}\text{F}$) because the specific heat of the high temperature combustion gas is roughly three times higher than the specific heat of the preheated combustion air.

The specified temperature variation of the preheated air is presently considered adequate. The operation of the MHD generator and downstream bottoming plant has not yet been investigated in sufficient detail to establish the exact criteria for an acceptable variation. The optimum variation can only be established by considering the total power system.

An inlet cycle air temperature of 500°F and a maximum reheat exit gas temperature of 900°F as assumed for Case I do not deviate significantly from operating conditions of present blast furnace stoves. For Case II with an assumed inlet cycle air temperature of 1100°F , the maximum exit temperature of the reheat gas has been limited to 1500°F because of reliability considerations in designing an uncooled, metallic grate and support system for the refractory matrix. An increase of the air inlet temperature results in a higher operating temperature of the grate and its support system. Consequently, thermal expansion and contraction will increase during initial heat up, normal cyclic operation and shutdown. This can cause misalignment of the flow passages and possible fracturing of the refractory. In addition, the design of a metallic grate at higher operating

temperatures becomes increasingly difficult because of considerations to creep, rupture and corrosion as well as availability and cost of high temperature alloys.

The use of a cooled metallic or refractory grate and its support system, leads to specific problems such as additional heat loss for a cooled system and possible structural instability for a refractory system. These specific problems do not appear unsolvable. However, since the overall plant thermal efficiency remain essentially the same for the two different preheater designs considered with air inlet temperatures of about 500°F (Case I) and 1100°F (Case II) respectively, the lower air inlet temperature of about 500°F is the preferred design condition because it is more in line with the operating conditions of present industrial blast furnace stove regenerators.

As previously stated, the preheaters will be fired with oil or a relatively clean low Btu fuel gas produced from coal in order to avoid serious operational problems related to seed and ash impurities. In order to minimize the fuel heat input to the preheater, preheating of combustion air for the preheater fuel is considered. Furthermore, recycling and preheating of combustion gases are considered. A preheat temperature of 1100°F is considered practical for the combustion air and recycle gas. This preheat temperature is governed by considerations to the overall MHD cycle, emission control of nitrogen oxides as well as practical burner operating conditions.

A maximum flame temperature or reheat combustion gas temperature of 3400°F has been specified mainly because this temperature is considered the maximum operating temperature for readily available refractory oxide materials such as alumina or magnesium oxide. For a temperature above 3400°F zirconium oxide would be a likely candidate material. However, experience with this material for MHD applications is limited to small scale experiments in the U. S. S. R. Therefore, air preheat temperatures in excess of 3000°F has not been considered for the ETF.

A maximum reheat gas pressure of 1.2 atm and an air pressure drop of 5% of stagnation were considered so as to keep auxiliary power requirements within reasonable limits.

Prior to heater changeover from a reheat to a blowdown cycle, the heater is considered purged of products of combustion to avoid contamination of the MHD combustion air. A purge cycle corresponding to 5% or less contamination of the air is assumed and considered practical.

Leakage between the air and reheat combustion gas will occur across the seats of the high temperature isolation valves. A maximum leakage of 5% of the air flow is assumed and considered practical.

2.11.3 Basic System Configuration and Mode of Operation Tradeoff Analysis

For the design of a regenerative heat exchanger system, a number of factors have to be taken into consideration. Such factors are listed in Table 2-32.

TABLE 2-32

MAJOR CONSIDERATIONS IN PREHEATER DESIGN

- Size and number of units
- Mode of operation
- Preheat air exit temperature and its fluctuation
- Inlet air temperature
- Maximum combustion gas temperature and its exhaust temperature
- Mass flow rates of air and combustion products and their ratio
- Dome or matrix top temperature
- Matrix support
- Reheat period
- Blowdown period
- Changeover time including purging and switchover valve closing and opening times
- Refractory materials
- Type of checkers
- Operating pressure and pressure losses
- Reheat burner design
- Spare capacity
- Thermal and mechanical stresses of refractory including compressive creep and fatigue from cyclic operation
- Matrix geometry.

Because numerous air heater system designs are possible, a tradeoff study was performed on various system design alternatives. The following basic system alternatives were investigated:

1. Number of heat exchanger units - 3 or 4
2. Type of reheat system - internal or external

3. Mode of burner operation - continuous (central burner system) or cyclic (individual burner system for each heater unit)
4. Mode of heat exchanger operation - series, staggered parallel
5. Spare capacity or redundancy - 0 to 1 unit.

The number of heat exchanger units considered was limited to 3 or 4 for the following reasons. First, a continuous flow of preheated air requires at least two heat exchanger units, one on blowdown and one on reheat. With such a system, the flow of combustion products from the reheat burners is interrupted during heater switchover. Thus, if a continuous flow of reheat combustion products is required in addition to that of preheated air, three heat exchanger units are required. In the event one heat exchanger is shut down for repair, the addition of a fourth unit would be necessary to assure continued flow of preheated air and reheat combustion gases. In the case of a four unit heater system, the reheat burners may be designed either for full system operational capacity when all four units are in operation (and reduced capacity as a three unit system when one is down for repair), or designed to enable full system capacity operation even as a three unit system. As more and more heat exchangers are added to the system, the size of the units may be reduced. However, it is noted that within the blast furnace stove industry, the practice is to design the preheater system with the minimum number of units, going up to ~35 feet in diameter before adding additional units for higher capacities. Based on this experience the number of heater units that were considered were 3 (no spare unit) or 4 (1 spare unit) resulting in heater matrix diameter of 20 ft or less.

Heat exchanger systems with both internal and external combustion chambers were considered. A recent development in the blast furnace stove industry has been the use of external combustion chambers. The main advantage of the recent use of external reheat combustion chambers compared to the previous use of internal combustion chambers relates to the temperature differences between the reheat combustion chamber and the checkerwork chamber. For the case of an internal combustion chamber, the hottest part of the combustion chamber is directly opposed to the coldest part of the checkerwork matrix chamber. Due to the temperature difference across the refractory partition which separates the combustion chamber

and matrix chamber, and the periodic temperature fluctuations which cause constant expansion and contraction of the refractory material, fracturing of the refractory partition has been experienced resulting in short-circuiting between the two chambers. In case of an external combustion chamber there is no common refractory wall between the combustion chamber and matrix chamber so that problems with short-circuiting is avoided. In addition, the temperature distribution within the combustion chamber is more uniform and therefore less likely to lead to detrimental thermal stress in the refractory insulation.

The use of individual reheat burners systems for each heat exchanger unit (cycling on and off during each cycle) or alternatively a central reheat burner system with isolation/distribution valves (continuously operating) were considered. Figure 2-105 shows typical plan layouts of each of these reheat burner systems including necessary valving.

Each of these designs has important advantages and disadvantages. For a system having continuous burner operation, problems related to unstable burner ignition and thermal cyclic stresses of the combustion chamber insulation are minimized. In addition, since the burner is always operating, a single burner control system can be used. A disadvantage of this system is that large valves must be used for isolation and distribution of the combustion gas to each heat exchanger. These valves must be designed for reliable operation at the highest operating temperature and pressure in the system and for relatively low heat loss. Although this appears feasible it imposes an additional reliability problem. Currently, to our knowledge, there are no known commercial blast furnace stove systems in operation in the United States or Europe with a continuous, single central reheat burner system. (26, 27)

For a heater system with individual burners for each heater unit low temperature valving may be used at the inlet to the individual burners. This would minimize valve heat losses and improve valve operation reliability. However, a more complicated burner control system is required. Although the burners must be cycled for operation of the heaters, this cyclic burner operation is standard practice for blast furnace stove heaters. Unstable burner operation is not considered a problem because of burner

Figure 2-105 a) Continuous Central Reheat Combustion System
b) Individual Reheat Combustion System

cycling. This cyclic mode of burner operation has also been successfully demonstrated in experimental operation of separately fired high temperature air preheaters for MHD applications.

The basic heater systems were evaluated for both series and staggered parallel modes of operation. For normal operation with four heat exchanger units in operation, two cycle sequences utilizing all four units were analyzed. The first cycle sequence shown in Figure 2-106 referred to as series operation has 1 unit on blowdown, 2 units on reheat and 1 unit switching at any point in time. In this mode, the reheat burners operate at only half their maximum design capacity and thus have extra reheat system capacity.

The second cycle sequence shown in Figure 2-107 has 2 units on blowdown, 1 unit on reheat and 1 unit switching at any point in time. In this mode referred to as staggered parallel operation, two heat exchangers are simultaneously supplying preheated air. By staggering the heat exchanger operation in time, one heater will be supplying relatively hot air while the other relatively cool air. Ideally, mixing the two flows should provide for a relatively constant preheated air temperature and then only a small amount of cooler bypass will be required to regulate the final temperature. The main advantage in this mode of operation is that a higher preheat air temperatures are possible than in series operation for the same total air and combustion gas flow rates.

For a three heat exchanger system, or in the event one heat exchanger must be shutdown for repairs in a four heater exchanger system, the systems were analyzed for operation in a series mode with 1 unit on blowdown, 1 unit on reheat and 1 unit switching as shown in Figure 2-108.

Results of the tradeoff analysis indicated the selection of a four unit heat exchanger system with individual, full capacity external reheat burners was the preferred preheater system. This system offers maximum operating flexibility and reliability. The main reasons for this selection are:

1. Spare capacity available with four heater units versus three heater units.
2. Higher reliability with individual reheat burners because high temperature shutoff valves between combustion chambers and matrix chambers can be eliminated.

Figure 2-106 Series Operation. 1 Blowdown - 2 Reheat - 1 Switching

Figure 2-107 Staggered Parallel Operation. 2 Blowdown - 1 Reheat - 1 Switching

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Figure 2-108 Series Operation. 1 Blowdown - 1 Reheat - 1 Switching

3. Increased operational flexibility resulting from the use of individual reheat burners.

2.11.4 Detailed System Description

Two conceptual designs of separately fired regenerative air preheaters have been developed. These are shown in the elevation and plan views in Figures 2-109 and 2-110, respectively. Shown in Figures 2-111 and 2-112 are the system flow diagrams, and in Tables 2-33 and 2-34 summaries of the air preheater design data. Both air preheater systems are designed to preheat 126 lbm/sec of air at 6 atm to a maximum temperature of 3000°F. However, referring to the flow diagrams in Figures 2-111 and 2-112, for the first design (Case I) the cycle air enters the preheater systems directly from the compressor at 500°F while in the second design (Case II), the air is preheated to 1100°F in a separate recuperative heat exchanger prior to entering the preheater system.

Both systems consist of four heat exchanger units. During normal operation, the systems would operate with all four units in a staggered-parallel mode as illustrated in Figure 2-107. In the event one unit would be down for repairs, full capacity would still be available while operating in a series mode with three units as shown in Figure 2-108. After the cycle air is preheated in its respective heat exchanger unit or units, the air is discharged into a common manifold and ducted to the MHD burner. During the delivery of the preheated air, the temperature will vary and decrease in time. To compensate for this temperature reduction during each cycle, cooler bypass air is mixed with the preheated air through the mixer/bypass duct to keep the air stream temperature constant to the MHD combustor. A bypass in the high temperature air outlet manifold to the exhaust stack is also provided for startup, individual operation of the preheater system from the rest of the plant and for emergency dump.

During heater switchover, the heat exchangers are depressurized prior to going on gas reheat and pressurized prior to going on air blowdown. In this way equal pressure is obtained across the valves so they are not required to operate against high differential pressures. A further important reason is to minimize fluctuations in the pressures and mass flow rates of the preheat cycle air and reheat gas streams. Because the

Figure 2-109 Case I & II - Typical Regenerative Air Heater, Partial Cross Section

Figure 2-110 System Layout, Plan View Case I and Case II

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Figure 2-111 Preheat System Flow Diagram Case I

Figure 2-112 Preheater System Flow Diagram Case II (Alternate Design)

TABLE 2-33

ETF AIR PREHEATER DESIGN DATA
SERIES OPERATION

(1 Blowdown, 1 Reheat, 1 Switching)

		3000°F AIR PREHEAT				2500°F AIR PREHEAT	
		CASE I (Reference Design)			CASE II (Alternate Design)	CASE I	CASE II
Matrix Geometry							
Hole Diameter	in	1	1 1/4 (alt.)	1 1/2 (alt.)	1	1	1
Hole-to-Hole Spacing/Hole Diameter (S/D)		1.5	1.45	1.42	1.5	1.5	1.5
Geometric Porosity	%	40	43	45	40	40	40
Matrix Height	ft	85	85	85	65	85	65
Matrix Diameter	ft	15	15	15	15	15	15
Matrix Material							
Composition		Al ₂ O ₃	Al ₂ O ₃	Al ₂ O ₃	Al ₂ O ₃	Al ₂ O ₃	Al ₂ O ₃
Porosity	%	18-21	18-21	18-21	18-21	18-21	18-21
Operating Conditions							
Blowdown/Reheat Cycle Period	min	20/20	20/20	20/20	20/20	20/20	20/20
Press./Depress./Purge Time	min	4/4/.25	4/4/.25	4/4/.25	4/4/.25	4/4/.25	4/4/.25
Blowdown/Reheat Specific Mass Flow Rate	lbm/sec-ft ² (bed area)	.71/.59	.71/.59	.71/.59	.71/.59	.71/.59	.71/.59
Reheat Combustion Gas Inlet Temperature	°F	3400	3400	3400	3400	2900	2900
Reheat Combustion Gas Outlet Temperature	°F	662-841	718-901	778-963	1261-1452	644-795	1212-1340
Air Inlet Temperature	°F	500	500	500	1100	500	1100
Air Outlet Temperature	°F	3245-3073	3182-3000	3119-2932	3252-3074	2771-2635	2765-2614
Air Outlet Temperature Drop	°F	172	182	187	178	136	151
Reheat Combustion Gas Inlet Pressure	atm	1.2	1.2	1.2	1.2	1.2	1.2
Air Inlet Pressure	atm	6.0	6.0	6.0	6.0	6.0	6.0
Reheat Combustion Gas Pressure Drop Across Matrix	atm	.050	.035	.027	.047	.047	.043
Air Pressure Drop Across Matrix		.011	.008	.006	.010	.010	.009
*Maximum Web Stress							
Material Temperature							
1000°F	lb/in ²	420	559	678	426	360	339
1500°F	lb/in ²	718	954	1157	727	613	578
2000°F	lb/in ²	721	958	1161	730	616	580
2500°F	lb/in ²	616	819	993	624	525	495
3000°F	lb/in ²	252	335	406	255	216	203

*Based on simplified model and using Eq (2-42)

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TABLE 2-34

ETF AIR PREHEATER DESIGN DATA
STAGGERED PARALLEL OPERATION
(2 Blowdown, 1 Reheat, 1 Switching)

		3000°F AIR PREHEAT		2500°F AIR PREHEAT	
		CASE I (Reference Design)	CASE II (Alternate Design)	CASE I	CASE II
<u>Matrix Geometry</u>					
Hole Diameter	in.	1	1	1	1
Hole-to-Hole Spacing/Hole Diameter (S/D)	-	1.5	1.5	1.5	1.5
Geometric Porosity	%	40	40	40	40
Matrix Height	ft	85	65	85	65
Matrix Diameter	ft	15	15	15	15
<u>Matrix Material</u>					
Composition		Al ₂ O ₃	Al ₂ O ₃	Al ₂ O ₃	Al ₂ O ₃
Porosity	%	18-21	18-21	18-21	18-21
<u>Operating Conditions</u>					
Blowdown/Reheat Cycle Period	min	40/20	40/20	40/20	40/20
Press./Depress./Purge Time	min	4/4/.25	4/4/.25	4/4/.25	4/4/.25
Blowdown/Reheat Specific Mass Flow Rate	lbm/sec-ft ² (bed area)	.355/.59	.355/.59	.355/.59	.355/.59
Reheat Combustion Gas Inlet Temperature	°F	3400	3400	2900	2900
Reheat Combustion Gas Outlet Temperature	°F	648-827	1247-1438	633-787	1201-1328
Air Inlet Temperature	°F	500	1100	500	1100
Air Outlet Temperature	°F	3259-3087	3265-3087	2784-2649	2775-2622
Air Outlet Temperature Drop	°F	172	178	135	153
Reheat Combustion Gas Inlet Pressure	atm	1.2	1.2	1.2	1.2
Air Inlet Pressure	atm	6.0	6.0	6.0	6
Reheat Combustion Gas Pressure Drop Across Matrix	atm	.050	.047	.047	.043
Air Pressure Drop Across Matrix		.0028	.0025	.0025	.0023
<u>*Maximum Web Stress</u>					
Material Temperature					
1000°F	lb/in ²	210	213	180	170
1500°F	lb/in ²	359	364	307	289
2000°F	lb/in ²	361	365	308	290
2500°F	lb/in ²	308	312	263	248
3000°F	lb/in ²	126	128	108	101

*Based on simplified model and using Eq. (2-42).

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pressurization air flow is intermittent, a smaller separate air compressor and air storage accumulator is used rather than adding capacity to the main system air compressor. The compressor continuously charges the accumulator which is blown down intermittently as required for pressurization of the heat exchanger vessels. The use of low temperature air for pressurizing the vessels may be damaging to the matrix. Therefore, for Case II where the temperature at the base of the matrix is relatively high, regenerative heat recovery heat exchanger is added to the system. This recovers the heat energy which would normally be lost during depressurization of a heater unit for preheating the pressurization air during the next cycle. In addition, preheating the pressurization air reduces the thermal shock on the refractories at the bottom. The pressurization system is sized for pressurizing the heat exchangers to 6 atm pressure in ~4 min.

Each heat exchanger unit is fired with its own reheat burner system which cycles on and off during each heater cycle. The reheat systems are designed for dual fuel operation for burning of either No. 2 fuel or low Btu gas. Although operation will be with oil firing at first, the dual fuel system was selected because the incremental costs of this compared to oil firing only was insignificant with respect to the total preheater system costs. Furthermore, future preheater system modifications would not be necessary for future operation with low Btu gas. The reheat burners are designed to withstand a pressure of 6 atm during the air blowdown period. During the reheat period, the burners are designed to operate at 1.2 atm with preheated air and recycle gas temperatures up to 1100°F. The burners are rated at 340 M Btu/hr thermal input from the fuel. They are designed to operate with a turndown ratio of 7:1 with oil, 10:1 with gas and with up to 150 - 200%.

Additional preheater system component descriptions and operating conditions are found in Section 2.11.6.

2.11.5 System Heat Losses

The total system heat losses for Case I and Case II are listed in Table 2-35 for both staggered parallel and series operation. The heat losses are broken down into two main groups, direct heat losses which are attributable directly to the air heater system and indirect heat losses which

TABLE 2-35

TOTAL SYSTEM HEAT LOSSES 3000°F AIR PREHEAT

	CASE I - Reference Design (Low Temp, 500°F Air Inlet)		CASE II - Alternate Design (Int. Temp, 1100°F Air Inlet)	
<u>Number of Operating Heater Units</u>	4	3	4	3
<u>Mode of Operation</u>	<u>Staggered - Parallel</u>	<u>Series</u>	<u>Staggered - Parallel</u>	<u>Series</u>
	2 blowdown 1 reheat 1 switching	1 blowdown 1 reheat 1 switching	2 blowdown 1 reheat 1 switching	1 blowdown 1 reheat 1 switching
<u>Direct Heat Loss</u>	10 ⁶ Btu/hr	10 ⁶ Btu/hr	10 ⁶ Btu/hr	10 ⁶ Btu/hr
Checkerwork Chamber	12.92	9.69	12.68	9.51
Combustion Chamber Plus Crossover Pipe	12.29	9.22	12.29	9.22
<u>Valves</u>				
High Temperature Air Outlet	4.66	4.82	4.66	4.82
MHD Burner Shutoff	4.82	4.82	4.82	4.82
<u>Piping</u>				
High Temperature Air Outlet	2.84	2.13	2.84	2.13
High Temperature Air to MHD Burner	<u>.44</u>	<u>.44</u>	<u>.44</u>	<u>.44</u>
Total Direct Heat Loss	37.97	31.12	37.73	30.94
<u>Indirect Heat Loss</u>				
Combustion Gas Exit	.29	.22	1.11	.83
Cycle Air Inlet	.26	.20	.22	.17
Burner Air Inlet	.22	.17	.22	.17
Fuel Inlet (Gas)	.14	.11	.14	.11
Recycle Gas Inlet	.12	.09	.12	.09
Miscellaneous	<u>.03</u>	<u>.03</u>	<u>.84</u>	<u>.84</u>
Total Indirect Heat Loss	1.06	.82	2.65	2.21

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occur in the ducting leading to and from the heater system. As seen, the major difference in heat loss between Case I and I are the indirect losses. These are higher for Case II due to the higher fluid temperature in this ducting.

2.11.6 Component Description and Engineering Analyses

2.11.6.1 Heat Exchanger Matrix Design and Operating Data

The preliminary system design tradeoff analysis in Section 2.11.3 resulted in the preferred selection of a four unit heat exchanger system with individual reheat burners for each heater unit. For this preferred system, arrangement detailed design analyses were made on the matrices for both the low (Case I) and intermediate (Case II) inlet air temperatures considered.

2.11.6.1.1 Method of Analysis

The designs of the heat exchanger matrices were based on analysis of the following:

1. Thermal performance
2. Pressure drop
3. Thermal and mechanical stresses
4. Material requirements

1. Thermal Performance

The thermodynamics of a regenerative heat exchanger is well developed⁽²⁸⁾ and can be described by the following differential equations which express the flow of heat within the matrix and fluids:

$$MC \frac{\partial T}{\partial t} = h_o A (T_g - T) \quad (2-41)$$

$$mc_p \frac{\partial T_g}{\partial t} = - \dot{M} L c_p \frac{\partial T_g}{\partial z} - h_o A (T_g - T) \quad (2-42)$$

where:

A	=	Heating surface area in the matrix, ft ²
C	=	Specific heat of the brick, Btu/lb/°F
L	=	Length, or height of the matrix, ft
M	=	Mass of brick in the matrix, lb
$\dot{M}$	=	Mass flow rate of the fluid, lb/hr
T	=	Checker brick temperature, °F
T _g	=	Heating or cooling fluid temperature, °F
c _p	=	Specific heat of the fluid, Btu/lb/°F
h _o	=	Overall heat-transfer coefficient, Btu/hr/ft ² /°F
m	=	Mass of fluid in the heater, lb
t	=	Time, hr
z	=	Vertical distance in the heater, ft

The complexity of these equations precludes a simple analytical solution and therefore a digital computer program developed by AERL was used to perform the thermal and fluid dynamic analysis of various designs. This computer program describes the performance of the heat exchanger based on a number of simplifying assumptions which have been verified experimentally in the AERL air heater facility and by others. ⁽²⁹⁾ The following assumptions have been made.

- a) Infinite conductivity in the matrix checkers in the direction normal to the flow.
- b) Zero conductivity in the matrix in the flow direction.
- c) Uniform flow distribution of the fluid through the matrix.
- d) Negligible heat losses to the surrounding environment.
- e) Negligible heat capacity of the fluids relative to the matrix ($m \cdot c_p \ll M \cdot C$).
- f) Constant fluid and material properties.
- g) Uniform heat transfer coefficient of each fluid.
- h) The operation of the heat exchangers is periodic.

Assumption a

The differential equations describing the performance of the regenerative heat exchangers are simplified if it is assumed that a zero gradient exists in the checker matrix normal to the direction of flow. This would be true only if an infinite thermal conductivity exists in the material normal to the flow. Nevertheless, the finite conductivity of the material can be taken into account by replacing the heat transfer coefficient with a corrected one, based on the actual thermal conductivity and thickness of the brick as described by Hausen. ⁽³⁰⁾

Assumption b

The assumption of zero conductivity in the matrix in the flow direction is based on the fact that for checker brick matrices with long flow paths and made of materials of relatively low thermal conductivity such as refractories, the axial heat transfer through the matrix material is negligibly small compared to the radial heat transfer to and from the matrix. This acts as though the thermal conductivity of the solid in the flow direction is zero.

Assumption c

Besides the mathematical problem of solving the differential equations for heat exchange in the matrix, an additional problem is that in most cases the flow distribution through the matrix is not known. However, investigation by others⁽³²⁾ in the flow patterns through the checkerwork of internal and external combustion chamber stoves have shown that by proper design, a uniform flow distribution may be assured. Based on this conclusion, the assumption of uniform flow is reasonable for properly designed units.

Assumption d

In the design of the heat exchanger vessel, adequate insulation must be provided so as to avoid over-temperature of the vessel wall, and minimize heat loss to the environment. In general, this heat loss is very small compared to the total heat transferred and may be assumed to be zero.

Assumption e

For the system being analyzed, the heat capacity of the fluids (air and combustion products) are negligible compared to that of the solid material.

Assumption f and g

The computational analysis is greatly simplified by assuming constant fluid and material properties and uniform heat transfer coefficients for each fluid. These assumptions have been found to give good agreement between theoretical and experimental data from the AERL preheater facility when these properties are based on their respective mean axial temperature.

Assumption h

This analysis assumes that if the heat exchanger system has been cycling through a repeated number of heating and cooling cycles until operation is obtained in which the temperature at any place in the fluid stream or matrix is the same as it was a full cycle earlier. Since regenerative heat exchangers are generally operated for very long periods of time before being shut down, cyclic equilibrium is generally achieved.

In Eqs. (2-41) and (2-42), heat transfer from the fluids to the matrix occurs in three ways: conduction, convection and radiation. During the reheat period most of the heat is transferred from the gases to the matrix by convection and radiation, and during the air blowdown period from the

matrix to the air by convection. During each of these periods, the heat is absorbed and given up in the matrix by conduction which takes place in the wall of the checkers. To calculate the heat transfer between the checker bricks and fluids, the overall heat transfer coefficient, h_o , in Eqs. (2-41) and (2-42) is expressed as

$$\frac{1}{h_o} = \frac{1}{h_c + h_r} + \frac{y_o}{3k_s} G \quad (2-43)$$

where

- h_c = convective heat transfer coefficient
- h_r = radiative heat transfer coefficient
- y_o = equivalent half wall thickness of checker flue
- k_s = thermal conductivity of checker material
- G = correction factor to thermal resistance of checker material

The convective heat transfer coefficient, h_c , used in the analysis is based on the standard turbulent heat transfer coefficient correlation for flow through a smooth pipe (Ref. 31)

$$h_c = 0.023 \frac{k}{d} Re^{0.8} Pr^{0.33} \quad (2-44)$$

where

- k = thermal conductivity of fluid
- d = hydraulic diameter of checker brick flue
- Re = Reynolds number
- Pr = Prandtl number

The numerical values calculated from this equation agree well with that found from imperial relations of others, ^(32, 33) and as determined from measurements in the AERL facility and blast furnace stoves. If the checker brick flow passages are offset or tapered, the actual convective heat transfer coefficient will be greater than that calculated due to the increased turbulence, and thus the calculated value will be on the conservative side.

In old blast furnace stove designs where flue diameters are relatively large (2 to 4 inches) and gas velocities relatively low (5 std. ft per sec) a large percent of the heat transfer is by radiation. Current state-of-the-art designs of blast furnace stoves have increased gas velocities up to 20 std. ft per sec ⁽³²⁾ and decrease flue diameter to 1.18 inches, ⁽²⁷⁾ and

thus shifted the influence of heat transfer by radiation to convection. For the matrix designs in this study where hole diameters down to 1 inch and gas velocities up to 25 std. ft per sec are being considered, the influence of radiation has been further reduced to a level where it is assumed to be zero, making the results of the analysis on the conservative side.

As previously mentioned, the differential equations are highly simplified if it is assumed that a zero gradient exists in the checker brick normal to the flow, which would prevail only if the thermal conductivity were infinite. To take into account the actual finite conductivity of the matrix material, the overall heat transfer coefficient is corrected by the value $y_o G / 3k_w$ which accounts for the thermal conductivity and thickness of the flue wall where the factor G corrects this thermal resistive value for the fact that the temperature gradient in the flue wall decreases toward the center. Hausen⁽³⁰⁾ calculated this correction factor for plane walls, solid cylinders and spheres and Butterfield, et al.⁽³⁴⁾ calculated this correction factor for hollow cylinders and square flues.

2. Pressure Drop

One of the important factors of the economic of the heater system operation is the pressure drop across the checker brick matrix because as this pressure drop increases, the auxiliary power requirement also increases. The pressure drop across the checker brick matrix is expressed by the equation:

$$\Delta P = \frac{fL}{d} \frac{\rho V^2}{2g_c} \quad (2-45)$$

where

- f = friction factor
- L = height of matrix
- d = hydraulic diameter of flow passages
- ρ = density of fluid
- V = velocity of gas in flow passages

This pressure drop can be easily calculated once the friction factor is determined. The friction factor depends on a number of factors including the roughness of the walls over which the fluids flow, the Reynolds number, the degree of misalignment, and the amount of dirt build up in the checker flues.

For high efficiency checkers, the probability of misalignment is greater due to the many small flues. Also for the same contamination build up per flue, the percentage of blocked area will be greater with high efficiency checkers over standard designs. Nevertheless, due to the large surface area in high efficiency checkers over standard ones, for the same total level of contamination per heat exchanger, the percent flue area affected in high efficiency checkers is generally less than or equal to the percent affected over standard designs.

Friction factors in blast furnace stoves have been studied and measured experimentally. Work by Rehms⁽³⁵⁾ has shown that the friction factor can more than double due to poor alignment of the checkers. Deterioration of the checkers with age can also cause large changes in the friction factors. A turbulent flow friction factor for a "perfect" flue of 0.05 is used in this analysis. This factor has correlated well with experimental data obtained from the AERL preheater facility and that found by Rehms for a "perfect" flue. For sizing of equipment, the pressure drop calculated for a "perfect flue" is multiplied by two to take into account misalignment of the checker flues, dirt build up and general degradation over time.

3. Thermal and Mechanical Stress Analysis

The design of the checker brick matrix involves analysis of thermal and mechanical stresses of the matrix. The thermal stresses are caused by the flow of heat into and out of the matrix elements during cyclic operation, by the axial and radial temperature gradients and by thermal shock. The mechanical stresses are compressive creep stresses due to the weight of the matrix, and any stresses created by thermal expansion.

There are basically two design philosophies for establishing stress limitations of refractory materials for preheaters. One philosophy is based on thermoelastic theory. Material properties are selected to avoid initiation of fracture from stress. In general, this requires materials which have high values of tensile strength, thermal conductivity, and thermal diffusivity combined with low values of thermal expansion, modulus of elasticity and Poisson's ratio.

The other philosophy for fracture resistance permits crack initiation and considers crack propagation based on fracture energy considerations.

This design philosophy is applied to brittle and more porous ceramics. In this case, the properties of the material should include low tensile strength, high modulus of elasticity and high Poisson's ratio.

This latter theory for establishing stress limitations applicable to brittle refractories has in recent years been advanced by Hasselman,⁽³⁶⁾ but practical use of refractories still relies much on experience.

The maximum thermal stresses related to the transfer of heat into and out of the matrix material, are tensile stresses occurring at the matrix hole surface upon cooling. To facilitate thermal stress design calculations, the checker brick was treated as a series of parallel, infinitely long cylindrical tubes as shown in Figure 2-113. Assuming only radial temperature gradients the circumferential (σ_θ) and axial (σ_z) stress at the hole surface are equal and shown by:

$$\sigma_\theta = \sigma_z = \frac{n \cdot E}{1 - \nu} (T_m - T_s) \quad (2-46)$$

where

- n = thermal expansion coefficient
- E = elastic modulus
- ν = Poisson ratio
- T_m = mean temperature of solid
- T_s = surface temperature at material/fluid interface

The relationship between the mean and gas surface temperature is given in the expressions:

$$T_m - T_s = S_1 \frac{h_o d}{k_s} (T_s - T_g) \quad (2-47)$$

$$T_m - T_s = S_2 \frac{d^2 c_p}{k_s} \frac{\dot{M}}{A} \frac{dT_g}{dz} \quad (2-48)$$

where

- A = matrix cross section area
- $\dot{M}$ = total fluid flow rate
- $R_a = r_a / r_o = 1.05 S/d$
- S = hole to hole spacing

$$S_1 = \text{shape factor} = \frac{R_a^4 (\ln R_a - 3/4) + R_a^2 - 1/4}{2(R_a^2 - 1)^2}$$

Figure 2-113 Checker Brick Cross Section Showing Assumed Cylinders Arranged in an Equilateral Triangular Array

$$S_2 = \text{shape factor} = R_a^2 \left[\frac{R_a^2 (\ln R_a - 3/4) + R_a^2 - 1/4}{8(R_a^2 - 1)^2} \right]$$

T_g = fluid temperature

c_p = specific heat of gas

d = hole diameter

h_o = heat transfer coefficient

k_s = thermal conductivity of solid

r_a = assumed radius of adiabatic circle

r_o = hole radius = $d/2$

$\frac{dT_g}{dz}$ = axial matrix temperature gradient

The above equations show that the maximum thermal stress varies and increase with the heat transfer coefficient, mass flow rate of air, axial temperature gradient, hole-to-hole spacing (S_1 and S_2) and hole diameter, and decrease with the thermal conductivity.

Accurate analysis of thermal stresses due to nonlinear temperature gradients within the matrix, the so-called "body" stresses, are quite complex. Fortunately, body stresses from the axial temperature distribution are relatively small since this temperature distribution can be made close to a linear temperature gradient. However, stresses due to the radial temperature gradient have been found to be significant at the periphery of the matrix because here the temperature gradient can be steep and also nonlinear.⁽³⁷⁾ Nevertheless, the high stress near the periphery of the matrix could be reduced by decreasing the thermal gradient through increasing the insulation thickness and by changing the brick design and geometry in this particular region.

In addition to maximum thermal stress limitation, compressive creep or hot load deformation must be considered. This is one of the most important considerations in selecting refractories for operation over long time periods. At the very top of the matrix where the matrix temperature is the hottest, susceptibility to compressive creep is highest. Nevertheless, because of the low compressive load at the top, destructive deformation does not usually occur at this location. At the very bottom of the matrix, the compressive load is the greatest, but because the temperature is the lowest and compressive creep resistance the greatest, destructive deformation does not usually occur here either. Generally, it is somewhere between

the top and the bottom of the matrix where the combination of the compressive load and compressive creep resistance causes the maximum distortion to be experienced. In addition to above mentioned factors, other considerations are thermal cyclic fatigue resistance and thermal shock. Thermal shock is particularly important during the initial heat up of the system and in such areas as the top checkers where the refractories are subject to repeated wide temperature fluctuations during each cycle.

State-of-the-art finite element thermostructural analysis computer programs will also be used to quantitatively calculate the temperature and thermal stress distribution in the checker brick matrix. These programs are of large scale and utilize finite element technology for the solution of several classes of engineering analysis problems. The finite element method has been shown to yield good solutions for many engineering problems. The matrix displacement method of analysis is employed throughout. By this method, the engineering structure is visualized as an assemblage of structural elements interconnected at a discrete number of nodal points. If the force-displacement relationships for the individual elements are known, the force-displacement relationship for the entire structure can be obtained by standard matrix methods. The finite element method is also applicable to many problems of non-structural type, with the essential properties of an element of the form encountered in structural analysis. For example, in heat transfer analyses, the finite elements are assigned conductivities which replace the stiffnesses equivalent to that of the actual structure used for structural analyses. Analyses may be nodal forces, body forces, displacements, pressures or temperatures. For heat transfer analyses, they include internal heat generation, convection, radiation, specified temperatures or heat flows.

4. Material Requirements

The preheater refractory materials represents the governing and limiting factors for the temperature which can be obtained in a practical heater design. The major preheater material considerations are:

1. Temperature capability
2. Chemical stability
3. Thermophysical properties
4. Availability and Cost

The indicated requirements for temperature capability and chemical stability can be further delineated for a clean fuel system as:

1. Temperature Capability
 - a) Thermal stability (vaporization and dissociation)
 - b) Melting and fusion
 - c) Phase stability
 - d) Thermal fracture resistance
 - e) Compressive creep resistance
 - f) Cycle fatigue resistance
2. Chemical Stability
 - a) Inert towards combustion gas and air
 - b) Compatibility with adjacent selected refractory material

Since maximum preheat temperatures of 3000°F are desired, thermal stability of the matrix material must include low vapor pressure and high melting or fusion temperature. Additionally, such properties as phase stability must be considered. This is important to the use of zirconia which undergoes a reversible phase transition from monoclinic to cubic as it is heated and cooled. The transition is often accompanied by the mechanical failure of the ZrO_2 structure. The additions of small amounts of ($\sim 10\%$) of other refractories such as MgO , CaO or YT_2O_3 stabilize the ZrO_2 by keeping it in the cubic structure, but the thermal stability of these additives over a period of time must also be considered. Thermal fracture and shock resistance, resistance to compressive creep and to fatigue from repetitive thermal cycling are additional important materials requirements for the matrix design as described in the previous section.

The second requirement for chemical stability involves oxidation resistance to heated air as well as corrosion resistance to combustion gases containing fuel impurities such as sulphur or ash.

The thermophysical properties of the refractory are another important consideration. Since the temperature drop in the matrix or air during a given period is dependent on the heat storage capacity of the matrix material, it is desirable to select materials with high specific heat and density. It is also desirable to have a high thermal conductivity to minimize the thermal resistance to heat transfer through the checker wall.

2.11.6.1.2 Material Selection

The maximum temperatures which determines the requirements of the refractory are found in the combustion chamber, dome, upper part of the matrix and high temperature air outlet ducting. In current blast furnace

stove designs where dome temperatures up to 2800°F are achieved, silica has been the recommended refractory.⁽³⁸⁾ The main advantages of silica are its excellent spalling resistance, very low coefficient of thermal expansion above 1200°F and its high hot load strength almost right up to its melting point. The main disadvantages are the low heat capacity (low density and specific heat) at all temperature and poor spalling resistance below 600°F. It is also relatively low in price compared to high purity alumina, magnesia, and mullite.

For refractory temperatures above 2800 - 2900°F, the applications of high purity magnesia and alumina are basically the only alternatives. Zirconia is also an alternative but experience with this material in large commercial air preheaters is very limited. In addition its very high cost and uncertainty in availability limits the practicability of considering this material at this time. Magnesia has the advantage of a very high heat capacity, and high temperature load bearing characteristics. On the negative side, magnesia has a high thermal coefficient of expansion which makes it susceptible to thermal shock. Alumina, like magnesia has a high heat capacity and hot strength, but in addition has good thermal shock resistance. Both magnesia and alumina are more costly than silica. The use of high purity magnesia has been tried in several plants in Europe,⁽³⁸⁾ however the results of these trials are not known. The use of high purity alumina checker bricks in commercial blast furnace stoves is not known to have been tried. However, high purity alumina (Norton AH-199B) cored brick matrices has been tested in the AERL air preheater facility⁽³⁷⁾ and other systems⁽³⁹⁾ with much success. To date, the AERL air preheater system has accumulated almost 2000 hours of cyclic operation with air preheat temperatures of over 3000°F. Throughout this period, the high alumina refractory lining and cored bricks have remained in excellent condition. Magnesia cored bricks (Norton MH-197) have also been tested in the AERL system in the top portion of the matrices under the same conditions as the alumina. After less than 100 hrs of high temperature operation, the magnesia cored brick had to be replaced due to severe thermal fracturing. Therefore, in the highest temperature regions of the heater system, the use of high purity alumina is considered the preferred refractory material.

For this design, high purity alumina refractory is being specified for the entire matrix, and hot face lining of the vessels (checkerwork chamber and combustion chamber) and high temperature air outlet ducting. In a more detailed analysis, a graded matrix would be considered where other refractory materials could be utilized in cooler regions where their particular limitations would not be exceeded.

2.11.6.1.3 Matrix Design and Operating Conditions

The thermal-mechanical design of the heat exchanger matrix is governed by many factors as previously listed in Table 2-32. In order to evaluate the interrelationship of the following six important design parameters:

1. Matrix volume
2. Checker brick hole size
3. Thermal web stress
4. Reheat gas pressure drop
5. Air blowdown cycle time
6. Matrix aspect (L/D) ratio

matrix design diagrams were developed as shown in Figures 2-114 and 2-115 for Case I and Case II referred to earlier. These diagrams were developed based on Eqs. (2-41) through (2-48) with the following assumptions:

1. The respective air blowdown and gas reheat mass flow rates are kept constant.
2. The respective air blowdown and gas reheat inlet temperature, outlet temperatures, and temperature variations over the cycle are kept constant.
3. The temperature difference between the matrix and the fluid is kept constant along the length of the matrix.
4. The axial temperature gradients of the solid and fluids are linear.
5. The air blowdown cycle time is equal to the gas reheat cycle time.
6. The matrix material and fluids are kept constant.

Figure 2-114 Matrix Design Diagram Case I (Reference Design)

Figure 2-115 Matrix Design Diagram Case II (Alternate Design)

7. There is no heat loss to the surroundings.
8. For a given checker brick hole diameter there is a minimum web thickness which is considered practical for fabrication. In terms of optimization of the heater matrix this establishes the lower limit of the matrix volume with respect to the web thickness. Figure 2-116 shows the assumed web thickness for a given hole diameter. As seen, in going from the larger hole diameter to the smaller diameters, the relative web thickness (S/D) increases because of practical manufacturing limitations. Above 1-1/4 inch hole diameters, the minimum web thickness for high volume production is basically a fixed percentage of the hole diameter.

The diagrams shown in Figures 2-114 and 2-115 for Case I and Case II were based upon the design parameters in Table 2-31. In determining the optimal design of the heat exchanger matrices, the constraints listed below were established. These constraints were based (where applicable) on current state-of-the-art blast furnace stove design practices and operating experiences with the AERL high temperature regenerative air heater facility as described below:

a. Maximum Specific Mass Flow Rate - 0.75 lb/sec-ft^2

With the demand for higher air flow rates and temperatures from blast furnace stove systems, specific mass flow rates in these systems have increased up to $\sim 0.6 \text{ lb/sec-ft}^2$ (20 std. ft/sec) (Ref. 32). The AERL facility was designed and has operated successfully at even higher specific mass flow rates of 0.9 lb/sec-ft^2 .

b. Maximum Thermal Web Stress - 1200 lb/in.^2 at 2000°F

This value is based on practical experience with the AERL preheater system and refractory manufacturer data on the high purity alumina planned for use in the heater matrix. This stress value includes a factor of 1.5 (see page 448) when calculating the web stress from Eq. (2-46).

c. Minimum Blowdown and Reheat Period - 15 minutes

As the heat exchanger system is cycled more frequently, the amount of matrix mass required for heat storage may be reduced. Nevertheless for large systems, a practical limit in switchover time is reached which is dependent on such factors as valve switching time, pressurization/depressurization

Figure 2-116 Checker Brick Geometry

time, and reheat burner startup and shutdown time. In addition, as the heaters are cycled more frequently, valve durability and overall system control is reduced. Based upon experience with the AERL regenerative air preheater system and that of the blast furnace stove industry where periods as low as 20 minutes have been used⁽²⁷⁾ a minimum blowdown and reheat period of 15 minutes is specified.

d. Maximum Reheat Gas Pressure Drop in Matrix - 0.05 atm

In order to minimize auxiliary power requirements a maximum pressure drop of 0.05 atm based on a "perfect flue" was specified. For actual sizing of equipment, this pressure drop is assumed to be double or 0.1 atm to take into account for misalignment of the flues, dirt build up, and general degradation over time. In addition to the pressure drop through the matrix, other losses will occur through the reheat burner, ducting and valving.

e. Minimum Matrix L/D - 4 to 1

The overall length to diameter ratio of the heat exchanger matrix is important in a number of ways. For low L/D ratios, problems related to nonuniform flow of the fluids becomes more pronounced causing reduced heat exchanger performance. In addition, as the checkerwork chambers increase in diameter, practical dome design limits are reached. Current blast furnace stove designs are generally designed with L/D ratios between 4 and 5. Therefore a lower limit has been placed at 4 to 1 to stay within current design practices.

f. Minimum Hole Diameter - 1.0 inches

Current high performance checkers for blast furnace stoves in the United States have hole diameters of 1-1/2 inches. However, in European stoves, hole diameter down to 1.18 inches (30 mm) have been used in checker bricks. Problems related to alignment of the flues and plugging due to contamination buildup limits the reduction of the hole diameter in practice to about 1 inch for large commercial systems. However, hole diameter down to 1/4 inch were used successfully in the AERL regenerative heater facility and other specialized systems.⁽³⁹⁾

With the above constraints, an acceptable design envelope was established as shown by the shaded regions in Figures 2-114 and 2-115. Since the cost of the heat exchanger system is basically a function of the

matrix volume, the optimal design is that which satisfies these constraints and has the smallest volume. Based on these diagrams, a number of important design guidelines are evident. For large heater systems, high efficiency checker bricks with very small holes, e. g., 1/4 inch diameter, are currently not practical even though they result in a very small matrix. This is because very short cycle times are required due to the very low mass and thus heat storage capacity (MCp) of the matrix. As the matrix hole size increases, a larger matrix is required to achieve a given performance. However, this also results in a more conservative design with respect to cycle times, susceptibility to plugging, pressure drops, and fatigue stresses. In addition, checker brick designs with very small holes results in either a heater system which has matrices with low aspect ratios which gives rise to poor fluid flow uniformity, or requires the use of many smaller units.

Based on these design diagrams, the designs summarized in Tables 2-33 and 2-34 were developed for Case I and Case II, each for 3000°F and 2500°F air preheat temperatures, and series and staggered-parallel modes of operation respectively. Figures 2-117 and 2-118 show the axial matrix temperature distributions for each of these designs.

The final design configuration was based on the worst case condition with one heater assumed shut down for repair. This results in series operation and with only three of the four heat exchanger units operating where one heater unit would be on blowdown, one heater unit on reheat and one heater unit switching at any period of time. In this mode of operation, the air blowdown and gas reheat cycles are equal. In normal operation, the system would operate in a staggered-parallel mode with two heaters on blowdown, one heater on reheat and one heater switching.

The performance of two alternate checker brick configurations with larger and more conservative hole diameters of 1-1/4, 1-1/2 inches were calculated for the same operating conditions as the 1 inch hole checker brick. Table 2-33 summarizes the results of all these calculations. It can be observed that even with a 1-1/2 inch hole standard checker design, an air preheat temperature of 3119-2932°F can be achieved, although the checker brick will be subjected to somewhat higher thermal web stresses. The pressure drop across the matrix however decreases.

Figure 2-117 Case I - Axial Matrix Temperature Distribution (Series and Staggered-Parallel Modes of Operation)

Figure 2-118 Case II - Axial Matrix Temperature Distribution (Series and Staggered-Parallel Modes of Operation)

For these designs, the thermal web stress analysis was based on the simplified model of treating the checker bricks as a series of parallel, infinitely long cylindrical tubes. In order to predict more accurately the actual stresses, state-of-the-art finite element thermostructural analysis computer programs were also used to quantitatively calculate the temperature and thermal stress distribution in the checker brick matrix for comparison with the simplified model. The finite element method of thermal stress analysis requires a large amount of programming inputs and computer time. Therefore, a detailed thermal stress analysis was performed only on the final selected matrix design under the most severe operating conditions and at the matrix temperature where the thermal fracture stress resistance of the refractory is a minimum. For alumina this temperature is around 2000-2200°F. Figure 2-119 shows the representative segment of the checker brick and Figure 2-120 shows the three dimensional structural model used in the analysis.

For the thermal analysis, the boundaries of the model were assumed to be adiabatic surfaces except for the hole where transient fluid temperatures and heat transfer coefficients were specified. For stress analysis, structural restraints were placed perpendicular to the planes of symmetry except in the hole as shown in Figure 2-120. The thermal stress analysis, did not include the "body" stresses caused by axial and radial temperature gradients in the matrix or from the compressive load of the matrix. The stresses due to the axial temperature gradient and from the weight of the matrix are known to be relatively small, however, the stresses due to the radial temperature gradient may be quite significant at the very outer periphery of the matrix where the radial temperature is steep.⁽³⁷⁾ Nevertheless, since this is really a local stress phenomena and to simplify the analysis, no radial temperature gradient in the matrix was assumed.

The results of the finite element stress analysis indicated the maximum principle stress in tension occurs at the surface of the hole during the blowdown cycle, as was expected. However, as shown in Figure 2-121, the maximum principle stress was higher by a factor of ~ 1.5 over that calculated by Eq. (2-46) for the simplified model. However, for design purposes, it is assumed that the relative stress values calculated from Eq. (2-46) still holds true and that the actual thermal web stress is higher by factor of 1.5.

Figure 2-119 Brick Geometry, Dimensions and Section Selected for Analysis

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Figure 2-120 Three-Dimensional Structural Model and Boundary Conditions

Figure 2-121 Maximum Principal Stresses in Checker Brick at 2000°F Series Operation

2.11.6.1.4 Checker Brick Design

The design of the checker brick involves consideration of the following factors which basically determines its manufacturing feasibility and costs for high volume production:

1. Shape, hydraulic diameter and wall thickness of the checker flues
2. Geometry and overall dimensions of checker brick
3. Refractory material
4. Interlocking arrangement of checker brick

The shape, hydraulic diameter and wall thickness of the checker flues determine the complexity of the mold fixture and the integrity of the molded shape in its unfired (green) state. For example, when the flue walls become very thin, they are easily fractured when the pressed brick is removed from the mold fixture. Also, as the flue diameter becomes small, the pins used to form the flues become flexible and they are subject to breakage, especially in high volume production presses.

The geometry and overall dimensions of the checker brick are important from both a material property and manufacturing capability point of view. As the checker bricks increase in size, pressing of the refractory powder to a uniform density becomes more difficult. Also overall press capacity may be a limiting factor.

The checker brick refractory material is important in both direct material costs and feasibility of manufacturing. For example, high purity refractories in addition to being expensive, are sometimes more difficult to press because of the low amount of binder (impurities) which can be used to maintain their structure in the green state. In designing the checker brick, the scheme for installing the checker bricks must be considered so that the structural integrity of the matrix assembly is maintained during initial assembly and heater operation. For example, when the checker work is installed in vertical column, twisting and lateral displacement of the checkers has been known to occur. A successful solution to this problem has been to install the checkers in a keyed system so that a single checker spans three bricks of the course underneath.⁽⁴⁰⁾ This same system is proposed in this design study. An interlocking arrangement between

the various checker bricks is also an important consideration in order to minimize misalignment of the checker flues. This is especially critical with small diameter flues where misalignment may cause excessive pressure drop, and faster plugging of the flues.

Shown in Figure 2-122 is the proposed checker brick design and in Figures 2-123, 2-124 and 2-125 three alternate designs. The interlocking keys used to align the checker flues for initial assembly during heater operation are also shown. The basic checker size and keying arrangements are similar to those currently used in blast furnace stove systems. These checker designs were studied by the Engineering and Manufacturing Departments of A. P. Green Refractories, Mexico, Missouri and were found feasible for high volume production using their 99 AD high alumina refractory.

Figure 2-122 1.0-Inch Hole Dia Checker Brick

Figure 2-123 1 1/4-Inch Hole Dia Checker Brick

Figure 2-124 1 1/4-Inch Hole Dia Checker Brick

Figure 2-125 1 1/2-Inch Hole Dia Checker Brick

2.11.6.2 Checkerwork Chamber and Combustion Chamber

The checkerwork and combustion chamber vessels are basically pressure vessels and their designs follow general considerations for pressure vessels. In the design of the vessels special attention was paid to three important factors which were the internal pressure, thermal expansion forces of the refractory and corrosion. Although the design of the vessel is dependent on the maximum internal fluid pressure, additional loads created by the radial and axial expansion of the insulation and checker matrix (see Section 2.11.6.2.4) must be accounted for. Additional factors which were considered include the dead loads of the shell, insulation and matrix.

2.11.6.2.1 Checkerwork Vessel Shell Design

The design of the checker chamber vessel shell conforms to basic ASME Boiler and Pressure Vessel Codes for Unfired Pressure Vessels, Section VIII, Division 1. Because of the preliminary nature of this design study, the required reinforcement for openings in the shell and heads were not analyzed. The following formula (Paragraph UG 27) was applied to calculate the vessel shell thickness exclusive of corrosion:

$$t_p = \frac{PR}{SE - .6P} \quad (2-49)$$

where

P = design pressure

R = inside radius of shell

S = maximum allowable stress

E = joint efficiency

The following conditions were used in Eq. (2-49):

$$P = \begin{matrix} P_{\text{internal}} \\ \text{pressure} \end{matrix} + \begin{matrix} P_{\text{refractory}} \\ \text{expansion} \\ \text{pressure} \end{matrix} = 75 + 50 = 125 \text{ psig}$$

R = 114.5 in.

S = $17.5 \times 10^3 \text{ lb/in}^2$ between -20 and 600°F (ASTM 516, Grade 70)

E = 1.0 for fully radiographed, full penetration welds

Based on these conditions

$$t_p = \frac{(125)(114.5)}{(17.5 \times 10^3)(1) - (.6)(125)} = .821 \text{ inches}$$

Since the vessel shell will be subject to corrosion, an additional thickness t_c , was added where

$$t_c = .125 \text{ inches}$$

The total required shell thickness is then

$$t = t_p + t_c = .821 + .125 = .946 \text{ inches}$$

A shell thickness of 1.0 inches was selected since this is a standard mill size.

The thickness of the top catenary head was calculated by approximating its shape as that of an ellipsoidal head and applying the following formula (Paragraph UA4) exclusive of corrosion.

$$t_p = \frac{PDK}{2SE - .2P} \quad (2-50)$$

where

$$K = \frac{1}{6} \left[2 + \left(\frac{D}{2h} \right)^2 \right] \quad (2-51)$$

and

P = design pressure

D = inside diameter of head at short

S = maximum allowable stress

E = joint efficiency

The following conditions were used:

$$P = P_{\text{internal pressure}} + P_{\text{refractory expansion pressure}} = 75 + 50 = 125 \text{ psig}$$

D = 276 inches

$$K = \frac{1}{6} [2 + (1.2)^2] = .57$$

S = 17.5×10^3 lb/in² between - 20 and 650°F (ASTM 516, Grade 70)

E = 1.0

Based on these conditions

$$t_p = \frac{(125)(276)(.57)}{(2)(17.5 \times 10^3)(1) - (.2)(125)} = .562 \text{ inches}$$

Including a provision for corrosion of

$$t_c = .250 \text{ inches}$$

the total required thickness of the top head is

$$t = t_p + t_c = .562 + .250 = .812 \text{ inches}$$

Since the required top head thickness was calculated using an approximate analysis, a top head thickness of 1.0 inches was chosen to be conservative. For the final design, a detailed stress analysis would be required to assure its adequacy.

The bottom head of the vessel, unlike the top head, also carries the dead load of the insulation, matrix and matrix support structure. This additional load is assumed to be equally distributed over the head area. The bottom head is a standard ellipsoidal head with an inside head radius equal to one forth the inside diameter of the skirt. The thickness of the bottom head was calculated by applying Eq. (2-50) and (2-51) for the following condition:

$$P = P_{\text{internal pressure}} + P_{\text{dead load pressure}} + P_{\text{refractory expansion pressure}} = 75 + 125 + 75 = 275 \text{ lb/in}^2$$

$$D = 229 \text{ inches}$$

$$K = \frac{1}{6} [2 + (2)^2] = 1$$

$$S = 17.5 \times 10^3 \text{ lb/in}^2 \text{ between } -20 \text{ and } 650^\circ\text{F (ASTM 516, Grade 70)}$$

$$E = 1$$

Based on these conditions, the thickness of the bottom head, exclusive of corrosion is

$$t_p = \frac{(275)(229)}{(2)(17.5 \times 10^3) - (.2)(275)} = 1.802 \text{ inches}$$

Including provision for corrosion of

$$t_c = .125 \text{ inches}$$

the total required bottom head thickness is

$$t = t_p + t_c = 1.802 + .125 = 1.927$$

Therefore a bottom head thickness of 2.0 inches was chosen because this is a standard mill size.

2.10.6.2.2 Combustion Chamber Vessel Shell Design

The design of the combustion chamber vessel must consider the same design factors as the heat exchanger vessel, less the effect of the checker-work. As with the checker chamber vessel, the required reinforcement for openings in the shell and head have not been included. Equation (2-49) was applied to calculate the combustion chamber shell thickness for the following conditions:

$$P = P_{\text{internal pressure}} + P_{\text{refractory expansion pressure}} = 75 + 50 = 125 \text{ psig}$$

$$R = 95.25 \text{ inches}$$

$$S = 17.5 \times 10^3 \text{ lb/in}^2 \text{ between } -20 \text{ and } 650^\circ\text{F (ASTM 516, Grade 70)}$$

$$E = 1.0$$

Based on these conditions

$$t_p = \frac{(120)(95.25)}{(17.5 \times 10^3)(1) - (.6)(125)} = .682$$

Including a liberal corrosion allowance of

$$t_c = .25 \text{ inches}$$

because of the high susceptibility of stress corrosion in the combustion chamber, the total required shell thickness is

$$t = t_p + t_c = .683 + .25 = .933$$

A combustion chamber shell thickness of 1.0 inches was selected because this is a standard mill size.

The thickness of the top catenary head was calculated, as with the heat exchanger vessel top head, by approximating its shape as that of an ellipsoidal head and applying Eq. (2-50) and (2-51), exclusive of corrosion. The following conditions were applied:

$$P = P_{\text{internal pressure}} + P_{\text{refractory expansion pressure}} = 75 + 50 = 125 \text{ psig}$$

$$D = 237.5 \text{ in.}$$

$$K = \frac{1}{6} [2 + (1.2)^2] = .57$$

$$S = 17.5 \times 10^3 \text{ lb/in}^2 \text{ between } -20 \text{ and } 650^\circ\text{F (ASTM 516, Grade 70)}$$

$$E = 1.0$$

Based on these conditions

$$t_p = \frac{(125)(237.5)(.57)}{(2)(17.5 \times 10^3)(1) - (.2)(125)} = .483 \text{ inches}$$

Including a corrosion allowance of

$$t_c = .25 \text{ inches}$$

the total required thickness of the top head is

$$t = t_p + t_c = .483 + .25 = .733 \text{ inches}$$

Since the top head thickness was calculated using an approximate analysis, a top head thickness of 1.0 inches was chosen to be conservative.

2.11.6.2.3 Refractory Dome Design

The construction of the refractory domes above the checkerwork and combustion chambers is important in that it must allow for axial expansion of the side wall refractory and be structurally stable. Independent axial expansion of the side wall refractory and dome is accomplished by utilizing a mushroom shape design as illustrated in Figure 2-109. The shape of the dome is also critical. Basically there are two standard shapes of domes, segmented spherical and catenary. The height of rise of these shapes are commonly expressed in inches per foot of span. For segmented spherical domes the rise should not be less than about 1 inch nor more than about 3 inches per foot of span. A dome which is too flat is subject to excessive stresses and one too steep lacks stability. For rises greater than 3 inches per foot of span, a catenary dome form has greater stability because the line of thrust in a dome follows a catenary curve. (41, 42)

Two important effects were considered in the design of the dome. First, during initial construction the brick surfaces are usually in full contact over the entire surface when cold. As the dome is heated, the dome will increase in curvature and thus change the original line of thrust. Another temperature effect occurs due to unequal thermal expansion between the hot inner and cold outer surfaces. Both of these temperature effects tend to cause the joints between the bricks to open and thus increase the compressive stress on the hot ends of the brick. If the stresses become too great, the hot edges of the brick may deform or break off. The first temperature effect can be reduced by using a high dome and the second by using insulation on the outside to reduce the temperature gradient between the inner and outer surfaces. Both of these solutions were taken into consideration in the dome designs.

The maximum compressive stress in a segmented spherical or catenary dome occurs at the base of the dome and is calculated from the equation:

$$\sigma_{\text{max compressive}} = \frac{\rho A}{\pi S} \frac{1}{\sin \theta} \quad (2-52)$$

where

ρ = density of refractory

A = mean surface area of dome

S = mean span of dome

θ = angle between tangent at ends of catenary or spherical curve at the horizontal

Shown in Figure 2-126 is a curve showing the change in the compressive stress for various rises per foot of span for segmented spherical and catenary domes in cold and heated conditions.

Since the weight increases with the rise for a constant span and the normal thrust component decreases with the rise, there is an optimum rise per unit span where the thrust load or compressive stress at the base is a minimum.

In a cold catenary or segmented spherical dome, the minimum stress occurs at a rise of approximately 3.5 in./ft of span. In the heated condition, unless full compensation is made for thermal expansion, the position of the line of thrust will change from that of the cold case. General practice for estimating the stresses in the heated condition has been to

Figure 2-126 Maximum Compressive Stress in Dome

multiply the values determined in the cold conductor by a factor of 2 to 3 depending on the temperature and type of refractory material. However, for a more exact analysis, the expression shown below was proposed by McDowell⁽⁴³⁾ for estimating this correction factor for arch roofs:

$$\sigma_{\text{max heated}} = \sigma_{\text{max cold}} \left[\frac{h + 1.5 T (1 - \cos \theta)}{h - T \cos \theta} \right] \quad (2-53)$$

where

h = rise of arch in feet

T = thickness of arch in feet

θ = central half angle of arch

Using this expression as a relative correction factor for the segmented spherical and catenary domes in the heated condition, the minimum normal stress occurs at apparently a rise of 4.5 in./ft of span for a catenary dome and 4.25 in./ft for a segmented spherical dome. Although the segmented spherical dome gives a slightly lower compressive stress a catenary dome with a rise of 4.5 in./ft of span was selected for the checkerwork and combustion chambers. This specialized application requires domes which must have maximum structural stability at very high temperatures and these requirements are best satisfied with a catenary dome design.

2.11.6.2.4 Insulation Design and Heat Loss

In order to reduce heat loss and to prevent the heater vessels from overheating, refractory insulation is provided. Heat loss through the vessel lowers the thermal performance of the heat exchanger and may also result in a non-uniform fluid temperature. Reduction of heat losses also has the advantage of reducing the radial temperature gradient in the matrix which can impose relatively large thermal stresses.

The heat loss through the refractory insulation is mainly by conduction through the refractory, and by radiation and convection from the inner and outer surfaces. The heat transmitted by conduction through the wall varies directly with the temperature difference between the outer and inner surfaces of the wall, the thermal conductivity of the refractory, and inversely with the thickness.

The heat which passes through the walls is dissipated to the outside environment by convection and radiation. The rate of heat transfer depends on the temperature of the surface and surroundings, rate of circulation of air, type of surface (horizontal, vertical), etc.

The heat losses and thermal gradients through the refractory insulation were calculated from the following equation

$$\frac{Q}{L} = \frac{T_i - T_o}{\frac{1}{2\pi r_1 h_1} + \frac{\ell \ln(r_2/r_1)}{2\pi k_a} + \frac{\ell \ln(r_3/r_2)}{2\pi k_b} + \frac{\ell \ln(r_4/r_3)}{2\pi k_c} \dots + \frac{1}{2\pi r_4 h_o}} \quad (2-54)$$

where

Q = rate of heat transfer

L = length

T_i, T_o = inside and outside fluid temperature

$r_1, r_2, r_3, r_4 \dots$ = radius of each layer (from inside to outside)

h_i, h_o = inside and outside overall (radiation + convection) heat transfer coefficient

$k_a, k_b, k_c \dots$ = average thermal conductivity of each layer (from inside to outside)

The thermal insulation was designed to provide a combination of good thermal properties for low heat loss and mechanical properties for adequate strength and abrasion resistance at the design temperature. Listed in Table 2-36 is a summary of the insulation design, calculated temperature distribution, and heat flux through the insulation. The heat loss to the outside environment was based on a standard 70°F day with no wind.

The insulation design consists of a composite refractory lining where the inner layer exposed to the maximum fluid temperature possesses good high temperature strength and abrasion resistance characteristics and the successive layers better insulation characteristics. The outer most layer next to the vessel shell has a low strength, high porosity characteristic so it can provide for radial and axial expansion of the stronger and more dense inner insulation.

The inner most hot face insulation is of high purity (99%) and density (18-21% porosity) alumina because of its high hot strength and abrasion

TABLE 2-36

CHECKERWORK CHAMBER COMBUSTION CHAMBER AND DOME INSULATION

DESCRIPTION	CASE NO	REF SKETCH	NO OF INSUL LAYERS	INSUL LAYER NO	INSULATION MANUFACTURER	INSUL SPEC OR EQV	INSUL DIMENSIONS			INTERFACE TEMP					HEAT FLUX Q BTU/HR FT ²
							O D IN	I D IN	THICK- NESS IN	t_1 °F	t_1 °F	t_2 °F	t_3 °F	t_0 °F	
CHECKERWORK CHAMBER															
3300 - 3000 °F HOT FACE LEVEL	I, II	3	4	F	A P GREEN	99AD	208	181	13 5	3300	2708	1942	1025	350	876
				G	A P GREEN	G33	217	208	4 5						
				H	A P GREEN	G33	226	217	4 5						
				I	A P GREEN	MINERAL WOOL	229	226	1 5						
3000 - 2500 °F HOT FACE LEVEL	I, II	3	4	F	A P GREEN	99AD	208	181	13 5	3000	2576	1982	780	290	628
				G	A P GREEN	G33	217	208	4 5						
				H	A P GREEN	G26	226	217	4 5						
				I	A P GREEN	MINERAL WOOL	229	226	1 5						
2500 - 1600 °F HOT FACE LEVEL	I, II	3	4	F	A P GREEN	99AD	208	181	13 5	2500	2186	1601	580	250	443
				G	A P GREEN	G30	217	208	4 5						
				H	A P GREEN	G23	226	217	4 5	2000	1787	1345	574	200	300
				I	A P GREEN	MINERAL WOOL	229	226	1 5						
1600 - 500 °F HOT FACE LEVEL	I	3	4	F	A P GREEN	99AD	208	181	13 5	1600	1455	1103	551	170	215
				G	A P GREEN	G28	217	208	4 5						
				H	A P GREEN	G23	226	217	4 5	500	465	354	207	105	52
				I	A P GREEN	MINERAL WOOL	229	226	1 5						
COMBUSTION CHAMBER	I, II	2	3	C	A P GREEN	99AD	180	144	18	3350	2490	1056	-	370	957
				D	A P GREEN	G30	189	180	4 5						
				E	A P GREEN	MINERAL WOOL	191	189	1						
CHECKERWORK CHAMBER DOME	I, II	1	2	A	A P GREEN	99AD	-	-	18	3300	2178			420	1247
				B	A P GREEN	G30	-	-	4 5						
COMBUSTION CHAMBER DOME	I, II	1	2	A	A P GREEN	99AD	-	-	18	3350	2210			425	1266
				B	A P GREEN	G30	-	-	4 5						

REF SKETCH NO 1
DOME CONSTRUCTION REF SKETCH NO 2
COMBUSTION CHAMBER REF SKETCH NO 3
CHECKERWORK CHAMBER

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resistance characteristics. However, in the high temperature regions of the checker and combustion chambers the hot face of this insulation may be somewhat plastic and unable to support a large load. For this reason, the insulation is made relatively thick (13 1/2 - 18 inches) to take advantage of the thermal gradient through the brick. In this fashion, the compressive loads will be carried by the cooler portion of the bricks which have more than adequate strength. To provide a margin of safety, the maximum mean temperature through this hot face insulation has been limited to 2900-3000°F which is within recommended limits for high alumina bricks. (42)

Next to the steel shell is a thin highly compressible layer of insulation to provide for thermal expansion of the insulation. In the design of the insulation, thermal expansion of the refractory must be accounted for if severe expansion problems are to be avoided. Thermal expansion of the insulation takes place both in the axial and radial directions. As the insulation expands radially, its movement will eventually be restricted by the outside steel shell. If this expansion is restricted, compressive forces will be generated which may cause destructive deformation of the refractory and/or high shell stresses.

In addition to the radial expansion, vertical expansion of the insulation also occurs. Owing to a radial compressive load, the axial expansion of the refractory generates a drag force which acts as a compressive load on the insulation and a tensile load on the steel shell. These forces are increased even further if the matrix is installed too tightly and not allowed to expand independent of the refractory insulation. (32, 44) In order to minimize these expansion forces in this design, a highly compressible insulation was used next to the steel shell to allow for axial and radial expansion of the insulation with minimum resultant forces. This insulation is designed to limit the refractory expansion pressure to 50-75 lb/in².

Another important consideration which was taken into account in the design of the refractory insulation had to do with inter-crystalline stress corrosion of the steel vessel. In recent years with the gradual increase in blast furnace stove temperatures, inter-crystalline stress corrosion in the steel shells has lead to structural failure of some vessels. Studies of this problem have shown that this problem is found to occur when the following conditions are present:

1. Condensation of corrosive acids formed from nitrogen and sulfur oxides produced at the higher combustion temperatures.
2. Stress concentrations above the yield point of the steel shell.
3. Formation of corrosive salts such as calcium nitrate formed by oxidation of nitrogen and its reaction with a calcium containing component of the insulation.

Inter-crystalline stress corrosion from corrosive acids can be prevented by maintaining a shell temperature above the dew point of the liquid solution containing the corrosive agent so that condensation is prevented.^(40, 45)

For this reason, the steel shells of the combustion chamber and upper portion of the checker chamber, where stress corrosion cracking is most susceptible, are designed to operate at 300°F or higher depending upon the gas composition. External insulation will be applied to the vessel shell where it is found necessary in order to maintain these temperatures during actual operation.

2.11.6.3 Ducting

2.11.6.3.1 Piping and Expansion Joints

The design of the air preheater system piping must take into consideration many of the factors pertinent to the design of the heat exchanger and combustion chamber vessels. For instance, the piping requires internal refractory insulation to prevent overheating, which creates refractory expansion forces in the pipe in addition to the normal pressure. Also, consideration must be given to the fluid velocity to keep erosion of the refractory insulation and fluid pressure drop to practical limits. Furthermore, consideration must be given to pipe support and thermal expansion.

Thermal expansion joints of the bellows type are considered. These will be installed in the piping system at locations based on current blast furnace stove practice. The bellows expansion joints will be fabricated from an austenitic chrome-nickel molybdenum steel, SA-240, Grade 316, to prevent stress corrosion cracks in the high stress area.

A refractory lined cross-over pipe is used to connect the combustion chamber to the checkerwork chamber. Due to the considerable size and weight of the vessels, the interconnection must be designed to take up any differential thermal expansion of the vessels both in the horizontal and vertical direction. To accomplish this, bellows expansion joints have been placed

at the far ends of the interconnecting pipe as illustrated in Figure 2-109. Since the bellows are unable to sustain any appreciable axial thermal or pressure induced reaction forces, tie rods are placed between the vessels to fix their centerline. These tie rods are connected to each vessel through a reinforcing band placed around the vessel and jointed to the flanges of each bellows. This reinforcing band is used to avoid deformation of the vessel walls from the high axial forces which may be developed in the pipe. Since the expansion bellows and duct will be subjected to high stress levels and therefore are prone to stress corrosion cracking, they will be fabricated from SA-240, Grade 316 steel.

The design standard for the piping is based on the ASME Power Piping Code. The following formula (Paragraph 104.1.2) was applied to calculate the required pipe wall thickness:

$$t_m = \frac{PD_o}{2(SE + Py)} + A \quad (2-55)$$

where

t_m = minimum required wall thickness

P = internal design pressure

D_o = outside diameter of pipe

SE = maximum allowable stress in material due to internal pressure and joint efficiency

A = corrosion allowance

y = .4 below 900°F - .7 above 1150°F

Shown in Table 2-37 are the design conditions, pipe sizes and wall thickness for the various pipes in the system. Taking into account the internal refractory insulation, the piping is sized so the fluid does not exceed a velocity of 200 ft/sec.

2.11.6.3.2 Piping Insulation and Heat Loss

The design of the piping insulation takes into consideration the same factors as described in Section 2.11.6.2.4. However, provisions for thermal expansion in the axial direction will be made by providing expansion joints longitudinally between the bricks at suitable intervals. The heat loss and thermal gradients through the refractory walls were calculated from Eq. (2-54). Table 2-38 is a summary of the insulation design temperature distributions through the insulation, and corresponding heat flux.

TABLE 2 - 37
SEAM WELDED PIPE

Description	Case	O. D. of	Mat. ASTM #	Design Pressure			Design Temp.	Design Stress	Wall Thickness			
		Normal Fluid		Ref. Exp.	Total	t _{min.}			t _{corr.}	t _{total}	t _{design}	
		in.		psig.	psig.	psig.	°F	lb/in ²				
Gas Reheat Piping												
Burner Air Inlet	I and II	56	A312 GR TP304	15	0	15	1200	5.1 × 10 ³	.082	.125	.207	.375
Burner Fuel (Gas) Inlet	I and II	36	A312 GR TP304	15	0	15	1200	5.1 × 10 ³	.053	.125	.178	.375
Burner Recycle Gas Inlet	I and II	28	A312 GR TP304	15	0	15	1200	5.1 × 10 ³	.041	.125	.166	.375
Burner-Checker Chamber												
Crossover	I and II	135	SA240 GR 316	75	125	200	500	17.9 × 10 ³	.750	.250	1.06	1.00
Combustion Gas Exit												
Nozzle	I	74	516 GR 70	75	75	150	500	17.5 × 10 ³	.316	.125	.441	.500
Nozzle	II	90	516 GR 70	75	75	150	500	17.5 × 10 ³	.384	.125	.509	.500
Manifold	I	58	A312 GR TP304	15	0	15	1000	8.3 × 10 ³	.052	.125	.177	.375
Manifold	II	90	516 GR 70	15	75	90	500	17.5 × 10 ³	.230	.125	.355	.375
Air Blowdown Piping												
Low Temperature Air Inlet												
Nozzle	I	32	516 GR 70	75	75	150	650	17.5 × 10 ³	.137	.125	.262	.50
Nozzle	II	46	516 GR 70	75	75	150	650	17.5 × 10 ³	.196	.125	.321	.50
Manifold	I	22	A53 GR B	75	0	75	650	12.7 × 10 ³	.065	.125	.190	.375
Manifold	II	28	A312 GR TP304	75	0	75	1200	5.1 × 10 ³	.205	.125	.330	.375
High Temperature Air Outlet	I and II	76	516 GR 70	75	175	225	500	17.5 × 10 ³	.486	.125	.611	.625
Auxiliary												
Pressurization and Depressurization	I	16	A53 GR B	100	0	100	650	12.7 × 10 ³	.063	.125	.188	.375
Pressurization and Depressurization	II	22	A312 GR TP304	100	0	100	1200	5.1 × 10 ³	.212	.125	.337	.375

TABLE 2-38
PIPE INSULATION

DESCRIPTION	CASE NO	REF SKETCH	NO OF INSUL LAYERS	INSUL LAYER NO	INSULATION MANUFACTURER	INSUL SPEC OR EQV	INSUL DIMENSIONS			INTERFACE TEMP				HEAT FLUX
							O D IN	I D IN	THICKNESS IN	t ₀ °F	t ₁ °F	t ₂ °F	t ₀ °F	Q BTU/HR-FT ²
GAS REHEAT PIPING														
BURNER AIR INLET	I, II	1	1	A	JOHNS-MANVILLE	THERMO-12	74	56	9	1100	-	-	110	60
BURNER FUEL INLET	I, II	1	1	A	JOHNS-MANVILLE	THERMO-12	54	36	9	1100	-	-	105	51
BURNER RECYCLE GAS INLET	I, II	1	1	A	JOHNS-MANVILLE	THERMO-12	46	28	9	1100	-	-	105	49
BURNER - CHECKER CHAMBER CROSSOVER	I, II	3	3	D	A P GREEN	99AD	121	94	13.5					
				E	A P GREEN	G33	127	121	3	3300	2425	1760	360	1040
				F	A P GREEN	G26	133	127	3					
COMBUSTOR GAS EXIT														
NOZZLE	I	2	2	B	A P GREEN	GREENCAST 94	65	57	4	850	740	-	175	227
				C	A P GREEN	VSL 50	73	65	4					
NOZZLE	II	2	2	B	A P GREEN	GREENCAST 94	76	68	4	1450	1300	-	200	298
				C	A P GREEN	VSL 50	89	76	6.5					
MANIFOLD	I	1	1	A	JOHNS-MANVILLE	THERMO-12	74	58	8	850	-	-	105	48
MANIFOLD	II	2	2	B	A P GREEN	GREENCAST 94	76	68	4	1450	1302	-	200	295
				C	A P GREEN	VSL 50	89.25	76	6.63					
AIR BLOWDOWN PIPING														
LOW TEMP AIR INLET														
NOZZLE	I	2	1	B & C	A P GREEN	GREENCAST 94	31	21.25	4.88	500	-	-	250	436
NOZZLE	II	2	2	B	A P GREEN	GREENCAST 94	35.25	27.25	4	1100	960	-	180	243
				C	A P GREEN	VSL 50	45	35.25	4.88					
MANIFOLD	I	1	1	A	JOHNS-MANVILLE	THERMO-12	30	22	4	500	-	-	105	37
MANIFOLD	II	1	1	A	JOHNS-MANVILLE	THERMO-12	46	28	9	1100	-	-		49
HIGH TEMP AIR OUTLET														
NOZZLE AND MANIFOLD	I, II	3	3	D	A P GREEN	99 AD	59.75	41.75	9	3200	2600	1600	350	910
				E	A P GREEN	G33	68.75	59.75	4.5					
				F	A P GREEN	G26	74.75	68.75	3					
AUXILIARY PIPING														
PRESS/DEPRESS	I	1	1	A	JOHNS-MANVILLE	THERMO-12	24	16	4	600	-	-	105	46
PRESS/DEPRESS	II	1	1	A	JOHNS-MANVILLE	THERMO-12	46	22	12	1200	-	-	105	34

REF SKETCH NO 1
INSULATION ON OUTSIDE OF PIPE

REF SKETCH NO 2
INSULATION ON INSIDE OF PIPE

REF SKETCH NO 3
INSULATION ON INSIDE OF PIPE

H2889

2.11.6.4 Switchover Valving

2.11.6.4.1 Valve Specifications

Because of the cyclic nature of the heat exchanger operation, switchover valves are required to isolate the reheat gas from the blowdown cycle air. The operating conditions of these valves are severe and are subjected to differential pressures as high as 6 atmospheres and a peak preheated air temperature of 3250°F. Following the practice of industrial blast furnace stove systems, refractory lined, water cooled gate shutoff valves will be used in the high temperature air ducts.

It is recognized that no industrial experience exists with switchover valves at these high temperatures as maximum blast stove furnace preheat temperatures are in the 2500°F temperature range. However, two six inch water cooled, refractory line gate valves, manufactured by S.P. Kinney Engineer, Carnegie, Pa. has been operated in the AERL preheater system at temperatures over 3200°F and have functioned satisfactorily. Thus, although some upgrading of current switchover valve designs may be necessary, the practical design and operation of switchover valves at air preheat temperatures of 3200°F are in principle considered technical feasible. This has also been verified in discussions with S.P. Kinney Engineers, who is a leading manufacture of high temperature switchover valves for blast stove furnace preheater applications. A general arrangement drawing of large high temperature valves proposed by S.P. Kinney Engineers for service up to 3200°F and 6 atmospheres pressure is shown in Figure 2-127. These valves are refractory lined and water cooled and are designed to open or close within 10 seconds. For service below 1200°F, uncooled gate and/or butterfly valves will be used to minimize heat loss.

Listed in Table 2-39 is a summary of the valve specifications.

2.11.6.4.2 Heat Loss

For the air preheater system the heat loss of the high temperature valving becomes important. Analytical treatment is complex and uncertain so that the actual heat loss for any given design can only be determined experimentally under actual operating conditions. However, experimental data from the high temperature gate valves in the AERL experimental preheater

Figure 2-127 High Temperature Gate Valve

TABLE 2 - 39
MAJOR VALVE SPECIFICATION

<u>Location</u>	<u>Case</u>	<u>Design Operating Temp.</u> °F	<u>Operating Medium</u>	<u>Valve Dia.</u> in.	<u>Valve Type</u>	<u>No. of Valves</u>
Low Temp. Air Inlet	I	500	Air	22	Isolation	4
	II	1100	Air	28	Isolation	4
High Temp Air Exit	I, II	3200	Air	42	Isolation	4
MHD Burner Air Shut off	I, II	3200	Air	42	Isolation	1
High Temp. Air to Stack	I, II	3200	Air	42	Isolation	1
High Temp. Air Mixer/Bypass	I, II	3200	Air	22	Control	1
Combustion Gas Exit	I	850	Comb. Gas	58	Isolation	4
	II	1450	Comb. Gas	68	Isolation	4
Pressurize/ Depressurize	I	600	Air	16	Isolation	6
	I, II	600	Air	16	Control	2
	II	1200	Air	22	Isolation	4
	II	600	Air	16	Control	2
Burner Air Inlet	I, II	1100	Air	56	Isolation and Control	4
Burner Fuel (Gas Inlet	I, II	1100	Gas	36	Isolation	8
Burner Fuel (Gas Inlet	I, II	1100	Gas	36	Control	4
Burner Recycle Gas Inlet	I, II	1100	Recycle Gas	28	Isolation and Control	4

facility and from a larger blast furnace stove hot blast valve⁽²⁷⁾ has provided valuable information for heat loss assessment. The following equation has been found to correlate well with the limited data available as shown in Figure 2-128.

$$\text{Nu} = .00206 \text{ Re}^{1.048} \text{ Pr}^{.3} \quad (2-56)$$

where

$$\text{Nu} = \text{Nusselt number} = \frac{hD}{k}$$

$$\text{Re} = \text{Reynolds number} = \frac{4\dot{m}}{\pi u D}$$

$$\text{Pr} = \text{Prantdl number} = \frac{uC_p}{k}$$

and

h = Heat transfer coefficient

D = Valve diameter

k = Thermal conductivity of fluid

$\dot{m}$ = Mass flow rate of fluid

u = Viscosity of fluid

C_p = Specific heat of fluid

The properties in the above equation are based on their bulk fluid temperature. For heat loss calculations, the heat transfer area is based on the valve flow diameter and width of valve body from flange to flange. The valve surface temperature is assumed to be that of the cooled metal. Shown in Figure 2-129 is the calculated heat loss through the 42-inch diameter high temperature air outlet gate valves for various air flow rates and temperature.

2.11.6.4.3 Valve Leakage

Air leakage will occur primarily across the seats of the large isolation valves. Based upon experimental data on air leakage across the seats of the high temperature gate valves in the AERL regenerative air preheater facility, the maximum total air leakage through the valves is estimated to be 1.85 lb/sec or approximately 1.5% of the total air flow. This is far below the maximum allowable of 5% in the design parameters.

2.11.6.5 Matrix Support Structure Design

The matrix support structure must be stable and movements from thermal expansion, and deformation from the dead load of the matrix and

Figure 2-128 Typical Data Correlation for Heat Transfer in High Temperature in High Temperature Gate Valves

Figure 2-129 Heat Loss Thru High Temperature Gate Valve

drag forces produced by thermal expansion of the matrix must be minimized. Excessive movement of the support structure can reduce the thermal performance by causing misalignment and possible blockage of the checker flues. Also under extreme conditions fracturing and subsequent failure of the entire matrix may result.

In the design of the support structure only uncooled metallic systems were considered. The design of an uncooled system becomes more difficult as the temperature increases due to decreasing creep, rupture and oxidation/corrosion resistance properties of alloys with increasing temperatures. The use of a cooled metallic or refractory support system would alleviate some of these problems. However, it leads to additional heat loss for a cooled system and to problems related to structural instability for a refractory support system. This together with the lack of operating experience with these alternative design possibilities limited our design considerations for the matrix support structure to uncooled metal or metal alloys designs.

The matrix support structure for each of the two cases (Case I and II) considered were investigated. The major difference between the two cases was the matrix support design temperature which was considered to be 850°F for Case I and 1450°F for Case II. The temperature difference between these two cases has an important effect on the choice of material and the size of the support structure.

The basic design concept for the support system is similar to that used in current blast furnace stove systems. In such a system the matrix rests on a number of small metal grids or grates with a given hole pattern. The grates are supported along their edges by horizontal girders which rest on vertical columns. Thus, by breaking the support system up into a number of small grates, expansion and contraction of each grate can occur independently and with minimum interaction with the plate adjacent to it.

The matrix support structure material is determined primarily by its mechanical strength and its oxidation/corrosion resistance properties at the design temperature. For Case I where the design metal temperature is 850°F, metal creep is not a significant problem. However for Case II, with a maximum metal temperature of 1450°F, the structure must consider creep in addition to rupture strength which can lower the allowable design stress substantially. For Case I and II the following alloys and design stresses were considered as shown in Table 2-40.

TABLE 2-40

MATRIX SUPPORT MATERIAL SELECTION

	<u>Case I</u>	<u>Case II</u>
Design Temperature	850°F	1450°F
Material Alloy	SA-240 Grade 304	Inconel 617
Stress Rupture Strength (100,000 hrs)	38,000 lb/in ² (est)	8000 lb/in ²
Design Stress	10,300 lb/in ²	2000 lb/in ²

Shown in Figures 2-130 and 2-131 are arrangement drawings of the matrix support assembly. The total assembly consists of 16 individual grates which are supported on all sides by girders which in turn rest on vertical columns. A keying arrangement between the girders and support columns fixes the position of the girders within given limits. The support columns are insulated with refractory to minimize their expansion and contraction during each heater cycle.

Between the grate and horizontal girders is a 1 1/2 inch thick spacer plate with machined slots to ensure free flow through all the checker flues. Instead of machining slots in the main grate, a separate spacer plate was used to eliminate the stress concentration which would occur in the slot. The matrix load is designed to be carried by the main grate alone with no support from the spacer plate. The entire matrix support system is independent of the heat exchanger vessel shell. Thus, no breaks are required in the side wall insulation for the matrix support.

The support system was divided into 16 segments in order to limit movement due to thermal cycling to an acceptable level and to hold the plate thickness within standard mill sizes. With this arrangement, the maximum expansion from ambient to its maximum temperature and the normal expansion and contraction during each cycle is .352 in. and .072 in. respectively for Case I, and .515 in. and .065 in. respectively for Case II.

In calculating the required grate thickness, the hole pattern in the grate must be accounted for. Due to the holes, high stress concentration are developed in the material between the holes (ligament stresses). In addition, a perforated grate is more elastic than a solid plate and will deform more under the same load. The thickness of the grates were calculated from modified flat plate equations for stress and deflection as developed by

Figure 2-130 Matrix Grate Support Assembly

Figure 2-131 Matrix Grate Support Assembly

G. Horvay⁽⁴⁶⁾ for stress analysis of perforated plates. Based on this analysis, the designed plate thickness for Case I is 3.0 inches and for Case II, 6.0 inches. The detailed analyses for the grates, horizontal girders and support columns follows.

2.11.6.5.1 Matrix Support System Design Analysis

1. Mechanical Properties of the elected Material

The stress-rupture and the oxidation/corrosion resistance properties at the design temperatures determine the selection of the material.

Design temperature	850°F	1450°F
Material selected	SA-240 Grade 304	Inconel 617
Stress rupture strength (100,000 hours)	38,000 psi (est)	8000 psi
Design stress	$s_m = 10,300$ psi	2000 psi
Modulus of elasticity at design temperature	$E = 26 \times 10^6$ psi	24.5×10^6 psi
Thermal expansion coefficient	$\alpha = 9.85 \times 10^{-6}$ in/in/°F	8.45×10^{-6}
	<u>Case I</u>	<u>Case II</u>

2. Matrix Support Structure Load

The following inputs were used to calculate the dead load the support structure must sustain

Matrix Diameter,	D = 15 ft	15 ft
Matrix Height,	H = 85 ft	65 ft
Matrix Density (40% porosity),	$\rho = 120$ lb/ft ³	120 lb/ft ³

The total weight the support structure must sustain is

$$W_o = \frac{\pi}{4} D^2 H \rho \quad = 1.8 \times 10^6 \text{ lbs} \quad 1.378 \times 10^6 \text{ lbs}$$

Case ICase II

The unit load on the matrix support is

$$\omega_o = H\rho \quad = 1.02 \times 10^4 \text{ lb/ft}^2 \quad 7800 \text{ lb/ft}^2$$

A safety factor of 1.5 was applied to account for additional loading of the support structure due to friction between insulation and matrix.

Therefore, for design purposes the following loads were used

$$\begin{aligned} \text{and } W &= 1.5 W_o &= 2.7 \times 10^6 \text{ lbs} & 2.07 \times 10^6 \text{ lbs} \\ \omega &= 1.5 \omega_o &= 106.25 \text{ lb/in.}^2 & 81.25 \text{ lb/in.}^2 \end{aligned}$$

The support structure design calculations were based on the following dimensions

$$\begin{aligned} \text{Typical grate size,} & &= 42 \text{ in.} \times 42 \text{ in.} & 42 \text{ in.} \times 42 \text{ in.} \\ \text{Dimension of grate support} & & & \\ \text{(distance between center-} & & & \\ \text{line of girders),} & &= 34 \text{ in.} \times 34 \text{ in.} & 34 \text{ in.} \times 24 \text{ in.} \end{aligned}$$

Then, the load per typical grate is

$$W_{\text{grate}} = A_{\text{grate}} \omega \quad = 1.87 \times 10^5 \text{ lb} \quad 1.43 \times 10^5 \text{ lb}$$

the load per typical support girder is

$$W_{\text{girder}} = 1/2 A_{\text{grate}} \omega \quad = 9.35 \times 10^4 \text{ lb} \quad 7.14 \times 10^4 \text{ lb}$$

and the load per typical support column is

$$W_{\text{column}} = A_{\text{grate}} \omega \quad = 1.87 \times 10^5 \text{ lb} \quad 1.43 \times 10^5 \text{ lb}$$

3. Support Column Design

Stress Analysis

The required cross-section area of a support column due to the compressive load is

$$A_{\text{column}} = \frac{W_{\text{col}}}{s_m} \quad = 18.16 \text{ in.}^2 \quad 71.5 \text{ in.}^2$$

Then, for a column with an outside diameter of

$$\text{O. D.}_{\text{col}} = 7.0 \text{ in.} \quad 12.0 \text{ in.}$$

the inside diameter is

	<u>Case I</u>	<u>Case II</u>
$I.D. = \left[(O.D.)^2 - \frac{4}{\pi} A_{col} \right]^{1/2}$	= 5.087 in.	7.28 in.

The following column inside diameters were selected

I.D. = 5.0 in.	7.0 in.
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Stability Analysis

To check the column for buckling, the buckling load, "P" is calculated.

The moment of inertia of the column is

$I = 1/4 \pi (R_1^4 - R_2^4)$	= 87.18 in. ⁴
-------------------------------	--------------------------

	900 in. ⁴
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Therefore, the buckling load, P, for a column length, "l", of 8 ft is

$P = \frac{\pi^2 E I}{4l^2}$	= 6.07 x 10 ⁵ lb
------------------------------	-----------------------------

	5.78 x 10 ⁶ lb
--	---------------------------

The safety factor for buckling is

$\eta = \frac{P}{W_{col}}$	= 3.25
----------------------------	--------

	40.42
--	-------

4. Support Girder Design

Stress Analysis

	<u>Case I</u>	<u>Case II</u>
Where L is the length of the girder between centerline of supports the maximum bending moment is	L = 30.5 in.	30.5 in.
Max M = 1/8 W _{girder} L	= 3.56 x 10 ⁵ in./lb	2.72 x 10 ⁵ in./lb
Assuming a thickness B/2 for the individual girder beams the height "H" of the girder is	B/2 = 2 in.	3 in.
$H = \left(\frac{6 \text{ Max } M}{s_m B} \right)^{1/2}$	= 7.2 in.	11.66 in.
The following girder heights were selected	H = 8.0 in.	12.0 in.

Stability Analysis

The deformation, y, (bending) of a girder due to the applied load is found from

$$\text{Max } y = - \frac{5 W_{\text{girder}} \ell^3}{384 E I}$$

where

$$I = \frac{1}{12} B H^3 \quad \quad \quad = 170.67 \text{ in.}^4 \quad \quad \quad 864 \text{ in.}^4$$

$$\text{Therefore,} \quad \quad \quad \text{Max } y = 1.08 \times 10^{-2} \text{ in.} \quad \quad \quad 1.726 \times 10^{-3} \text{ in.}$$

Thermal Expansion Analysis

The linear axial thermal expansion of a typical girder during initial start up and during a normal cyclic operation is

$$\Delta L = \alpha L \Delta T$$

During initial startup	$\Delta T = 850^{\circ}\text{F}$	1450 ^o F
	L = 34 in.	34 in.
	$\Delta L = .285 \text{ in.}$	.417 in.
During normal cyclic operation	$\Delta T = 173^{\circ}\text{F}$	183 ^o F
	L = 34 in.	34 in.
	$\Delta L = .058 \text{ in.}$	.053 in.

Case I

Case II

5. Perforated Plate Calculations

Stress Analysis

The hole pattern in the perforated plate is designed to match the hole pattern in the checker work.

$D = 1.0 \text{ in.}$	1.0 in.
$2h = 0.5 \text{ in.}$	0.5 in.
$2R = 1.5 \text{ in.}$	1.5 in.

According to Horvay the maximum stress in a perforated plate can be found using a correction factor, y , which is a function of the solidity ratio

where	$\eta = \frac{2h}{2R}$	$= 0.33$	0.33
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The maximum allowable ligament stress s_{lig} is equal to the design stress s_m

$s_{lig} = s_m$	$= 10,300 \text{ psi}$	2000 psi
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From Roark⁽⁴⁷⁾ the maximum stress in a solid plate is

$s_{solid} = \beta \frac{\omega b^2}{t^2}$	$y(\eta) = 3$	3
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and from Horvay where the plate thickness t is

$t = \left(\beta \frac{\omega b^2 y}{s_m} \right)^{1/2}$	$= 3.2$	6.35
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Because of very conservative assumptions, the following standard plate thicknesses were selected

$t = 3.0 \text{ in.}$	6.0 in.
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Case ICase IIStability Analysis

According to Horvay, displacements in a perforated plate may be calculated as for a solid plate using modified elastic constants E^* and ν^* .

For a plate size (between centerlines of supports)

= 34 in. x 34 in.

34 in. x 34 in.

and from Roark, the deformation, y , of a solid plate is

$$\text{Max } y = \alpha \frac{\omega b^4}{E t^3}$$

From Horvay for a perforated plate with a $\eta = 0.33$

$$E^* = 0.24 E$$

$$= 6.24 \times 10^6 \text{ psi}$$

$$5.88 \times 10^6 \text{ psi}$$

Therefore,

$$\text{Max } y = \alpha \frac{\omega b^4}{E^* t^3}$$

$$= 3.73 \times 10^{-2} \text{ in.}$$

$$3.78 \times 10^{-3} \text{ in.}$$

Thermal Expansion Analysis

The thermal expansion of a typical plate is

$$\Delta L = \alpha L_{\text{plate}} \Delta T$$

During initial start up for

$$L_{\text{plate}} = 42 \text{ in.}$$

$$42 \text{ in.}$$

and

$$\Delta T = 850^\circ\text{F}$$

$$1450^\circ\text{F}$$

the thermal expansion is

$$\Delta L = 0.352 \text{ in.}$$

$$0.515 \text{ in.}$$

During normal system cycling where the cyclic expansion and contraction is

$$\Delta T = 173^\circ\text{F}$$

$$183^\circ\text{F}$$

$$\Delta L = 0.0716 \text{ in.}$$

$$0.0649 \text{ in.}$$

2.11.6.6 Reheat Burner Design

The reheat combustion gas system for each heat exchanger is rated at 340 million Btu/hr thermal input from the fuel. The burners are designed for vertical firing and for dual fuel operation with either No. 2 fuel oil or low Btu gas produced from coal. In order to both control the flame temperature and NO_x emission, the system is designed to recycle a fraction of the heater cooled exhaust gases. The recycled gases are introduced into the combustion chamber through an injection ring arranged around the outer periphery of the burner. By directing the recycled gases parallel to the flame, they also serve to partially blanket the flame from the combustion chamber walls. In order to reduce the fuel requirement the combustion air and recycled gas are pre-heated to 1100°F prior to entering the burner.

The general arrangement of the reheat burner as proposed by Bloom Engineering Company, Pittsburgh, Pa. is shown in Figure 2-132. The burner is designed to operate with a turn down ratio of 7:1 on oil and 10:1 with gas and with up to 150-200% excess air. When operating with fuel oil, 100 psi steam is considered used for atomization.

The design of the combustion chamber and its size and aspect ratio (L/D) are basically determined by the burner type (diffusion, pre-mixed), its specific heat release rate and gas velocity which give stable and complete combustion. Stable combustion is an important criterion. Pulsations in blast furnace stove burner systems are known to occur under certain flow conditions. When the pulsating frequency approaches the resonant frequency of the combustion chamber and associated system, serious damage to the system may occur.

An important consideration in the design of the combustion chamber has to do with the allowable compressive load on the refractory insulation. Whereas in the checker chamber, the compressive load increases as the refractory temperature decreases, the refractory in the combustion chamber is subjected to a relatively constant high temperature over the entire length.

Figure 2-132 Pressurization-Depressurization Air Heat Storage Vessel

Therefore, in order not to exceed the compressive load of the refractory, the length of the combustion chamber becomes important. Furthermore, the surface area of the combustion chamber should be minimized to minimize heat loss.

For burning of either No. 2 fuel oil or low Btu coal gas, Bloom Engineering estimates that their burners will give a flame length of 42 feet and a flame diameter of 10 feet. The recommended combustion chamber size is 12 feet in diameter and 48 feet in length. Although detailed analysis would be required, excessive compressive creep of the insulation at the bottom of the combustion chamber is not expected to occur for the combustion diameter dimensions.

In the reheat combustion gas system separate shut-off valves is used for burner air, fuel gas and recycle gas. For the fuel gas, two valves in series are used. This fuel gas valve arrangement is critical because any leakage during the blowdown cycle would result in high pressure and temperature air entering the fuel gas stream. To safe guard against this, a second valve is used with a suitable vent between the two.

2.11.6.7 Auxiliary Pressurization/Depressurization

Pressurization and depressurization of the heat exchanger vessels is required prior to heater switchover in order to equalize pressure across the valves. In this way the valves are not required to open against a large pressure differential. Furthermore, fluctuations in the pressure and mass flow rate of the high temperature air are minimized.

The pressurization/depressurization systems are shown schematically for Case I and II in Figures 2-111 and 2-112. Rather than the use of a full capacity air compressor for intermittent operation as each vessel is pressurized, a compressor-accumulator concept was used. This permits the use of a smaller compressor for continuously charging of the accumulator which is blown-down during pressurization of the heater vessels. The use of low temperature air for pressurizing the vessels may be damaging to the checkerwork and insulation. Therefore, for Case II where the temperature at the base of the matrix is relatively high, a regenerative heat recovery heat exchanger is added to the system in order to recover the heat energy which otherwise

would be lost during depressurization of a heater unit. The heat recovered is utilized for preheating the pressurization air during the next cycle. In addition to recovery of heat energy the preheating of the pressurization air reduces the thermal shock on the refractories at the bottom. The pressurization system is sized for pressurizing the heat exchangers to 6 atmospheres in approximately 4 minutes at a flow rate of 10 lbs/sec. The air accumulator vessel was assumed to be spherical with a diameter of approximately 21 ft. It would store 3900 lbs of air at 250 lb/in² pressure.

The basic design of the heat recovery heat exchanger is shown in Figure 2-133. The vessel is 30 ft in length and 6 ft in length with a 12 inch refractory insulation lining. The heat storage matrix is 20 ft in length and 4 ft in diameter. The checker brick holes are 1/2 in. in diameter with 3/8 in. web thickness. The checkers are made of the same alumina as the main heat exchanger checkerwork.

2.11.6.8 Controls and Instrumentation

Controls and instrumentation are an important items for safe and reliable operation of the air preheater system. The control system will incorporate manual and automatic regulation of the pressure, temperature, and mass flow rate of the blowdown cycle air, reheat combustion gas, and pressurization/depressurization air. The system will also incorporate both manual and automatic control of heater switching.

Listed below are the various control systems and instrumentation which will be incorporated into the system.

2.11.6.8.1 Controls

a. Reheat Burner Control System

The reheat combustion control system will regulate the mass flow rates and mixture ratios of the fuel, combustion air and recycle gas so as to satisfy the energy balance for preheating the blowdown cycle air to the desired temperature. Shown in Figure 2-134 is a diagram of the combustion control system.

The combustion control system can be broken down into 6 distant functions.

1. Basic reheat burner air flow rate control.
2. Basic fuel/air and recycle gas/air ratio control.

Figure 2-133 Heat Recovery Heat Exchanger

Figure 2-134 High Temperature Air Preheater Combustion Gas Control System

3. Preheat air temperature control to fine tune combustion air flow rate.
4. O_2 control to fine tune fuel flow rate.
5. Dome temperature control to fine tune recycle gas flow and provide protection against overheating dome, top portion of matrix and other heater components.
6. Stack temperature control to provide protection against overheating of matrix support and other various heater components.

In the reheat combustion system, the prime set point is the reheat combustion air flow. The nominal combustion air flow rate set point is determined by an energy balance based upon the blowdown cycle air flow rate, desired preheat air temperature, and cycle time. To provide for automatic control of the preheat air temperature, the nominal combustion air flow rate set point is biased by the difference between the actual and desired preheat air temperature. The combustion gas exhaust temperature from the preheater is also measured but is normally not controlled unless it exceeds a given set point temperature. However, if the combustion gas exhaust temperature does exceed the set point temperature, this control circuit will override the preheat air temperature bias and adjust the combustion air flow rate set point to bring this temperature back into line. For this case, the combustion gas flow rate is stack temperature limited and thus the desired preheat air temperature will not be achieved unless other operating parameters such as the cycle time or dome temperature are changed or the heater design modified as required to lower the combustion gas exhaust temperature.

The nominal fuel and recycle gas flow rates are determined based upon the measured combustion air flow rate, and the nominal fuel/air and recycle gas/air ratios. For fine control of the amount of excess air, a O_2 content controller is used to bias the fuel/air ratio set point. The dome temperature is controlled by the amount of recycled gas in the combustion products. Fine control of the heater dome temperature is done by the dome temperature controller which is used to bias the nominal air/recycle gas set point. The heater dome temperature is measured by a radiation pyrometer sighted on the top checkers.

The reheat burner system will also include on line gas analysis of nitrogen oxides. Emission control of nitrogen oxides becomes increasingly important with increase of the air preheat temperature since this requires higher combustion temperatures which promote the formation of nitrogen oxides during combustion of the fuel. Safety interlocks will also be incorporated into the reheat burner system to monitor burner ignition and stability throughout its operating envelope.

b. Blowdown Air Control System

During normal plant operation, the blowdown cycle air flow rate will be controlled by the MHD burner control system. However, the blowdown air control system will also provide for independent regulation of the flow rate and pressure of the blowdown air to the regenerative heat exchangers as required for warm up, shut down, or individual testing of the heater.

c. Preheated Air Temperature Control System

The preheated air temperature control system will provide a constant temperature of the preheated air to the MHD burner by mixing a required amount of cooler bypass air from the compressor with the preheated air.

d. Auxiliary Pressurization/Depressurization Air Control System

The auxiliary pressurization/depressurization air control system will regulate the pressure and flow rate of air for pressurization of the heat exchangers prior to going on the blowdown portion of the cycle and the rate of depressurization prior to gas reheat.

e. Cycle Sequence Control System

A process sequence controller will be used to regulate the system cycle sequence. It will be designed for automatic or manual operation and will incorporate safety interlocks in all components. In particular, the heater system will be interlocked so that a heater cannot be taken off air blowdown or gas reheat without another unit already on such a cycle. This will insure a continuous flow of preheated air and combustion products as required for the MHD power cycle. A typical air blowdown and gas reheat cycle would be as follows:

1. Initiate air blowdown cycle
 - a) Burner fuel and recycle gas valves closes.
 - b) After preset time delay to purge system, burner air valve closes.

- c) Combustion gas exhaust valve closes.
 - d) Pressurization valve opens to equalize pressure across valve.
 - e) Pressurization valve closes.
 - f) Blowdown cycle air inlet valve opens.
 - g) Blowdown cycle air outlet hot valve opens. The heater is now on air blowdown.
2. Initiate gas reheat cycle
- a) Blowdown cycle air inlet valve closes.
 - b) Blowdown cycle air outlet hot valve closes.
 - c) Depressurization valve opens until pressure in heater is reduced to desired level.
 - d) Combustion gas exhaust valve opens.
 - e) Depressurization valve closes.
 - f) Burner air valve opens.
 - g) Burner recycle gas valve opens.
 - h) Burner fuel valve opens. With the onset of burner ignition, the heater is now on gas reheat.

Under normal conditions, the combustion chamber brickwork should be hot enough to ignite the fuel. However, if a flame is not detected within a preset time, the main fuel valve will close and the control system will energize a pilot ignition system. After a pilot flame has been detected, the main fuel valve will again be opened. If the pilot and/or main flame now fails to ignite, an alarm will indicate a malfunction which will need correction.

2.11.6.8.2 Instrumentation

a. Temperature Sensors

<u>Device</u>	<u>Location</u>	<u>Maximum Temperature Capability</u>
T/C	Blowdown air to inlet of heaters	1400°F
Optical Pyrometer	High temperature blow-down air at heater outlet	3400°F
T/C	Bypass air to mixer	1200°F
T/C, Optical Pyrometer	Mixer discharge temperature	3300°F
T/C	Combustion air for reheat burners	1200°F
T/C	Low Btu fuel for reheat burners	1200°F

<u>Device</u>	<u>Location</u>	<u>Maximum Temperature Capability</u>
T/C	Combustion gas discharge at heater outlet	1500°F
Optical Pyrometer	Top of Bed	3400°F
Optical Pyrometer	Dome	3400°F
T/C	Support grate	1500°F
T/C	Pressurization air inlet	1200°F
T/C	Depressurization air inlet to auxiliary heat exchanger	1500°F
T/C	Bed temperature profile (axial and radial)	3000, 2000 1500, 1000°F
T/C	Insulation (axial and radial) in heat exchanger	3000, 2000, 1500, 1000, 500°F
Optical Pyrometer T/C	Insulation (axial and radial) in combustion chamber	3500 (Optical), 3000, 2000, 1500, 1000°F
T/C	Heater shell	800°F
T/C	Heater ducting	800°F

b. Pressure Sensors

<u>Device</u>	<u>Location</u>	<u>Pressure Range</u>
Pressure Transmitter	Blowdown air inlet from preheater	150 psi
Pressure Transmitter	Blowdown air inlet to heaters	150 psi
Pressure Transmitter	High temperature blowdown air outlet from heaters	150 psi
Pressure Transmitter	Bypass air to mixer	150 psi
Pressure Transmitter	Mixer outlet	150 psi
Pressure Transmitter	Reheat combustion air	5 psi
Pressure Transmitter	Reheat combustion fuel	5 psi
Pressure Transmitter	Reheat combustion chamber	5 psi
Differential Pressure Transmitter	ΔP across bed	5 psi
Differential Pressure	ΔP between high temperature air outlet and pressurization line	150 psi system 0 diff. switching

<u>Device</u>	<u>Location</u>	<u>Pressure Range</u>
Differential Pressure Transmitter	ΔP between combustion chamber and heater	150 psi system 0 diff. switching

c. Gas Analysis

<u>Device</u>	<u>Location</u>	<u>Type</u>
CO Gas Analyzer	Combustion gas exhaust	Infrared
CO ₂ Gas Analyzer	Combustion gas exhaust	Infrared
O ₂ Gas Analyzer	Combustion gas exhaust	Paramagnetic
NO _x Gas Analyzer	Combustion gas exhaust	Chemiluminescence

2.11.7 Waste Heat Recovery System

The exhaust gas from the high temperature regenerative air pre-heater system contains heat which is recovered in order to minimize stack losses and improve overall plant efficiency. The heat content of the exhaust gas is utilized partly for coal drying and partly for preheating the combustion air used in the combustion of the fuel. Shown in Figure 2-135 is the waste heat recovery system flow diagram used in conjunction with the reference HTAH (Case I) system. As shown in the flow diagram, exhaust gas from the HTAH at 700°F is split with ~ 120,000 lbm/hr going to the coal processing system and the remainder (about 250,000 lbm/hr) sent into the waste heat exchanger.

The waste heat exchanger is a Ljungstrom Case 3-type unit. This unit is designed for low temperature heat recovery in situations where sulphuric acid deposition and corrosion may be a problem. The Ljungstrom is placed on the suction side of the F.D. fan, warming 80°F ambient air to 428°F before it enters the F.D. fan. This arrangement was influenced by the low stack temperature desired, the pressure drop across the LTAH and HTAH, and the pressure limitation of the Ljungstrom unit.

The cooled exhaust gas at 280°F is then routed to the main plant stack except for 93,000 lbm/hr returned to the HTAH via the recycle gas fan and the LTAH. Bleed streams downstream of the two fans provide air and gas to the bottoming plant oil gun windboxes for startup and cooling during MHD operation.

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Figure 2-135 MHD-ETF Waste Heat Recovery System

For off-design operation during times when ambient air temperature is below 80°F, a steam coil incorporated in the Ljungstrom will prevent excess acid condensation. Leakage between the air and gas streams in the Ljungstrom is expected to be low due to the small pressure differential between the two gas streams.

Shown in Table 2-41 is the HTAH waste heat recovery instrumentation.

2.11.8 High Temperature Air Preheater Test Plan

The high temperature air heater test plan will provide for independent evaluation of the regenerative air preheater system. The manner of testing will be such that the preheater will be operated under conditions of increasing severity. The basic test plan and schedule are shown in Table 2-42 and Figure 2-136. As seen, operation of the preheater system is limited to a preheat of 2500°F during the first two years of operation. This is done to maximize reliability of the preheater system while still providing sufficient air preheat temperature to demonstrate acceptable overall plant performance. After evaluation of the overall plant has been completed for a 2500°F air preheat, the preheat temperature will be increased to 3000°F during the third and fourth years to take advantage of the additional power plant performance at this temperature.

2.11.9 Weight and Costs Estimate Breakdown

A detailed weight and cost estimate breakdown of the preferred system concept (Case I) is shown in Table 2-43.

TABLE 2-41

HTAH-WHR INSTRUMENTATION

<u>Measurement</u>	<u>Instrument or Method</u>	<u>Indication</u>	<u>Location</u>	<u>Equivalent Range</u>
<u>Temperature</u>				
Gas	C-CR T. C.	0/1/0	Ljungstrom Inlet	0-1000 F
	C-CR T. C.	0/1/1	Recycle Fan Outlet	0-1000 F
	C-CR T. C.	0/1/1	Ljungstrom Outlet to Stack	0-1000 F
Air	C-CR T. C.	0/1/1	F. D. Fan Outlet	0-1000 F
<u>Pressure</u>				
Air	Pressure Transducer	0/1/1	F. D. Fan Outlet	0-10 psig
Gas	Pressure Transducer	0/1/1	Recycle Fan Outlet	0-10 psig

TABLE 2-42
HTAH TEST PLAN

YEAR 1

1. Perform initial checkout and cold flow preheater system.
2. Initiate warm up of system for 2000°F air preheat.
3. At 2000°F air preheat and at 50% MCR, operate system under following order of cycle sequences:
 - a) 2 blowdown, 1 reheat, 1 switching
 - b) 1 blowdown, 2 reheat, 1 switching
 - c) 1 blowdown, 1 reheat, 1 switching
4. Same as 3 except at 75% MCR.
5. Same as 3 except at 100% MCR.
6. Repeat 3, 4, and 5 for 2500°F air preheat.

YEAR 3

7. Repeat 3, 4, and 5 for 3000°F air preheat.

YEAR 4

8. Convert system for operation with low Btu fuel gas and repeat 7.

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Figure 2-136 HTAH Test Schedule

TABLE 2-43

REGENERATIVE AIR PREHEATER WEIGHT (TONS) AND COST (1,000\$)
ESTIMATE ETF (4 HEATER SYSTEM)

Case I
Low Temperature
500°F Air Inlet

	<u>Weight</u> (tons)	<u>Material and</u> <u>Shop Lab</u>	<u>Field</u> <u>Lab</u> (1,000\$)	<u>Total</u> (1,000\$)
<u>Vessel:</u>				
Matrix	3,318	5,972	1,080	7,052
Vessel Insulation	4,337	5,134	1,902	7,036
Steel Shell	747	1,405	927	2,332
Matrix Support	<u>128</u>	<u>2,313</u>	<u>128</u>	<u>2,441</u>
Sub Total	8,530	14,824	4,037	18,861
<u>Reheat Burner:</u>				
Comb. Chamber Insul.	2,087	2,506	809	3,315
Steel Shell	247	464	306	770
Burner	<u>30</u>	<u>560</u>		<u>560</u>
Sub Total	2,364	3,530	1,115	4,645
<u>Ducting:</u>				
Insulation	624	667	451	1,118
Piping	184	460	152	612
Valves	<u>387</u>	<u>1,812</u>		<u>1,812</u>
Sub Total	1,195	2,939	603	3,542
<u>Instrumentation Controls:</u>		400		400
<u>Manufacturing Engineering:</u>		1,000		1,000
<u>*Auxiliary Equipment:</u>	<u>100</u>	<u>230</u>		<u>230</u>
Total	<u>12,189</u>	<u>22,923</u>	<u>5,755</u>	<u>28,678</u>

* Does not include combustion air burner blower, foundations or overhead cranes.

2.12 LBTU GASIFIER (ALTERNATE CLEAN FUEL FOR HIGH TEMPERATURE AIR HEATER)

2.12.1 Performance and Tradeoff Analysis

The clean fuel requirement for the separately fired air heaters constitute between 25% and 33% of the total fuel requirement for the plant. In the case of an oil-fired air heater this amounts to between 45,000 and 60,000 gallons of oil a day in addition to the main coal requirements. The option of providing the clean fuel for the separately fired heater as low Btu gas generated by the gasification of part of the main coal supply has the important attraction of making the plant dependent on a single source of energy; coal. The use of coal gas to fire the air heaters has the further potential of providing an additional use of the MHD generator and bottoming plant exhaust gases for gasifying the coal and/or preheating the fuel gases prior to combustion. Thus it has the potential of improving the overall process efficiency.

In evaluating the various low Btu gasification processes available or under development for utilization in the ETF design, four main performance criteria were identified and compared in the trade-off analysis.

1. Reliability/Experience. Coal conversion processes can generally be divided into two categories based on their state of development: those processes that were in commercial operation as town gas and industrial gas plants prior to the availability of large volumes of natural gas and those which are now under development for the purpose of filling the void anticipated with the projected natural gas shortage. Since the ETF is directed at demonstrating the MHD process it should not be encumbered with untried subsystem if avoidable. Therefore, consideration of processes to convert coal to low Btu gas for the ETF were limited to processes with established commercial experience. It was recognized, however, that these established processes are characteristically of low capacity, suitable for the ETF, but limited as far as commercial MHD plants are concerned. Some of the processes currently under development will have much higher thruputs and may be more suitable for a commercial sized MHD plant.

2. Low ash carry over. Since the refractories of the air pre-heater burner and regenerator are susceptible to attack by components of the coal ash, the concentration of ash particles in the fuel gas must be minimized. Since neither the concentration in the product stream nor the allowable concentrations are known at this time the approach taken was to select processes that by design should give the lowest carry-over. This approach generally favors the fixed bed processes which utilize large lumps of coal (1 to 4" in size) with little relative motion within the bed. It was considered that fluidized bed and entrained bed processes that depend on the reaction of finer air-borne particles would generally experience higher particle carry over.
3. High Coal Conversion Efficiency. The thermal efficiency $\left(\frac{\text{Btu of Product Gas}}{\text{Btu of Coal}} \right)$ of the coal conversion process should be high in order to obtain attractive economics and overall plant efficiencies.
4. Temperature of Product Gas. It is necessary to use the fuel gas at a high temperature in order to achieve high flame temperatures in the air heater burner. It is an advantage, therefore, if the fuel gas is produced at a high temperature and can be used without cooling. In the ETF application cooling should not be necessary since, with efficient sulfur removal in the main MHD combustion gases, cleaning of the low Btu fuel gas should not be required to meet EPA standards. Use of the fuel gas as it exits from gasifier, therefore, should be possible.

In the selection procedure the operating characteristics of fifteen low Btu gasification processes were determined and tabulated. The results of this effort, shown in Table 2-44, were used in the initial screening procedure with the final selection being based on subsequent communication with the selected process developers.

As Table 2-44 shows seven of the fifteen processes reviewed could be eliminated on the basis that they had not been operated commercially

TABLE 2 - 44

LOW BTU GAS PRODUCERS

(Refs. 1-5)

* Commercial Size

** Projected Commercial Size

Type	O ₂ /Air	Prod. HHV	Prod. Temp.	Pressure	Efficiency		Status		Private Communication
			°F		Cold %	Hot %			
<u>Fixed Bed</u>									
Lurgi	x/x	285/196	700	23-28	63	>90	Comm.	* 12' ϕ ~ 800 TPD [18 plants]	x
Wellman-Galusha	-/x	-/168	1200	1	75	93	Comm.	* 10' ϕ ~ 84 TPD [over 150 plants]	x
WD/GI	x/x	280/175	1000	1	77		Comm.	* 12' [over 100 plants]	x
Wilputte	-/x	-/170	1000	1	75	90	Comm.	10' ϕ - 50 TPD [12 plants]	x
<u>Fluidized Bed</u>									
Battelle/Carbide	-/x	—	1700	6	—		Bench	—	
BCR	-/x	-/1600	2000	15	88		PDU		
CO ₂ Acct.	-/x	-/380	1500	10	77		Pilot	30 TPD ~ 4' ϕ x 70'	
U-Gas	-/x	-/154	1700	23	79		Pilot	6 TPD ** 22' ϕ x 50' ~ 3000 TPD	
Westinghouse	-/x	-/135	1200	14	72	(94)	PDU	PDU ~ 1200 lbs/hr. (14 TPD)	x
Winkler	x/x	288/126	1500	1	72		Comm.	*18' ϕ x 75' ~ 1100 TPD (16 plants - 3 still in operation)	x
<u>Entrained Bed</u>									
B&W	x/x	301-102	1800	1-20	74	(95)	Semi Comm.	** 7.5' ϕ x 220' ~ 3000 TPD at 175 psig. [duPont - ran 1 400 TPD plant for 1 yr.]	
C-E	-/x	-/127	1600	1	69		Pilot	9' ϕ x 79' ~ 120 TPD	
Foster-Wheeler	-/x	-/177	2000	23	—		Pilot Design	480 TPD	
K-T	x/-	286/-	2700	1	75		Comm.	*4 head gasifier ≈ 860 TPD (13 plants)	x
Texaco	x/-	253/-	500	23	—		Comm.*	*Common. on Liq. Fuel ~ 70 plants **9' ϕ x 15' ~ 1900 TPD (650 psig) (coal)	x

References for Table 2-37

1. "The Gasification of Coal," L.K. Mudge, G.F. Scheifelbein, C.T. Li, and R.H. Moore, A Battelle Energy Program Report July 1974
2. "Synthetic Fuel Data Handbook," compiled by T.A. Henderson, Cameron Engineers, Inc., Denver, 1975
3. "Handbook of Gasifiers and Gas Treatment Systems," Dravo Corp., Pittsburgh, 1976.
4. "New Fossil-Fueled Power Plant Process Based on Lurgi Pressure Gasification of Coal," Paul F.H. Rudolph, presented at Joint Conf. of Chemical Inst. of Canada and the Am. Chem. Soc., May, 1970.
5. Private Communications as Indicated in Column

and thus were of questionable reliability at the time. An eighth process, the Babcock and Wilcox Entrained Bed, is classified as semi-commercial in that it did operate a commercial plant for one year a number of years ago, but this hardly compares with the experience associated with the operation of the other commercial processes. Texaco has had extensive experience gasifying liquid fuels with oxygen and is planning to operate one of its reactors on coal with air in the near future. As operating data will not be available in the immediate future, judgement on its applicability to MHD cycle will have to wait. The Koppers-Totzek process has been used extensively in Europe and Asian locations, but it is designed to function with oxygen and will not accommodate air without extensive modification. The Davy Powergas Co. has been active in developing the Winkler Process for the United States market. European experience of this process has been at atmospheric pressure and reportedly experiences high particulate carry-over. Davy Powergas representatives indicate, however, that operation at higher pressure will alleviate this condition and that they expect to demonstrate this in the near future. Relatively low carbon burn-out rates would require that char/ash residue be sent to the MHD combustor for highest efficiency.

The elimination of eleven of the fifteen processes for the reasons given in the preceding paragraph reduced the remaining processes under consideration to the four fixed bed designs. In comparing these processes it appears that because the Lurgi process operates at a somewhat lower temperature than the others, it produces more tars that require special clean-up procedures and costs. Additionally the Lurgi process operates at elevated pressures requiring the use of compressors and lock hoppers with their associated costs. The differences in the basic operating characteristics of the other fixed bed processes, Wilputte, Wellman-Galusha, and Woodall-Duckham, are not obvious from the available literature. It is judged that any of these processes would be suitable for producing sufficient clean gas for the ETF air heaters. Data exchanges with the McDowell-Wellman Co., developers of the Wellman-Galusha process, has demonstrated this company's technological competence and cooperation and has led us to select this process for initial ETF design purposes. It is recognized that

the limited thruput of the fixed bed designs may present problems in scaling up the ETF to a commercial size plant. It is projected, however, that the higher thruput low Btu gasification processes such as Texaco, Babcock and Wilcox, and Winkler will be fully demonstrated at that time and will be available for full size MHD plants. It is in fact, also recognized that these demonstrations may be completed by the time of ETF construction. In this case direct substitution of the more advanced process for the fixed bed design will be possible without appreciable impact on the overall process except for the simplification of the gasifier section itself.

In summary, as a result of the initial screening studies, the Wellman-Galusha gasification process was selected to produce clean fuel for the ETF air heaters for the following reasons.

1. It is a commercial process and does not involve any developmental uncertainties.
2. The Wellman-Galusha gasifier is currently specified for several DOE Low Btu gasifier programs so that a wealth of operating data on several coals and under a range of operating requirements should be available prior to ETF operation.
3. The requirement of the ETF are such that its low-ash fuel demands can be met by a battery of six Wellman-Galusha gasifiers. Such a battery has been designed for other applications so that the specifications were immediately available.
4. Since the process uses large lumps of coal, 1" - 2", it is projected to have a minimum of fine particles carry over.
5. Its hot gas thermal efficiency is projected to be greater than 90% with greater than 99% burnout of carbon from the ash.
6. The process delivers a hot product gas at approximately 1000⁰F suitable for burning in the air heater burner with little or no additional pre-heating required.*
7. When the product gas is used at the temperature at which it is produced no liquids condense and there should be no need for a liquid processing unit or clean-up.
8. The process operates with an air oxidizer and does not require an oxygen plant.

9. The McDowell-Wellman Engineering Co. were particularly responsive to inquiries.

2. 12. 2 Design

The design of the gasifier system is based on data exchanged between AERL, Inc. and the McDowell Wellman Engineering Co., the developer of the Wellman-Galusha gasifier. The basic cycle design incorporating a coal gasifier to generate an ash-free fuel for the air preheater was shown in Figure 1-2(b) in Section 1. 3. 2. 1. As the cycle diagram shows coal and air are fed to the gasifier to produce a low Btu gas which is then burned with preheated air (1100°F) to heat the refractories of the regenerative high temperature air preheater. The cycle arrangement as shown, employs recycle of some of the flue gases to the air heater when MHD combustion air preheat temperatures of less than 3000°F are considered.

The design basis for the low Btu gas system was to provide the same heat input to the high temperature air heaters as provided in the reference design which uses oil. As Table 1-16 (Section 1. 4. 1. 2) shows this requirement is 128 MW for 3000°F preheat case and 105 MW for the 2500°F case. The fuel requirements for these cases are 100 and 77 MW respectively. If the system is designed for the larger capacity requirement of the 3000°F preheat case the coal feed rate will be 495 tons/day. This is based on using Montana Rosebud Coal in the "as received" condition in a gasification process that has a hot gas efficiency of 90%. In order to reduce the quantity of water being carried through the gasifier and into the air heater burner, the coal will be dried, utilizing flue gas heat, to 5% moisture content. The composition of the dried coal is given in Table 2-45. The charge of dried coal to the gasifier will then be 409 tons/day. Processing of this quantity of coal will require six Wellman-Galusha gasifiers. A design package for combining the output of six gasifiers is shown in Figure 2-137.

The package consists of:

- 6 - 10 ft Agitator Type Gasifiers
- 6 - 8 ft Brick-lined Cyclones
- 1 - 70 TPH Bucket Elevator
- 1 - 92 ft Conveyor

TABLE 2-45
MONTANA ROSEBUD COAL ANALYSIS*
(Dried to 5% moisture)

Proximate Analysis

Moisture	5.0%
Vol. Matter	36.1
Fixed Carbon	48.2
Ash	10.7

Ultimate Analysis

H	4.82%
C	64.08
N	0.97
O	18.39
S	1.05
Ash	10.7

*See Previous Table for "as received" Analysis.

Figure 2-137 Six Wellman-Galusha Gasifier Units for Production of Lbtu Gas for Separate Firing of the High Temperature Air Preheater System.

- 1 - Tripper
- 1 - Automatic Flop Gate
- 4 - 42" Expansion Joints
- 6 - 24" Shutoff Valves
- 6 - Control Panels with Instrumentation
- 1 - Building

A sketch of an individual Wellman-Galusha gasifier with its attached cyclone is shown in Figure 2-138. The gasifier is water-jacketed and consequently does not require a refractory lining. The speed and position of the agitator, which is also water-cooled, can be controlled, depending on coal characteristics and feed rate, to maintain a uniform coal bed. A revolving step-type grate is mounted at the bottom of the gasifier to distribute the air-steam blast throughout the coal bed and to force the ash into the ash bin.

In operation, crushed coal is fed to the coal bin and feeding compartment. Coal flow in and out of the feeding compartment is maintained by gravity and controlled by four side valves. Input air is passed over the water in the water-jacket to pick up moisture required for the air-steam injection. Saturation of the air with moisture is regulated by adjusting the temperature of the water in the jacket and the air flow. The air-steam blast is distributed through the grate, into the ash zone. As it passes upward, combustion and gasification reactions occur generating a product gas rich in CO , H_2 , CO_2 , and N_2 . The hot gas dries and preheats the incoming coal before leaving the gasifier. Ash is withdrawn through the eccentric grate and collected in the ash bin, for subsequent disposal.

Gas leaving the gasifier reactor is passed through a cyclone to remove dust particles. The composition of this product gas is given in Table 2-46.

McDowell-Wellman calculate that in the case of Montana Rosebud coal, dried to 5% moisture, the gas leaving the cyclone will be between 630 and 680°F. This temperature is lower than that indicated in the initial screening study, which was based on the results produced using bituminous coal. It is assumed, however, that the moisture and oxygen content of the lower grade Rosebud coal will have a similar effect in reducing the product

WELLMAN-GALUSHA AGITATED GASIFIER

Figure 2-138 Wellman-Galusha Agitated Gasifier

TABLE 2-46
PRODUCT GAS CHARACTERISTICS
(Montant Rosebud Feed)*

Composition (Vol. percent)

CO	24.43
CH ₄	2.93
CO ₂	4.40
H ₂	9.92
N ₂	47.76
H ₂ S	0.22
H ₂ O	10.36
Average Mol. Wt.	24.75
Product Gas HHV (wet basis) Btu/SCF	139.8
HHV of Tar Component	15.4
Product Gas Temperature (°F)	630-680
Heat Capacity (dry gas) (Btu/SCF)	.0196
Thermal E	
Thermal Efficiency $\left(\frac{\text{Btu of Product}}{\text{Btu of Coal}} \right)$	> 91%

*Dried to 5% moisture

gas temperature of the other processes. Calculations were performed at AERL to indicate the level of preheating of both the fuel gas and combustion air needed to achieve the necessary flame temperature in the burner of the regenerative air heater. The results of these calculations for cases using stoichiometric quantities of air as well as with 5% excess air are given in Figure 2-139. The data shows that if both the gas and air are preheated to 1000° , for example, and 5% excess air is used, an adiabatic flame temperature of 3500° can be attained. The flame temperature calculations also showed that if no additional heating is provided for the fuel gas and if the air is preheated to 1100°F an adiabatic flame temperature of 3450°F is indicated. Experience at AERL in operating high temperature air heaters shows that 3450°F should be sufficiently high for delivering air at $3000\text{-}3200^{\circ}\text{F}$ from the regenerative heaters. The latter arrangement, heating the combustion air to 1100°F and using the fuel gas at the temperature at which it exits from the gasifier, eliminates the need for a fuel gas heater.

The mass and heat balances for the high temperature air heater using low Btu fuel from coal gasification are given in Table 2-47 (this section) for preheat temperatures of 2500°F , 2900°F and 3000°F . For the highest preheat temperature of 3000°F no recycle gas is employed to maintain the required high flame temperature. For preheat temperatures of less than 3000°F recycle gas can be employed. In addition to increasing the mass flow this will also have the effect of lowering the flame temperature from the calculated level of 3450°F shown in Figure 2-139. For a preheat temperature of 2900°F the flame temperature is reduced to 3270°F by gas recycling. The option of achieving the higher flame temperature, however, can be exercised by eliminating the recycle gas and increasing the fuel requirements from 99 to 108 MW with a small increase in the combustion air flow. A trade-off study to compare the differences in cycle efficiency obtained for air preheats of 2900°F and 3000°F against the added fuel requirements of 9 MW will be made in order to select between the two options.

A third option, that of operating the cycle with an air preheat temperature of 2500°F is also considered. For this case, an adiabatic flame temperature of 2900°F was deemed to be sufficiently high. Again the calculated flame temperature was reduced to this level by the addition of recycle

Figure 2-139 Flame Temperature Upon Combustion of Preheated Low Btu Gas

TABLE 2-47

MASS AND HEAT BALANCES--H. T. AIR HEATER
WITH LOW BTU FUEL FROM COAL GASIFIER

<u>Mass Flow Rates:</u>		<u>2500°F</u> <u>Preheat</u>	<u>2900°F</u> <u>Preheat</u>	<u>3000°F</u> <u>Preheat</u>
Fuel	10 ³ x pph	108.07	132.10	143.98
Combustion Air	10 ³ x pph	182.65	223.53	243.65
Recycle Gas	10 ³ x pph	<u>96.91</u>	<u>32.00</u>	<u>-----</u>
Total	10 ³ x pph	387.63	387.63	387.63
<u>Heat Inputs</u>				
Fuel (HHV)	MW	82.09	99.00	108.30
Air (1100°F)	MW	13.49	16.76	18.30
Recycle Gas (1100°F)	MW	<u>7.52</u>	<u>2.50</u>	<u>-----</u>
		103.10	118.26	126.60
<u>Heat Outputs</u>				
MHD Cycle Air	MW	53.20	73.50	73.50
Heaters for Air + Recycle Gas	MW	26.30	18.76	27.10
Coal Drying	MW	4.50	4.50	4.50
Recycle Gas Heat	MW	3.20	1.70	1.70
<u>Stack Loss:</u>				
Sensible Heat	MW	2.90	4.10	4.10
Latent Heat	MW	4.50	6.00	6.00
Total Heat Losses from Air Heater System	MW	<u>8.50</u>	<u>9.70</u>	<u>9.70</u>
Total		103.10	118.26	126.60

gas. The reduction in the flow requirements, from 126.00 to 103.50 MW, was achieved by reductions in the fuel, air and recycle flows from that of the higher temperature cases. Mass flows and heat balances for the three options are given in Table 2-47 with gasifier operating requirements given in Table 2-48.

To conduct tests using Illinois No. 6 coal (instead of Montana Rosebud reference coal) the battery of six Wellman-Galusha gasifiers will be adequate. The coal feed rate will be 7% less than the comparable options using Montana Rosebud, due to the higher heating value of the Illinois coal. The volume of gas produced per lb of coal fed will be slightly higher in the case of the Illinois coal and the heating value will also be slightly higher, 149 Btu/SCF vs 140 Btu/SCF. Other operating parameters, i.e., air feed, cooling water and electric power are essentially the same for each coal, but the temperature of the product gas will be about 100°F higher in the case of Illinois No. 6. Coal analysis, dried to 5% moisture, product gas characteristics and gasifier operating requirements for utilizing Illinois No. 6 coal are given in Tables 2-49, 2-50 and 2-51, respectively.

2.12.3 Instrumentation and Controls

The gasifier output is directly controlled by metering the air input. Since measurement and control of air streams is highly developed technology it follows that the gasifier output is also readily controlled. The control loop is made sensitive to the demand for the product gas and any change in demand is transmitted to the air feed line through pressure transducers. The pneumatically operated valves in the air feed line thus respond to changes in demand in a response time of less than 1 sec.

The gasifier can be started from a dormant state in about 5 hrs and can be readily turned down to less than 25% of its normal capacity without affecting the quality of the gas. The gasifier can be banked (zero output) for a period of days with only a few minutes of air injection to maintain the combustion zone temperature.

Because of the versatility of the gasifier in regards to turn down and because of its ability to follow directly load demands no storage facilities for product fuel gas are required.

TABLE 2-48
TASK II DESIGN SPECIFICATIONS FOR COAL GASIFIER
(Montana Rosebud Feed)*

		2500°F Preheat	2900°F Preheat	3000°F Preheat
Coal Feed Rate to Gasifier		28.33	34.08	37.28
(dried to 5% Moisture)	$10^3 \times \text{pph}$			
Air Feed Rate	$10^3 \times \text{pph}$	76.29	91.78	100.40
Water Requirements	$10^3 \times \text{gph}$			
Recycle		21.45	25.56	27.96
Make-up		1.42	1.70	1.86
Electric Power	$10^3 \times \text{kWh}$	6.11	7.36	8.05
Number of Gasifiers		6	6	6

*Dried to 5% moisture

TABLE 2-49
ILLINOIS NO. 6 COAL ANALYSIS*
(dried to 5% moisture)

Proximate Analysis

Moisture	5.0%
Vol. Matter	39.6
Fixed Carbon	43.5
Ash	11.9

Ultimate Analysis

H	5.15%
C	65.07
N	1.25
O	13.20
S	3.44
Ash	11.89

See previous table for "as received" analysis

TABLE 2-50
PRODUCT GAS CHARACTERISTICS
(Illinois No. 6 Feed)*

Composition

CO	25.10%
CH ₄	2.76
CO ₂	3.77
H ₂	12.48
N ₂	47.29
H ₂ S	0.69
H ₂ O	7.92
Mol. Wt.	24.17
Product Gas, HHV, wet basis, Btu/SCF	148.5
HHV of Tar Component	16.3
Product Gas Temperature (°F)	784
Heat Capacity (dry gas) (Btu/SCF)	.0196
Thermal Efficiency $\left(\frac{\text{Btu Product}}{\text{Btu of Coal}} \right)$	> 91%

*Dried to 5% moisture

TABLE 2-51
GASIFIER OPERATING CONDITIONS
(Illinois No. 6 Feed)*

		<u>2500°F Preheat</u>	<u>2900°F Preheat</u>	<u>3000°F Preheat</u>
Coal Feed Rate	10 ³ x pph	26.47	31.84	34.83
Air Feed Rate	10 ³ x pph	71.46	85.97	94.04
Water Requirements	10 ³ x gph			
Recycle		20.04	23.88	26.12
Make-up		1.32	1.59	1.74
Electric Power	10 ³ x kWh	5.70	6.88	7.52
Number of Gasifiers		6	6	6

*Dried to 5% moisture.

The following is a list of gas producer instrumentation and control equipment required for one (1) unit. Duplicate equipment will be supplied for each of the six units included in the battery:

Instruments:

1. One (1) 100 psi water pressure gauge with 4-1/2 in dial, for the by-pass piping system.
2. One (1) single pen circular chart recorder for agitator water pressure, with pressure switch.
3. One (1) 7" scale angled indicating glass thermometer, range 30° to 240°F, safe to 300°F, for the saturation pipe.
4. One (1) saturation recording control thermometer, including circular chart graduated 50° to 200°F, 24 hrs/rev, two pens gas filled thermal system, automatic reset, integral four-position by-pass panel, bronze armored capillary, poly-vinyl covered copper bulb, combination filter and pressure regulator, and diaphragm operated control valve, for saturation system.
5. One (1) recording thermometer, including circular chart graduated 0° to 100°F, 24 hrs/rev, single pen, mercury filled thermal system, and bronze armored tubing for offtake temperature.
6. One (1) recording controller for offtake pressure including circular chart graduated 0 to 10 units, 24 hrs/rev, single pen, brass bellows sensing element, automatic reset, integral four-position by-pass panel, combination filter and regulator, to operate 12" butterfly control valve.
7. Five (5) manometers with 36" adjustable scales in tenths of inches.
8. Fuel bin high and low level indicators with individual warning lights.

Valves:

1. Ash valve and assembly, rack and pinion operated.
2. Saturation manual blast valve, gate type, in the saturation elbow.
3. Vent valve, gate type, double disc.
4. Four (4) disc valves, for the coal bin.
5. 12" standard pattern butterfly control valve, to use with recording pressure controller.
6. 1" size diaphragm control valve, for use with saturation recording control thermometer.
7. 1/2" gate valve, in water jacket overflow piping.
8. 3" flanged gate valve, with iron body and brass disc, in the water jacket overflow piping.
9. 1" globe valve, in the water inlet by-pass piping.
10. 3/8" globe valve, in the water inlet by-pass piping.
11. 1/2" globe valve, in the water inlet by-pass piping.
12. 1" angle disc valve, in the water inlet-by-pass piping.