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ADVANCED THERMIONIC TECHNOLOGY PROGRAM

Summary Report

October 1984

Work Performed Under Contract Nos. AC02-76ET11292
AC21-83MC20352

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

By
Thermo Electron Corporation
Waltham, Massachusetts

and

Rasor Associates, Inc.
Sunnyvale, California

Technical Information Center
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PREFACE

This report summarizes the progress made by the Advanced Thermionic Technology Program during the past several years. This Program, sponsored by the U.S. Department of Energy, has had as its goal adapting thermionic devices to generate electricity in a terrestrial (i.e., combustion) environment. The technology has previously been developed for astronautical applications.

The report is organized in four volumes, each focused as much as possible on the needs of a particular audience. Volume 1 contains Part A, the Executive Summary. This Executive Summary describes the accomplishments of the Program in brief, but assumes the reader's familiarity with the thermionic process and the technical issues associated with the Program. For this reason, Volume 1 also contains Part B, a minimally technical overview of the Advanced Thermionic Technology Program. It is suggested that readers just being introduced to the Program review both portions of Volume 1 before consulting the more technical volumes which follow.

Volume 2 (Part C) concentrates on the progress made in developing and fabricating the "current generation" of chemical vapor deposited hot shell thermionic converters and is addressed to those primarily concerned with today's capabilities in terrestrial thermionic technology. Volume 3 (Part D) contains the results of systems studies of primary interest to those involved in identifying and evaluating applications for thermionics. Volume 4 (Part E) is a highly technical discussion of the attempts made by the Program to push the state-of-the-art beyond the current generation of converters and is directed toward potential researchers engaged in this same task. These technical discussions are complemented with Appendices where appropriate.

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PART D

THERMIONIC POWER SYSTEMS

STUDIES

1. INDUSTRIAL COGENERATION

1.1 Introduction

Many energy conversion systems are presently available to provide co-generation, including steam turbines, gas turbines, and diesel engines. In almost every case they provide process heat in the form of hot water or steam. Yet, more than half of the heat required by industry is needed in the form of direct heat at temperatures typically between 650 K and 1900 K. This point is illustrated in Exhibit D-1.1, where 26 industrial processes representing 50% of the energy consumed by industry are compared [1].

Some of the direct heat required by these industries is generated by exo-thermic processes themselves. An example in the steel industry is the use of coke to reduce iron ore in blast furnaces. Much of the direct heat used by the steel industry, however, is produced by natural gas, coke gas, coal, or oil-fired burners. Similarly, the petroleum, cement, ethylene, and iron industries make substantial use of burners to generate the high-temperature gas needed by each process.

As a general rule of thumb, cogeneration technologies are most attractive to industries when those technologies naturally produce a ratio of electrical to thermal output which closely matches the demand within the industrial facilities themselves. Several of the industries which consume the largest amounts of energy have an electrical-to-thermal ratio of about ten percent, as can be seen in Exhibit D-1.1. This closely matches the electrical efficiency

EXHIBIT D-1.1

INDUSTRY ENERGY CONSUMPTION

INDUSTRY NAME	TOTAL* ENERGY CONSUMPTION (10^{12} Btu, 1975)	DIRECT HEAT REQ.** (PERCENT OF TOTAL THERMAL)	ELECTRIC/THERMAL** RATIO (E/T)
Steel	3140.4	92	0.07
Petroleum Refining	2854.4	80	0.03
Corrugated Paper	498.6	0	0.14
Cement	413.8	100	0.08
Ethylene	250.5	88	0.002
Glass Containers	140.0	100	0.11
Chlorine/Caustic	128.4	0	1.03
Folding Boxboard	109.1	0	0.16
Writing Paper	105.6	0	0.22
Motor Vehicles	103.7	27	0.31
Gray Iron Foundries	103.1	92	0.35
Styrene	91.0	16	0.01
Tires	76.1	0	0.38
Sawmills	72.0	0	0.10
Meat Packing	70.6	6	0.32
Alumina	70.5	41	0.11
Newsprint	62.0	0	0.68
Fabric Mills	49.5	9	0.95
Malt Beverage	46.8	5	0.13
Baking	41.3	63	0.24
LDPE	28.3	0	2.17
PVC	18.8	0	0.67
HDPE	15.8	0	0.89
Nylon	13.2	0	0.94
SB Rubber	12.3	25	0.10
Copper-Arbiter Proc.	0.7	0	0.34
TOTAL	8516.7		

* Based on 1975 national data
** Based on 1985 projections

of thermionic converters. Thermionic cogeneration has several other unique advantages relative to alternative technologies for cogeneration which should lead to a much broader application of cogeneration in industry. These advantages accrue from the much higher temperatures at which thermionic energy conversion takes place, its suitability for very small as well as large process heaters, and, of course, its production of direct heat rather than process steam. In fact, thermionics can even be coupled to more conventional cogeneration technologies (e.g., steam turbines) to extend their applicability to processes requiring a greater electrical-to-thermal ratio than either cogeneration technology alone can provide.

A generalized schematic of a thermionic cogeneration "burner module" (TCBM) is presented in Exhibit D-1.2 as it would be used to replace a conventional furnace. Ambient air at temperature T_3 is drawn past the collectors of a thermionic array. This air cools the thermionic collectors and is preheated to a temperature T_4. Fuel is mixed with the preheated air and burned in the combustor section, providing high-temperature combustion gases at temperature T_1 which enter the thermionic converter section. The air preheat significantly increases T_1. Heat is transferred from the combustion gases to the emitters of the thermionic converters, and the combustion gases leave at T_2, which is the gas temperature as it enters a furnace or heat exchanger for process heating.

Thermodynamically, the module takes heat Q_h from the combustion gases and rejects heat Q_c to the combustion air, producing electrical power with an efficiency, given by $(Q_h - Q_c)/Q_h$, which is on the order of ten percent for the current generation of converters. Because the voltage of this energy is

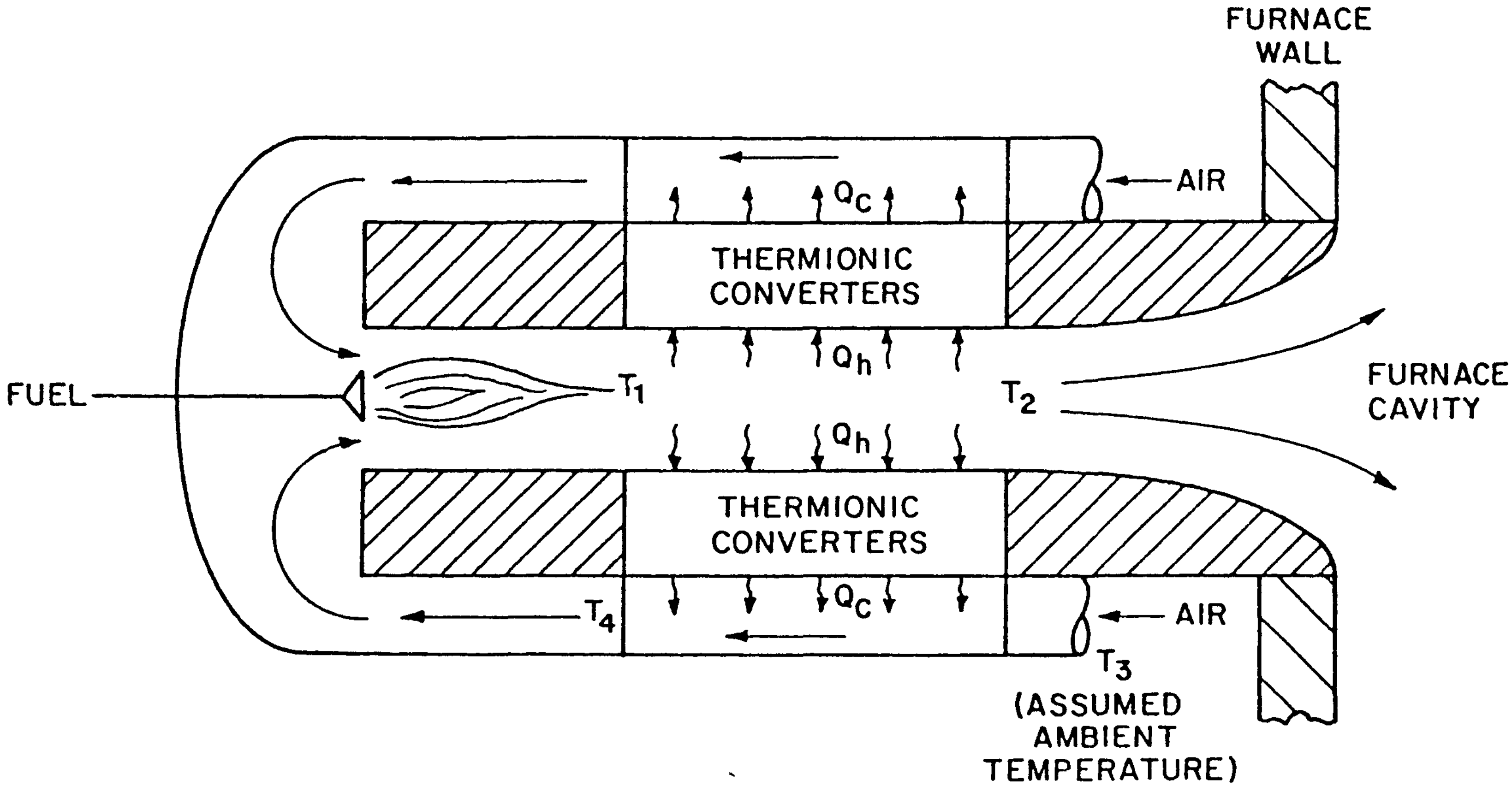


Exhibit D-1.2

Schematic of Thermionic Cogeneration Burner Module as a Replacement for a Conventional Burner controlled by the series-parallel connections within the thermionic array itself, the module is readily adapted to specialized voltage or current requirements (e.g., DC operation) of particular processes. However, integration of the array with the burner requires careful coupling of the heat transfer rates with the thermionic characteristics to maintain proper emitter and collector temperatures for efficient operation. In practice, air preheat temperatures above 1100 K or array outlet temperatures (i.e., T_2) below 1600 K are currently impractical.

In certain processes it is possible to add secondary air to the primary air used in the combustor. This secondary air can draw additional heat from the collectors without requiring a higher collector temperature, thereby improving overall system efficiency. This air is ultimately mixed with the combustion gases and fed to the industrial process. In addition to eliminating the air preheat temperature restriction on thermionic power, the use of secondary air also permits use of fuel-rich combustion in the thermionic converter with combustion completed by the secondary air prior to entering the furnace. This approach is useful in minimizing NO_x generation in the high-temperature combustion required for the thermionic converter.

Two specific design concepts for a thermionic cogeneration burner have been advanced, along with a limited number of designs applicable to particular industrial processes. Exhibit D-1.3 illustrates the first of the general designs.

In this design heat pipes lining the periphery of the combustion region collect heat and deliver it to thermionic converters at the rear of the combustor. By "integrating" the heat flux over the combustor length, the heat

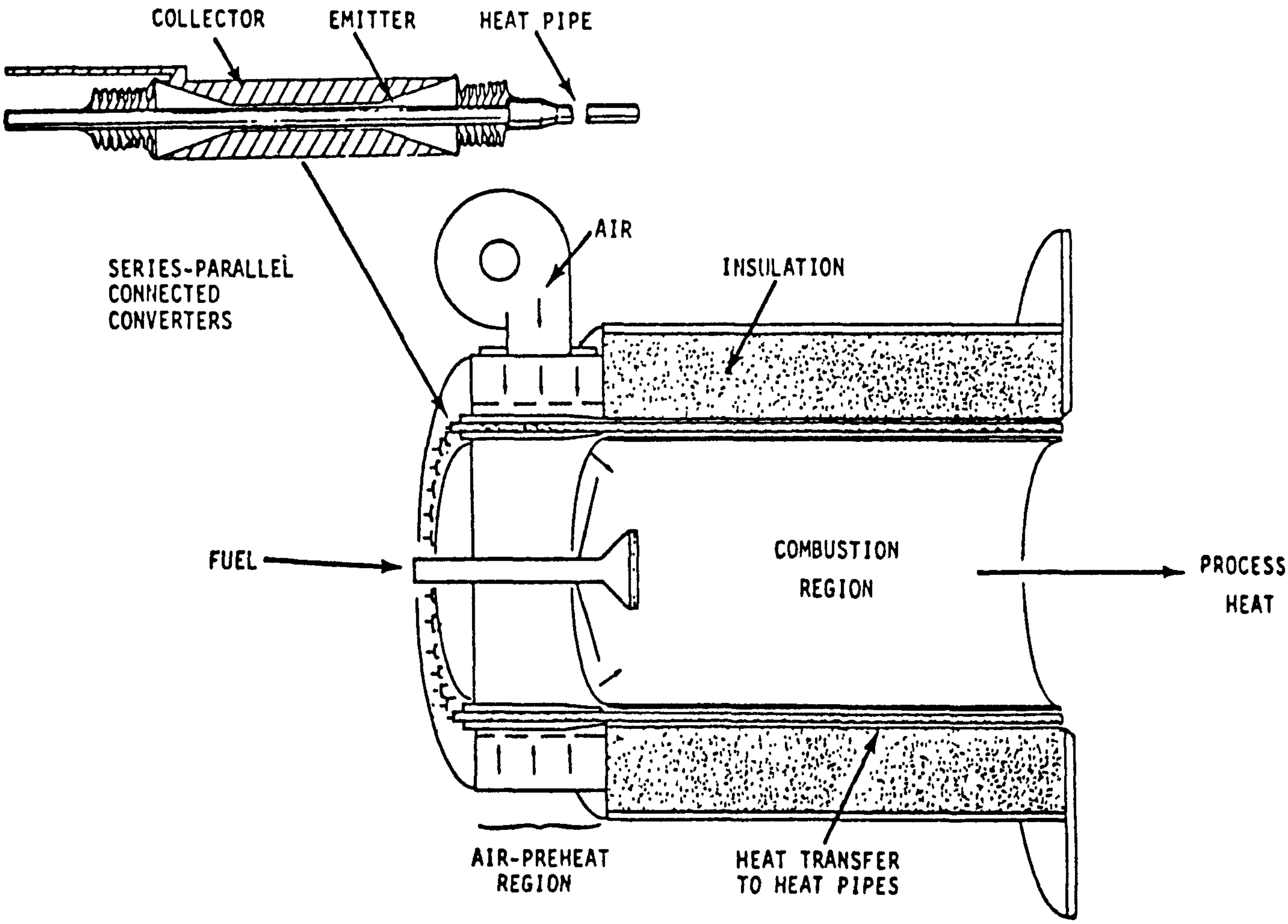


Exhibit D-1.3
Thermionic Cogeneration Combustor: General Design 1

pipes eliminate any problems associated with heat flux variations. Their use also permits converter operation with an isothermal emitter at the optimum current density and cesium pressure.

The air used for combustion is preheated using jet impingement against the collectors of each converter. The pumping power used by the air blower to accomplish this is about six percent of the electric output power of the combustor. Little of this energy is lost, since it ends up in the process heat stream. The converters are connected electrically in a series-parallel array to produce the desired output voltage and current.

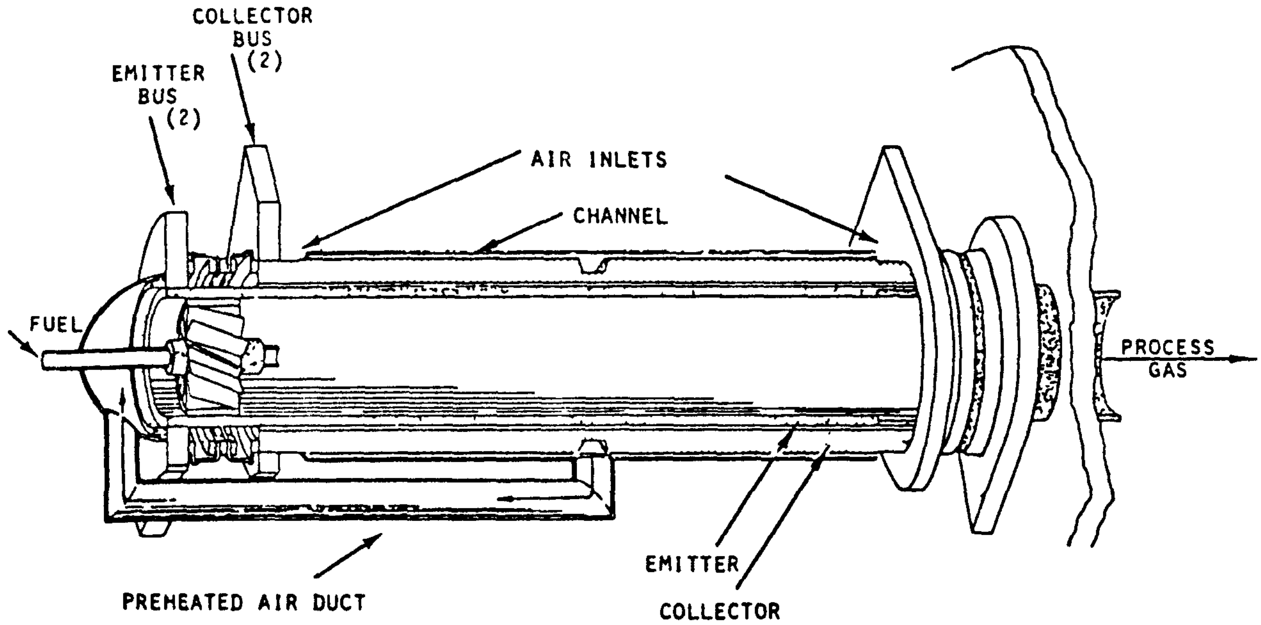
An example of the performance possible with this design is given in Exhibit D-1.4. In this case, 17 kWe is produced by a thermionic combustor sized to replace a 423 kWe (1.4 MBtu/hr) burner. The disadvantages of this design approach include the additional cost of the heat pipes, the necessity to surround the heat pipe array with a high-temperature wall, and the need to insulate that wall. The performance characteristics shown are approximately the best which can be achieved with the current generation of converters, and they are substantially less than may be achievable with more advanced thermionics.

The second approach involves the fire-tube converter shown in Exhibit D-1.5. In this approach, combustion occurs within a cylindrical emitter at the center of each converter. This process eliminates the need for an additional hot wall and its insulation. Variations in heat flux within the emitter are easily accommodated by the use of a coaxial heat pipe which integrates the local fluxes. The fire-tube converter is described in much greater detail in Part E, Section 3.2.

EXHIBIT D-1.4

COGENERATION THERMIONIC BURNER DESIGN 1

TOTAL AIR	120%
PREHEATER AIR TEMPERATURE, IN/OUT	300/816 K
PROCESS GAS	12.7 kg/hr
Temperature	1922 K
Duty	423 kWt
CONVERTER OPERATING POINT	
Emitter	1800 K
Collector	950 K
V_B	1.9 eV
Current Density	10 A/cm^2
Lead Power	100 W
Number	186
NET OUTPUT	
Power	17.5 kWe
Current	137 A
Voltage	128 volts DC



Fxhibit D-1.5

Fire-Tube Converter Design

The individual converters are assembled within a cylindrical enclosure, forming a combustor module, as shown in Exhibit D-1.6. During operation, combustion air blows into the plenum formed by the enclosure and then flows to the center of each converter through channels in the collectors. Following this preheating, the air is ducted to the burner end where it is mixed with fuel in the combustion region. As in the first design, the converters are electrically connected in a series-parallel array to obtain the desired output voltage and current.

A computer model of a thermionic cogeneration burner which can incorporate secondary air was developed and used to investigate burner performance and to optimize designs. Exhibit D-1.7 presents model calculations of electricity generated per million Btu of fuel fired as a function of the exhaust gas temperature supplied to the process. Secondary air is used as required. The computer model indicates increasing thermionic power with decreasing furnace gas temperature down to temperatures of 1200 K to 1300 K. Below this break-point, the thermionic power decreases with decreasing furnace gas temperature, since the high secondary air flow required reduces the air preheat and results in a lower than optimum combustion temperature and thermionic converter performance. Additional study is required to determine if modifications can be made to the thermionic burner to improve the thermionic power in this region. However, at present, applications to processes which can use higher gas temperatures directly appear more promising.

The remainder of this section focuses on particular processes which, because of temperature and/or other process characteristics, may be especially good candidates for early use of thermionic cogeneration. Several such processes have been studied, including: copper smelting, steam "trigeneration," steel

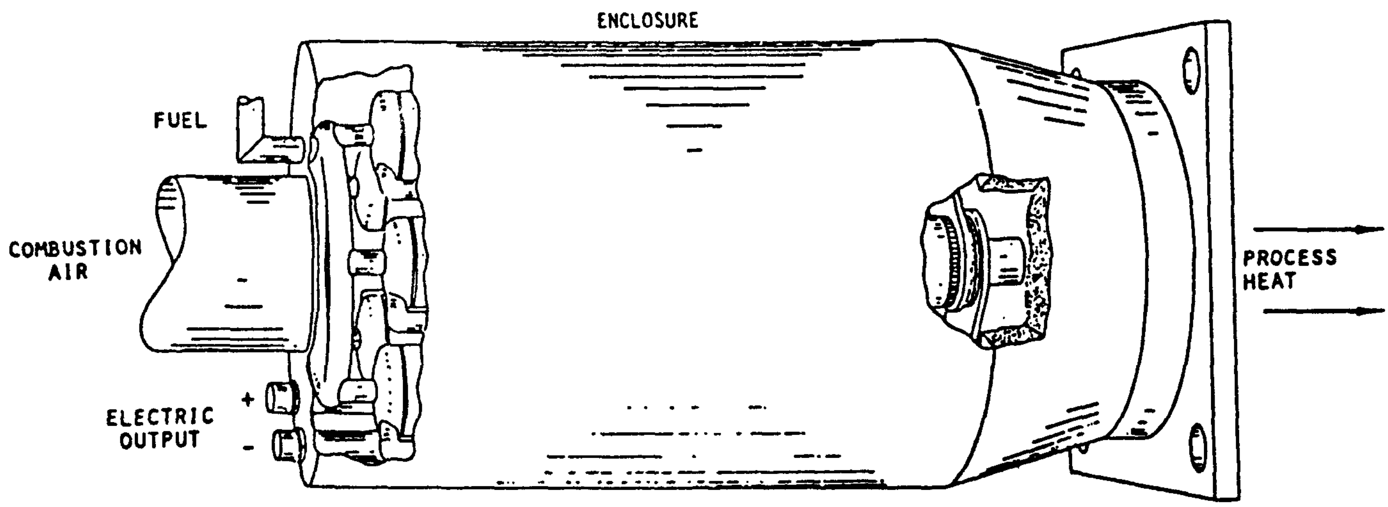


Exhibit D-1.6

Thermionic Cogeneration Combustor: General Design 2 (Fire-Tube Converter)

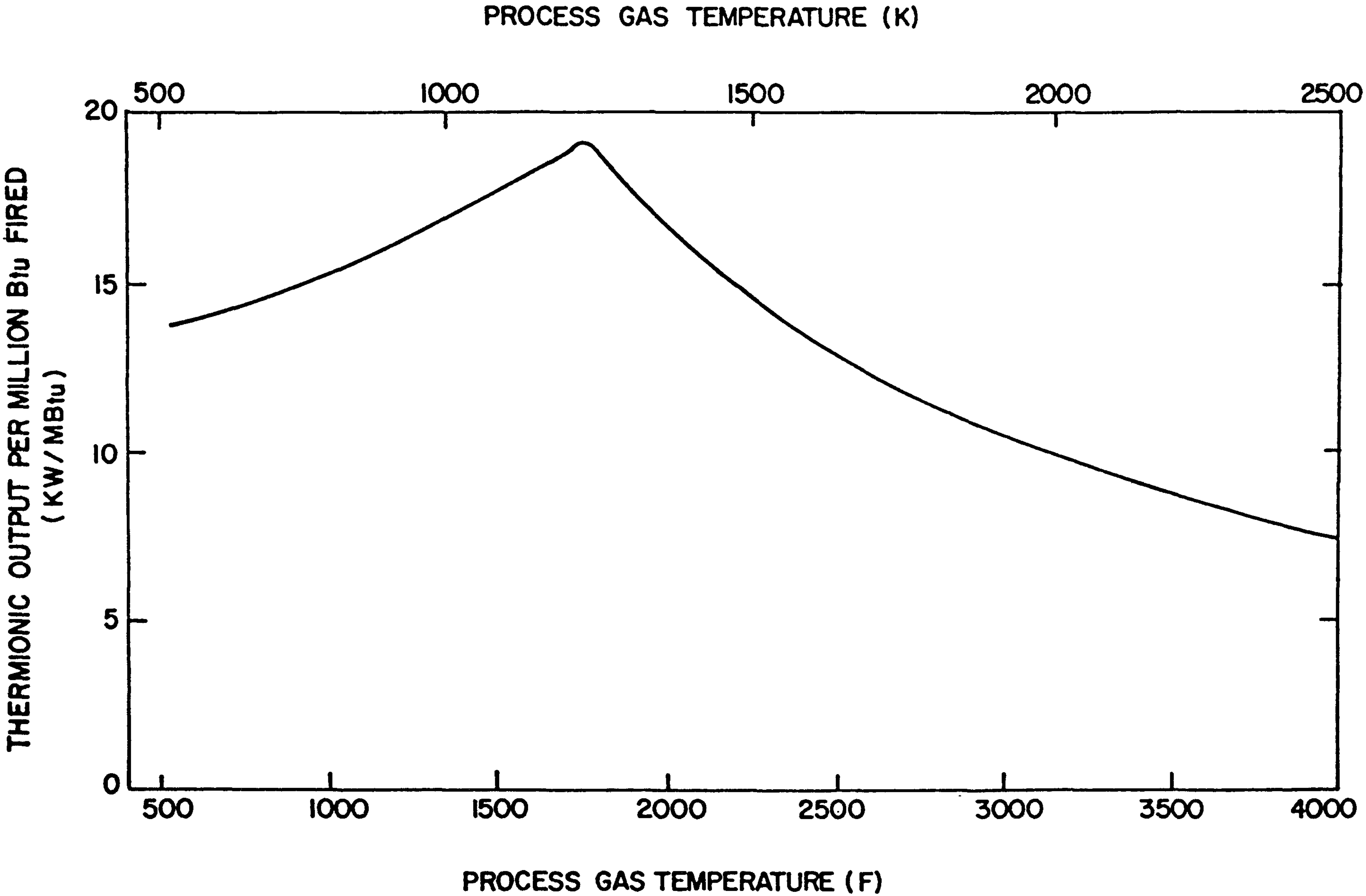


Exhibit D-1.7

Thermionic Generation as Function of Gas Temperature Supplied to Burner slab reheating, heat treating furnace gas generation, hot oil heating, glass melting, and edible oil hydrogenation.

1.2 Copper Refining

The widely used pyrometallurgical process for producing copper metal from copper/iron sulfide ores consists of several distinct stages: 1) ore mining and preparation; 2) roasting; 3) reverberatory smelting; 4) conversion; 5) fire refining; and, 6) electrolytic refining. Direct fuel energy is primarily used in the reverberatory smelting furnace and fire refining. The roasting process does not require external energy as long as the sulfur content of the ore concentrate is greater than 24%. The conversion process is exothermic except for compressed air pumps requiring electrical energy; this electricity is routinely provided by a steam turbine and waste heat boiler operating on the effluent gases of the smelting furnace. Electrical energy at low voltage is purchased, however, for the electrolytic refining of impure copper from the fire refining furnace into pure copper. About 86% of the copper produced in the United States is currently electrorefined.

To define the potential of thermionic cogeneration for copper production, a relatively modern pyrometallurgical process, the "Noranda" process, was considered. In this high-efficiency process, the roasting, smelting, and converting steps are carried out in a single furnace with a very significant improvement in energy efficiency since the exothermic converting process partially supplies the energy for the smelting process.

In Exhibit D-1.8, a schematic of the "Noranda" furnace is illustrated with material flows given for an 800 ton/day ore concentration furnace producing

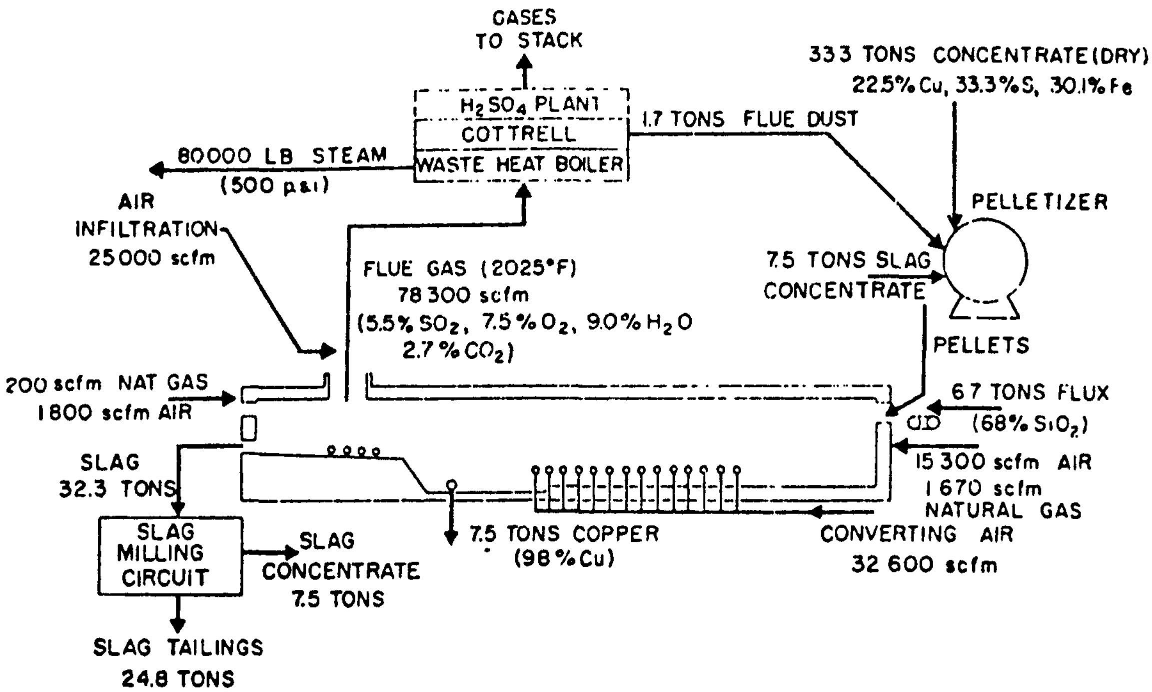


Exhibit D-1.8

Material Flowsheet (Per Hour) for Noranda Process Plant
(800 Tons Concentrate/Day)

180.8 tons Cu/day. In Exhibits D-1.9 and D-1.10, respectively, the material and energy balances are given. The surface temperature of the melt in the furnace is 1500 K and the stack gases leave at 1600 K. Natural gas burners operating with low excess air are used to provide the required thermal energy to the furnace.

Thermionic cogeneration is ideal for use with the Noranda process both because of this high temperature and because the low-voltage DC thermionic electrical power produced by the natural gas burners can be directly used by the electrorefining tank house. Presently, high-voltage AC is transformed and rectified to .2 V DC with 15% of the electrical energy lost in the transforming/rectifying.

From Exhibit D-1.8 the conventional plant fuel input rate is 1870 scfm of natural gas, equivalent to a fuel thermal input rate (i.e., at 1050 Btu/scfm) of 117.8 x 10^6 Btu/hr. The most efficient electrorefining house in the U.S. uses 159 kWh/ton Cu at a voltage of .185 V DC. Addition of an allowance of 12 kWh for lead losses and 30 kWh for rectifier losses results in a purchased AC energy requirement of 201 kWh/ton Cu. With a production rate of 180.8 tons/day of 98% Cu, the purchased electrical energy amounts to 1.300 x 10^7 kWh/year (i.e., equivalent to a steady power of 1484 kW).

For the plant with the thermionic cogeneration burner incorporated, the fuel input is increased by 3.76% to 122.2 x 10^6 Btu/hr. With use of a thermionic burner with 1100 K air preheat, 10% excess air, and 10% thermionic converter efficiency (including internal lead losses in the diode), 11 kWh of electrical energy is produced per 10^6 Btu fuel used. This value corresponds to a power output of 1344 Kw and 182.4 Kwh/ton of cathode Cu. With

EXHIBIT D-1.9

PROJECTED MATERIAL BALANCE FOR
800 TPD NORANDA PROCESS PLANT

	Tons/ day	Tons/ hr	% Cu	% Fe	% S	% SiO_2	% Zn	% H_2O
<u>Smelting Circuit</u>								
<u>In</u>								
Concentrate	800	33.3	22.46	30.1	33.3	4.8	3.4	10
Slag conc.	180	7.5	50	17	7	10	1.5	10
Flux	161.6	6.7	--	4.6	2.2	68.4	--	3
Fume	40	1.7	--	--	--	--	--	10
<u>Out</u>								
Copper	180.8	7.5	98	0.2	1.5	--	--	
Slag	774.4	32.3	12	36.1	1.0	21.6	3.0	
Fume	40	1.7						
<u>Milling Circuit</u>								
<u>In</u>								
Slag	774.4	32.3	12	36.1	1.0	21.6	3.0	
<u>Out</u>								
Slag conc.	180	7.5	50	17	7	10	--	
Tailings	594.4	24.8	0.5	42.0	--	25.1	--	

Losses in slag tailings = 1.66%

EXHIBIT D-1.10

PROJECTED HEAT BALANCE FOR 800 TPD
NORANDA PROCESS REACTOR

Item	Btu/Day	%
HEAT INPUT		
Converting reactions at 300 K:		
$FeS + \frac{3}{2} O_2 \rightarrow FeO + SO_2$	1795×10^6	42.7
$CuS + O_2 \rightarrow Cu + SO_2$	710×10^6	16.9
$ZnS + \frac{3}{2} O_2 \rightarrow ZnO + SO_2$	172×10^6	4.1
$3FeO + \frac{1}{2} O_2 \rightarrow Fe_3O_4$	208×10^6	5.0
$FeS_2 + O_2 \rightarrow FeS + SO_2$	143×10^6	3.4
Net available heat from fuel*	1169×10^6	27.9
Total	4197×10^6	100.0
HEAT OUTPUT		
Heat content of reactor gas at 1600 K (excluding combustion gas):		
N_2 in converting air	1820×10^6	43.3
H_2O from feed	535×10^6	12.7
SO_2 produced	583×10^6	13.9
Excess oxygen	55×10^6	1.3
Heat content of slag at 1500 K	774×10^6	18.5
Heat content of blister Cu at 1450 K	113×10^6	2.7
Heat content of flue dust	28×10^6	0.7
Heat loss (radiation, water cooling)	289×10^6	6.9
Total	4197×10^6	100.0

* Fuel requirement:

Heat required = $4197 \times 10^6 - 3028 \times 10^6 = 1169 \times 10^6$ Btu/day
Fuel required at 37% utilization efficiency = $1169 \times 10^6/0.37$ = 3160×10^6 Btu/day = 3.95×10^6 Btu/dry ton copper concentrate = 131×10^6 Btu/hour.

23.4 kWh allotted for bus bar losses, a net of 159 kWh/ton Cu results, eliminating the need for purchased power for electrorefining.

Economically, the saving in energy cost by use of the thermionic cogeneration burner must justify its incremental capital cost above a conventional burner. The annual energy savings varies with the cost of fuel and electric power, with an illustrative example given in Exhibit D-1.11. The year savings per kW of thermionic generating capacity in this example are (for a steady power of 1344 kW):

at	8¢ kWh,	\$630/kW
	6¢ KWh,	\$437/kW
	4¢/kWh,	\$243/kW

These potential savings can justify a substantial investment in the thermionic burner.

1.3 Steam "Trigeneration"

Steam cogeneration is becoming increasingly popular where low-temperature steam is needed for process heating. Steam is generated at high pressure and temperature in a fuel-fired boiler and used to generate electrical power in a non-condensing turbine with the turbine exhaust at the desired process steam conditions. Thermionic converters can be used in the steam boiler providing additional electrical power above that provided by the steam turbine. This "trigeneration" system would provide improved energy efficiency relative to steam cogeneration with no change to the steam cogeneration system other than substitution of the conventional burner by the thermionic burner.

EXHIBIT D-1.11

COMPARISON OF YEARLY ENERGY COSTS
(100% PLANT UTILIZATION FACTOR)

	With Conventional Burner	With Thermionic Cogeneration Burner
Fuel Thermal Consumption	117.8×10^6 Btu/hr	122.2×10^6 Btu/hr
Fuel Cost at $\$5/10^6$ Btu	5.159×10^6 \$/yr	5.352×10^6 \$/yr
Purchased Electric Power*	201 kWh/ton Cathode Cu	0 kWh
	1.300×10^7 kWh/yr	0 kWh/yr
Electric Power Cost*		
at 8¢/kWh	1.040×10^6 \$/yr	0 \$/yr
at 6¢/kWh	0.780×10^6 \$/yr	0 \$/yr
at 4¢/kWh	0.520×10^6 \$/yr	0 \$/yr
Total Energy Costs		
at 8¢/kWh	6.199×10^6 \$/yr	5.352×10^6 \$/yr
at 6¢/kWh	5.939×10^6 \$/yr	5.352×10^6 \$/yr
at 4¢/kWh	5.679×10^6 \$/yr	5.352×10^6 \$/yr

Yearly Value of Energy Savings

at 8¢/kWh - $847,000

at 6¢/kWh - $587,000

at 4¢/kWh - $327,000

* For electrorefining

A comparison of heat balances for three options of generating process steam is given in Exhibit D-1.12 and an energy and energy cost comparison is presented in Exhibit D-1.13. For the thermionic/steam trigeneration system, it is assumed that excess air of 100% is used in the thermionic burner to maximize the thermionic power output. The boiler efficiency is reduced to 80% instead of 85% as for the other options to reflect the lower combustion gas temperature to the boiler.

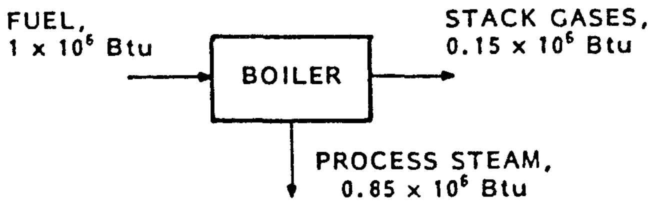
Use of steam cogeneration is attractive from an energy efficiency standpoint, as indicated in Exhibit D-1.13. Because of the inefficiencies associated with separate electricity generation by the utility, which cogeneration avoids, steam cogeneration has an energy utilization efficiency of 117%. Net energy cost to the user is reduced from $5.88 for 10^6 Btu of process heat to $3.37. Adding thermionic trigeneration raises the overall energy efficiency to 126% and reduces the net energy cost to $2.74 for 10^6 Btu of process heat.

1.4 Steel Slab Reheating

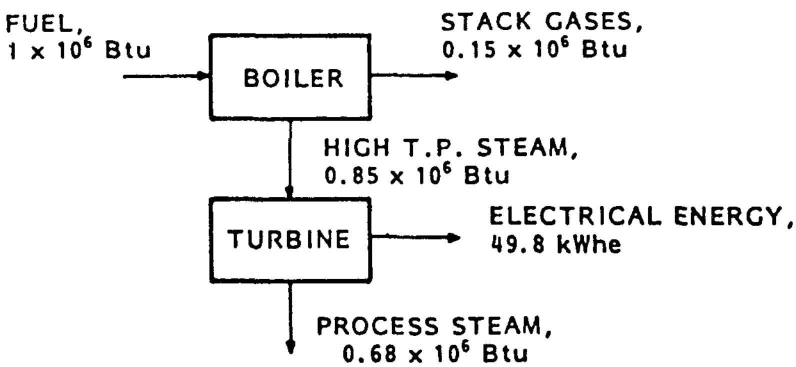
In a steel slab reheat furnace, steel slabs are heated to around 1475 K prior to being rolled in a mill. A walking beam reheat furnace manufactured by the Holcroft Division of Thermo Electron is shown in Exhibit D-1.14. The reheat furance and the associated rolling mill require a large amount of fuel and electricity.

Banks of burners are mounted on the sides of the furance so that they fire above and below the slab. The flue gases from the furnace are ducted to the recuperator where the incoming combustion air for the burners is preheated.

(I) STEAM BOILER



(II) STEAM COGENERATION



(III) THERMIONIC BURNER/STEAM TRIGENERATION

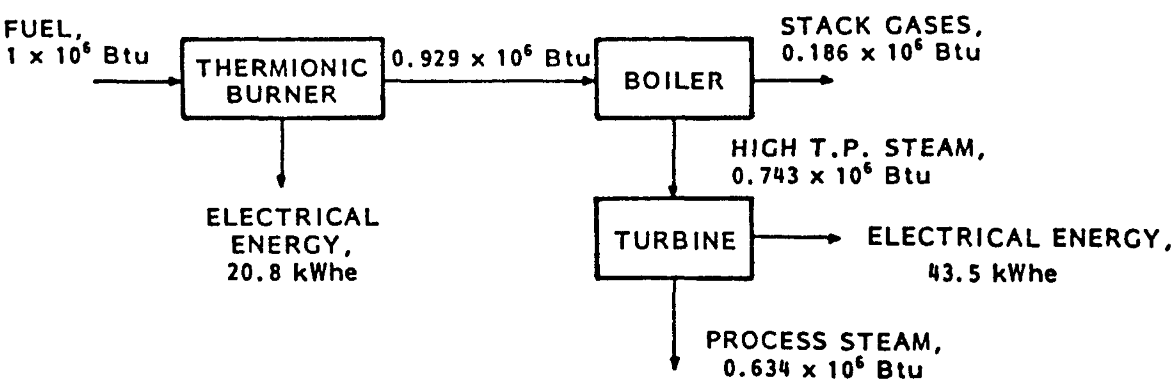


Exhibit D-1.12

Energy Balance for Three Options for Generating Process Steam

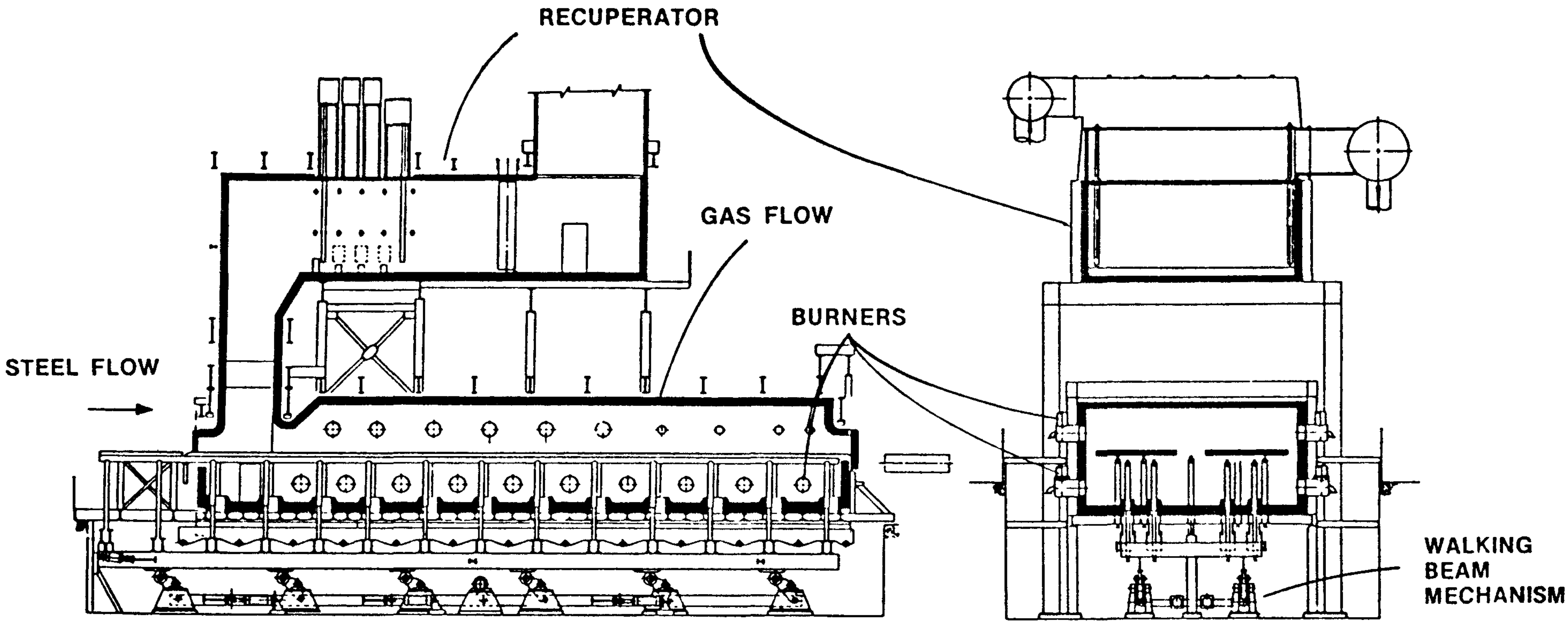
EXHIBIT D-1.13

ENERGY COST COMPARISONS OF THREE OPTIONS
FOR GENERATING PROCESS STEAM

System	Fuel Input (Btu)	Process Heat (Btu)	Electric Energy (kWh)	Overall Energy Efficiency* (%)	Fuel Cost** ($)	Electric Power** Credit ($)	Net Energy Cost to User ($)
Steam Boiler	1.176×10^6	1×10^6	0	85	5.880	0	5.880
Steam Cogeneration	1.471×10^6	1×10^6	49.8	116.6	7.355	3.984	3.371
Thermionic Burner/Steam Trigeneration	1.577×10^6	1×10^6	64.3	126.1	7.885	5.144	2.741

$$\text{* Overall Energy Efficiency} = \frac{\text{Process Heat} + \dfrac{(\text{Electric Energy})\ (3413)}{.35}}{\text{Fuel Input}}$$

** Based on Fuel Cost of $\$5/10^6$ Btu, Electric Energy Cost of $.08/kWh.



(a) LONGITUDINAL SECTION

(b) CROSS SECTION

Exhibit D-1.14

Water-Cooled Rail System in Walking Beam Furnace: (a) Longitudinal Section, (b) Cross Section. (Courtesy Holcroft Division, Thermo Electron Corp).

A drawing of the type of Bloom burners used on this furnace is shown in Exhibit D-1.15.

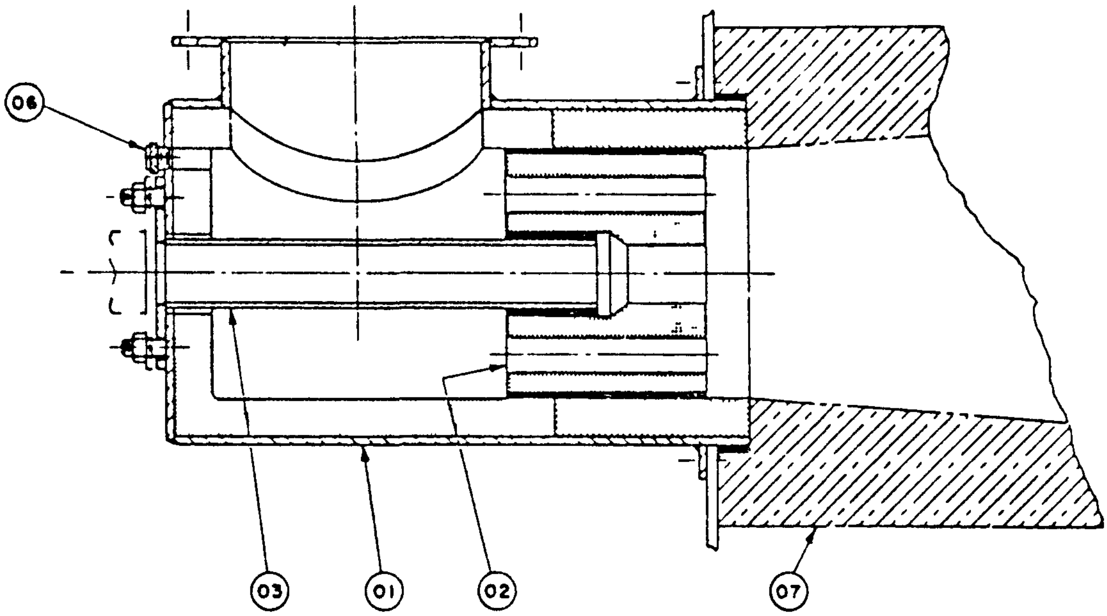
Thermionic technology can be readily incorporated into Bloom burners. The thermionic converters are similar to the flame-fired converters described earlier. The air for cooling the collectors of the converters is split into two streams. One stream provides combustion air and the other stream is used to dilute the combustion gases to the temperature desired for the process.

Exhibit D-1.16 shows the calculated fuel consumption of such a thermionic combustor as a function of steel production rate. Exhibit D-1.17 shows the corresponding electrical output.

1.5 Heat Treating Furnace Gas Generation

The endothermic gas generator provides carbon monoxide and hydrogen for the atmospheres of heat treating furnaces. Natural gas is catalytically reacted in a gas-heated retort to provide an atmosphere of 20% carbon monoxide, 40% hydrogen, and 40% nitrogen. A diagram of an endothermic gas generator with thermionic converters added is given in Exhibit D-1.18. Exhibit D-1.19 shows the characteristics of such a generator when sized for 30? atmospheres.

Ther endothermic gas generator is of interest for early thermionic application because it operates at constant power level and has a very small electrical power requirement of approximately four kilowatts. In Exhibit D-1.18, a utility interactive power conditioner is assumed. However, if the motors and solenoids were converted to DC and a battery was provided for start-up, the generator could be made independent of the utility system.



Part Number	Description
01	Body
02	Baffle
03	Nozzle Assembly
06	Observation Port
07	Port Block

Exhibit D-1.15

Bloom Burner Used on Steel Slab Reheat Furnance

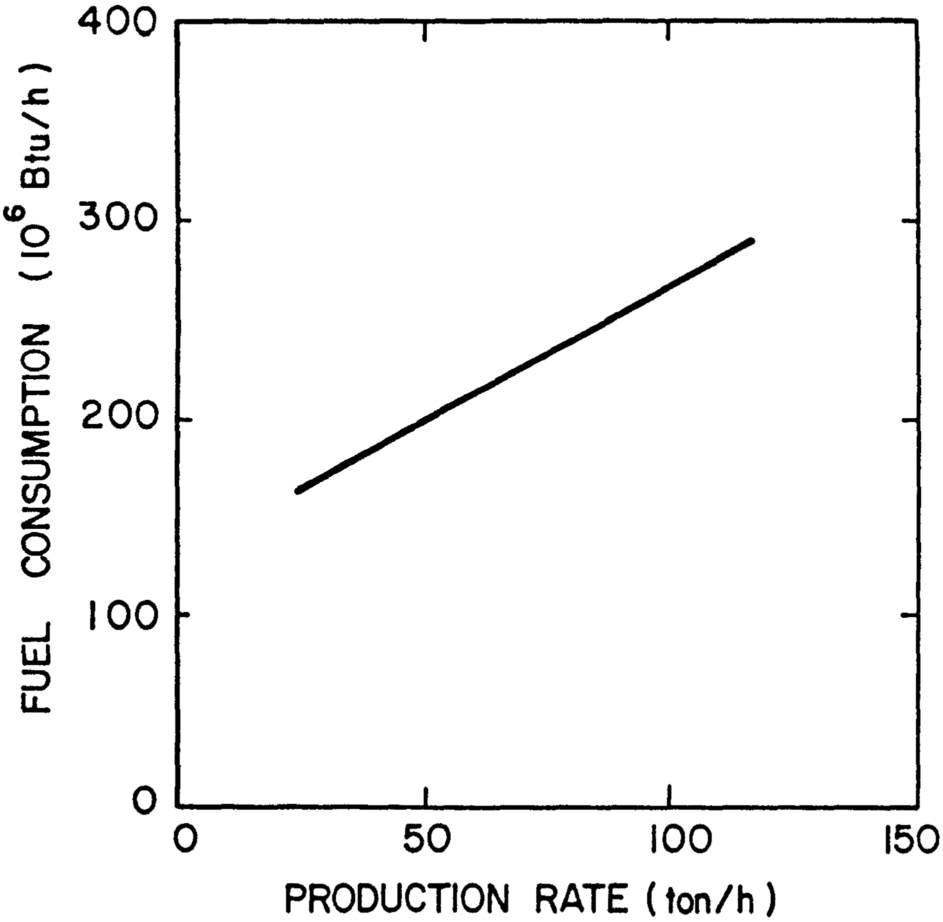


Exhibit D-1.16

Fuel Consumption Versus Production Rate
of Holcroft Walking Beam Furnace

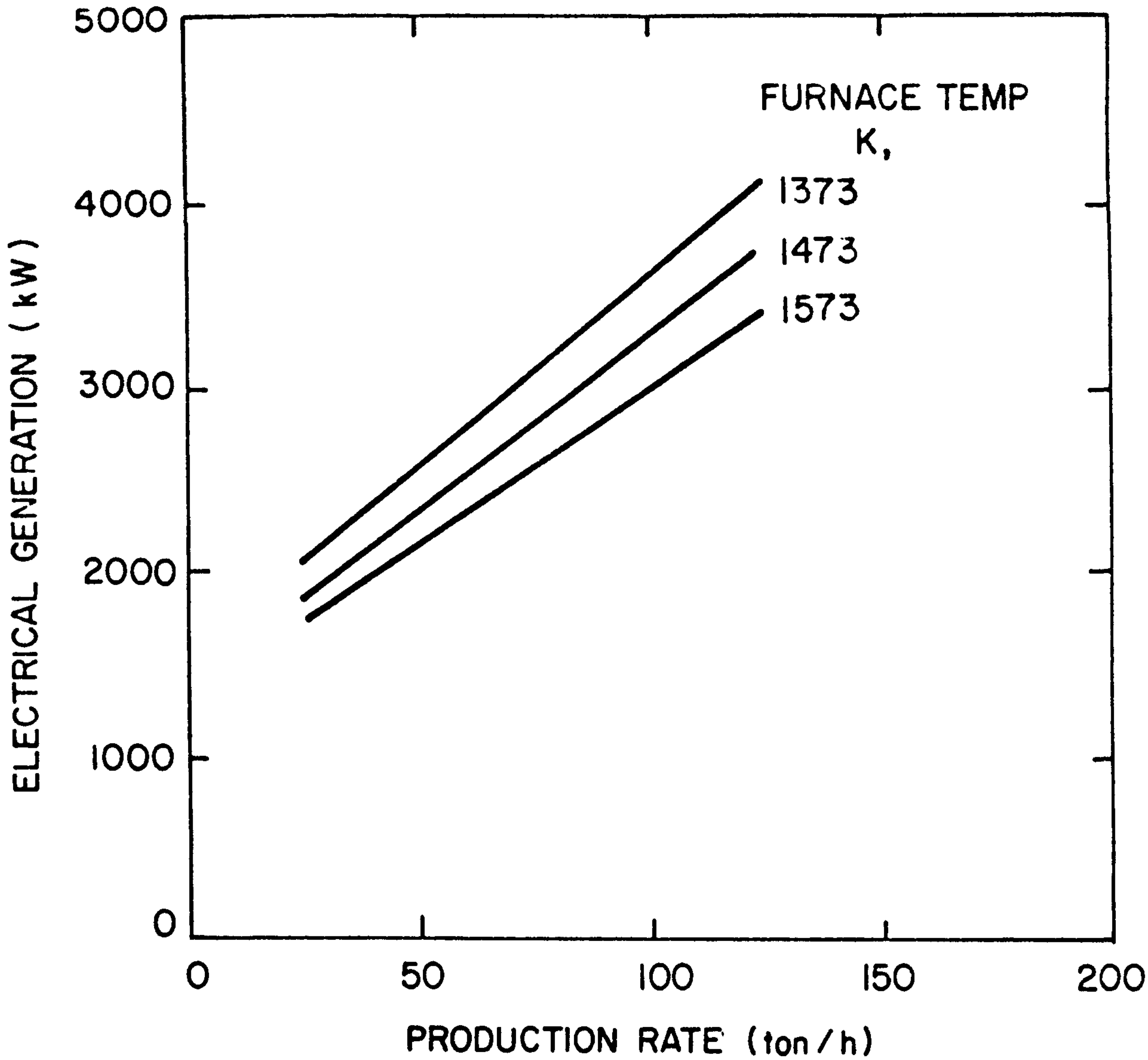


Exhibit D-1.17

Electrical Generation from Holcroft Walking Beam Furnace

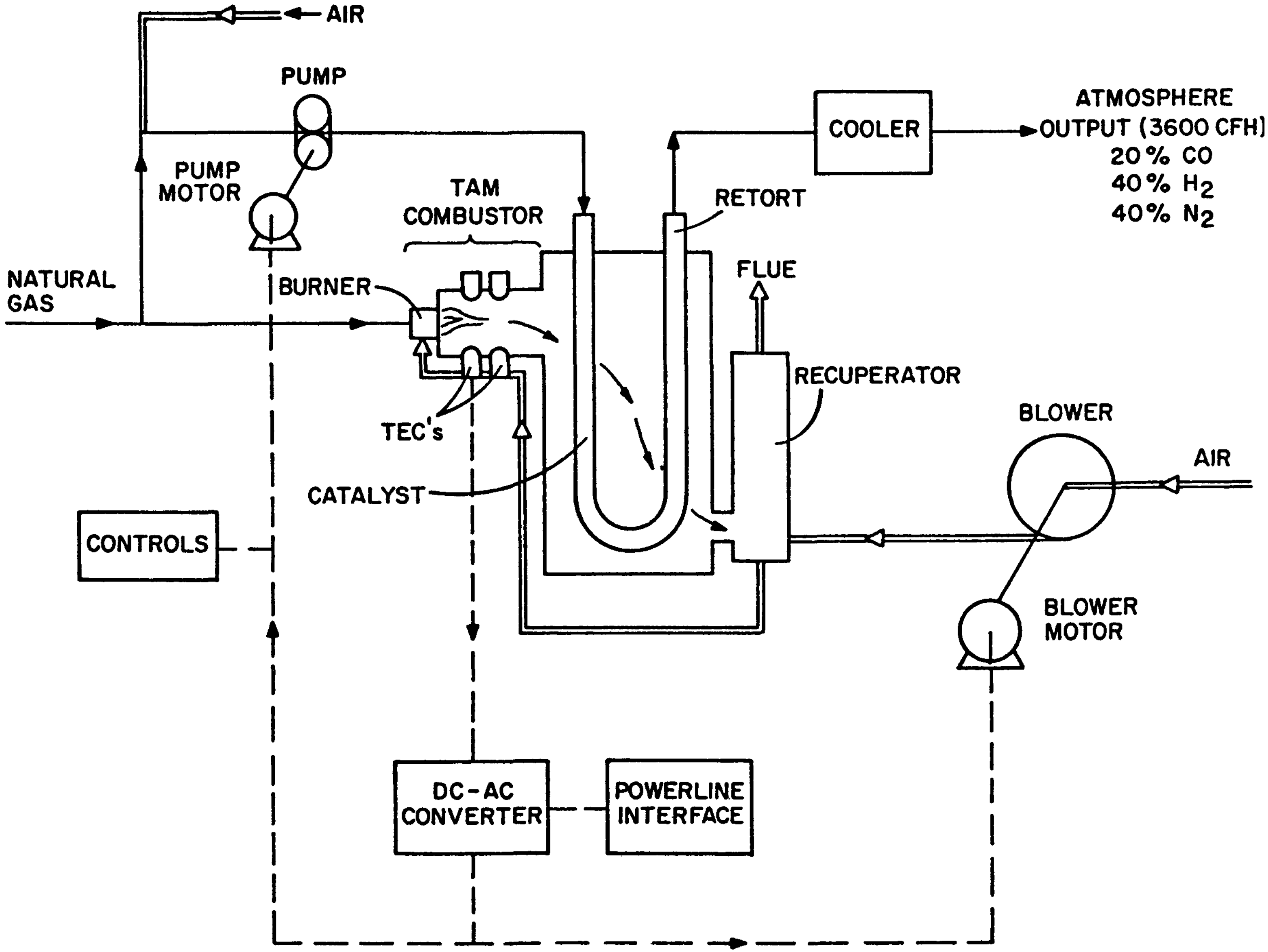


Exhibit D-1.18

Diagram of Endothermic Gas Generator with TCBMs

EXHIBIT D-1.19

CHARACTERISTICS OF THE 302-ATMOSPHERE GENERATOR WITH TCBMS

Atmosphere Generated, CFH (20% CO, 40% H_2, 40% N_2)	3600
Natural Gas Consumed, CFH	1210
Nominal Heat Input, Btu/hr	200,000
Converter Emitter Temperature, K	1700
Power Density, Watts/cm^2	2.6
DC Output, Kilowatts	4.1

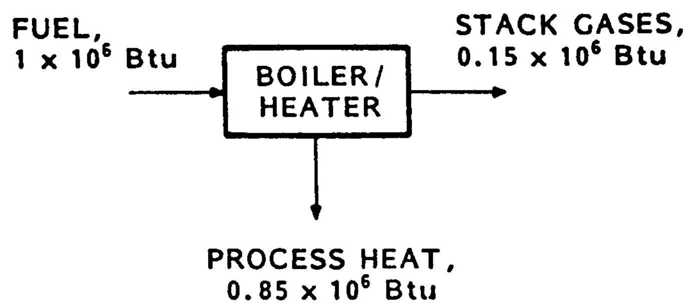
1.6 Hot Oil Heating

Dowtherm A vapor generators (or hot oil heaters) are used as process heat sources in the chemical industry where tight temperature control is required in the range of 475 K to 675 K. These heaters are offered commercially by several suppliers. The conventional gas or fuel oil burner in these units can be readily replaced by the TCBM. Since this is a low-temperature application, the percent excess air to the thermionic burner can be increased to 100% to maximize the thermionic power output to 20 kWh/10^6 Btu fuel. In Exhibit D-1.20 and D-1.21, the energy flows and costs are compared. By use of the TCBM, the overall energy efficiency is improved by a factor of 1.15, and the net energy cost to the user is reduced by a factor of .75.

1.7 Glass Melting

Large glass melting furnaces are generally of the dual regenerative type with periodically reversing flow which gives a high degree of air preheat to the burners. They are thus not well suited for use with a TCBM because of their variable temperature. Usually, smaller glass melting furnaces do not use regeneration so that they have resulting poor energy efficiencies. The TCBM can be applied to these small furnaces with a significant improvement in both energy efficiency and net energy cost as indicated in Exhibits D-1.22 and D-1.23. Because of the high combustion gas temperature required, low excess air must be used, which limits the thermionic power to 10.6 kW/10^6 Btu fuel. Even so, 97.4 kWh is produced per ton of glass melted, the overall energy efficiency is increased by a factor of 1.5, and the net energy cost to the user is decreased by a factor of .86.

(I) CONVENTIONAL BURNER



(II) THERMIONIC BURNER

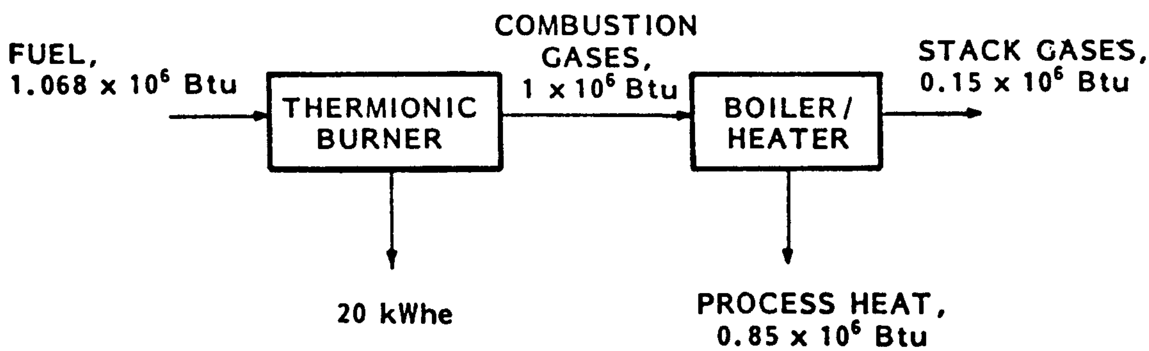


Exhibit D-1.20

Energy Balance for Dowtherm A or Hot Oil Industrial Boiler-Heater

EXHIBIT D-1.21

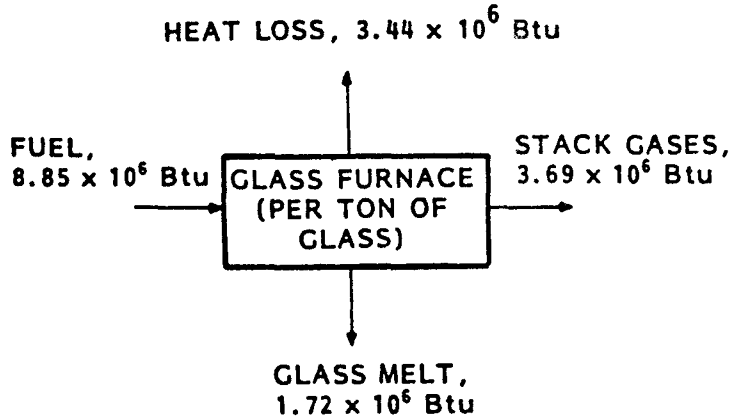
ENERGY COST COMPARISON FOR DOWTHERM A OR HOT OIL
INDUSTRIAL BOILER/HEATER (.85 x 10^6 BTU PROCESS HEAT)

Burner	Fuel Input (Btu)	Electric Energy (kWh)	Overall Energy Efficiency* (%)	Fuel Cost** ($)	Electric Energy Credit** ($)	Net Energy Cost to User ($)
Conventional	1 x 10^6	0	85	5.00	0	5.00
Thermionic	1.068 x 10^6	20	97.8	5.34	1.60	3.74

$$\text{* Overall Energy Efficiency} = \frac{\text{Heat to Process} + \dfrac{(\text{Electric Energy})\ (3413)}{.35}}{\text{Fuel Input}}$$

** Based on Fuel Cost of \$5/10^6 Btu, Electric Energy Cost of \$.08/kWh.

(I) CONVENTIONAL BURNER



(II) THERMIONIC BURNER

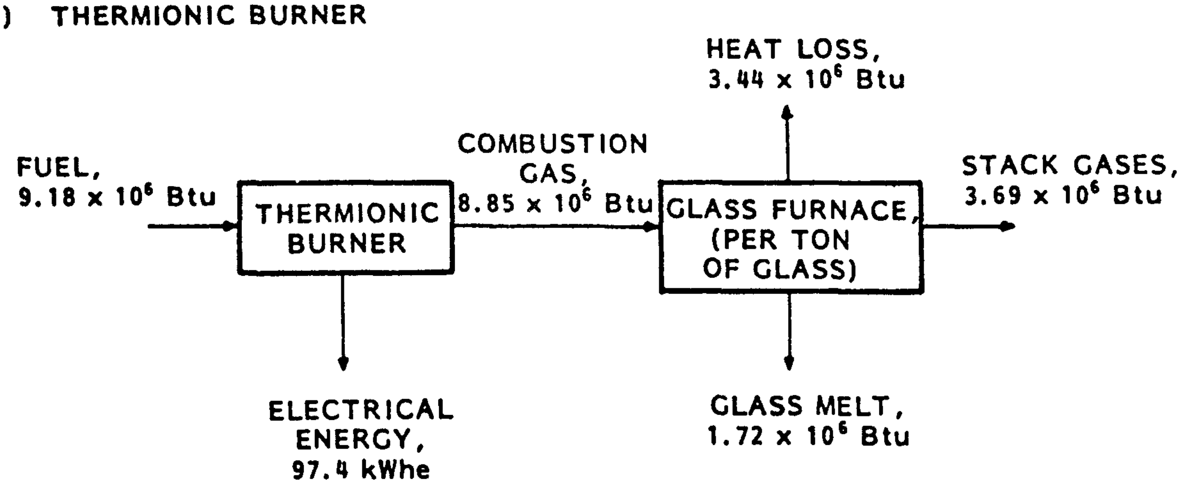


Exhibit D-1.22

Energy Balance for Small Unrecuperated Glass Furnaces

EXHIBIT D-1.23

ENERGY COST COMPARISON FOR GLASS FURNACE

(PER TON OF GLASS MELTED $\equiv$ 1.72 x 10^6 BTU)

Burner	Fuel Input (Btu)	Electric Energy (kWh)	Overall Energy Efficiency* (%)	Fuel Cost** ($)	Electric Energy Credit** ($)	Net Energy Cost to User ($)
Conventional	8.85 x 10^6	0	19.4	44.23	0	44.23
Thermionic	9.18 x 10^6	97.4	29.1	45.89	7.79	38.10

$$* \text{ Overall Energy Efficiency} = \frac{\text{Heat to Process} + \dfrac{(\text{Electric Energy})\ (3413)}{.35}}{\text{Fuel Input}}$$

** Based on Fuel Cost of $\$5/10^6$ Btu, Electric Energy Cost of $.08/kWh.

In hybrid glass furnaces, a combination of natural gas and electric (glass resistance) heating is used. According to data supplied to Thermo Electron [2], the electric input on such a furnace is about 7.4% of the gas heat. Moreover, AC power is used at a potential level of 100 to 500 Volts. Thus it would be difficult to integrate a TCBM engineering prototype into a hybrid glass melter. This application is also complicated because it uses a reverberatory furnace which is not consistent with the TCBM design concept. Another reservation relative to the glass melter is that the atmosphere is quite dirty. Thus a compatibility test of the silicon carbide hot shell in this hostile environment would appear to be required before this application can be considered. Therefore, it is not recommended that this application be identified with the TCBM until the foregoing problems have been addressed.

1.8 Edible Oil Hydrogenation

In this application a thermionic cogeneration burner is integrated into a hot oil heater used in the hydrogenation of edible oils with hydrogen produced by the electrolysis of water. This cogeneration application has the following special features:

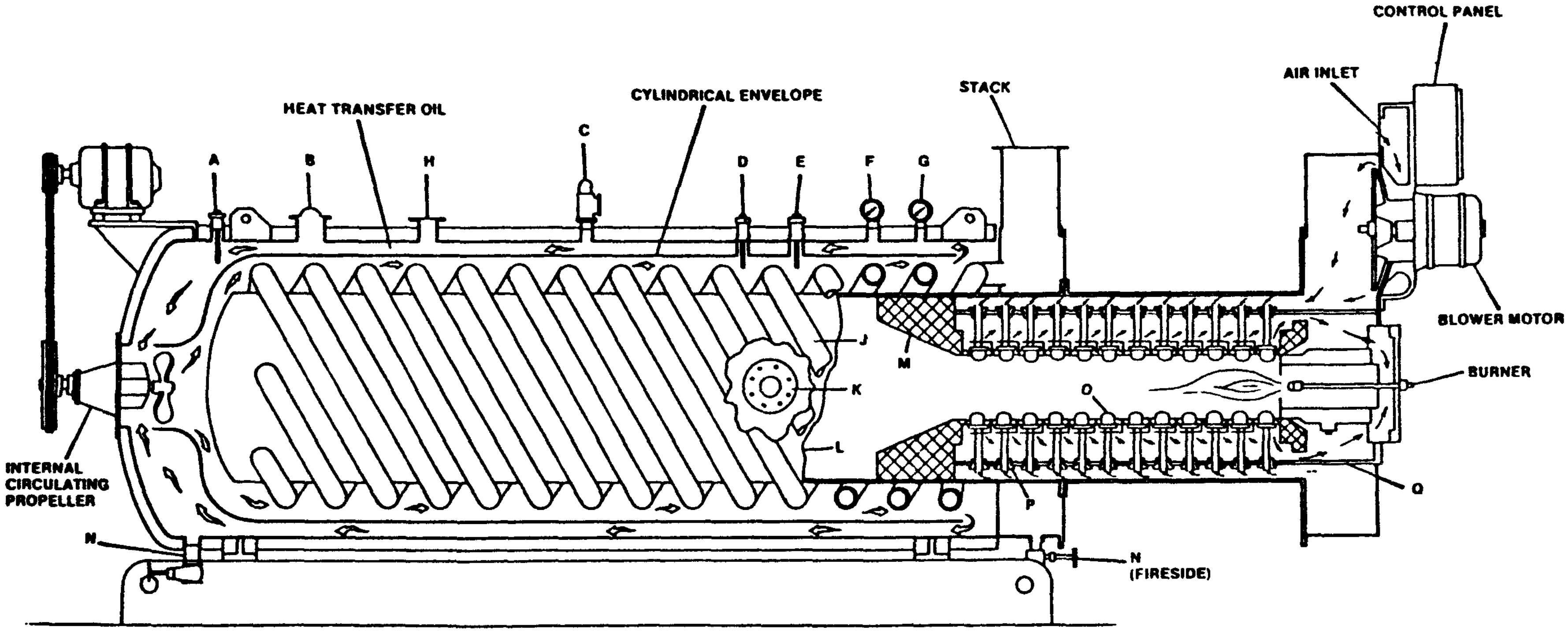
- The low-voltage DC from the thermionic converter module can be used to generate hydrogen without power conditioning.

- Catalytic hydrogenation of oils is an important industrial process (e.g., hydrogenated soybean oil provides about 60% of the fat in the U.S. diet). The edible oils are produced in 67 plants in this country, ranging from 25 to 750 million pounds of annual production. Nationwide, production is 13×10^9 pounds annually - corresponding to an energy consumption of 40×10^{12} Btu (approximately 7,000,000 bbls petroleum equivalent).

- Hydrogenation is a growth industry and will become increasingly important in areas other than edible oils (e.g., in petroleum refining where the declining H/C ratio of the crude requires additional hydrogen for optimum product output).

A conceptual design of a full-scale TCBM applied to an industrial hot edible oil heater is illustrated in Exhibit D-1.24 and the specifications of the oil heater are shown in Exhibit D-1.25. A monthly average energy flow sheet of a large conventional edible oil processing plant is presented in Exhibit D-1.26. Fuel energy requirements are summarized in Exhibit D-1.27.

The hydrogen in large plants currently comes primarily from natural gas/steam reforming. Hydrogen production by electrolysis of water has become increasingly attractive with the increasing cost of natural gas. The fuel and electric requirements for this process are indicated in Exhibit D-1.28 (i.e., for the same plant as in Exhibit D-1.26). While this route is now practical for the smaller plants where reforming is not desirable because of the small scale, the increasing cost of natural gas with deregulation may provide incentive for even the largest plants to take this route of hydrogen production. With electrolysis, the use of 6.35×10^9 Btu of natural gas and 5.5×10^9 Btu of steam for the reforming operation is replaced by 1.47×10^9 kWh (5.02×10^9 Btu) for water electrolysis. In Exhibit D-1.29, the energy



A. Thermostat-Internal Propeller
B. Rupture Disc. 45 PSIG
C. Relief Valve 30 PSIG
D. Limit Thermostats - 2
E. Modulating Thermostat
F. Pressure Gauge
G. Temperature Gauge
H. Expension Tank Connection
J. Helical Coil
K. Hot Oil Inlet and Outlet Both Sides
L. Furnace
M. Refractory
N. Drain
O. Thermionic Converter
P. Insulator
Q. Plenum Wall

Exhibit D-1.24

Conceptual Design of an Industrial Oil Heater using a Full-Scale Thermionic Cogeneration Burner Module

EXHIBIT D-1.25

INDUSTRIAL HEAT TRANSFER LIQUID HEATER SPECIFICATIONS

(Cleaver Brooks Model IPT700-55 "Peak Temp")

Fuel	Natural Gas
Fuel Input Rating	7,143,000 Btu/hr
Overall Length	227 in.
Overall Width	60 in.
Overall Height	89 in.
Dry Weight	9800 lb
Operating Weight	13,226 lb
Oil Inlet and Outlet Lines	4 in.
Maximum Operating Temperature	590 K

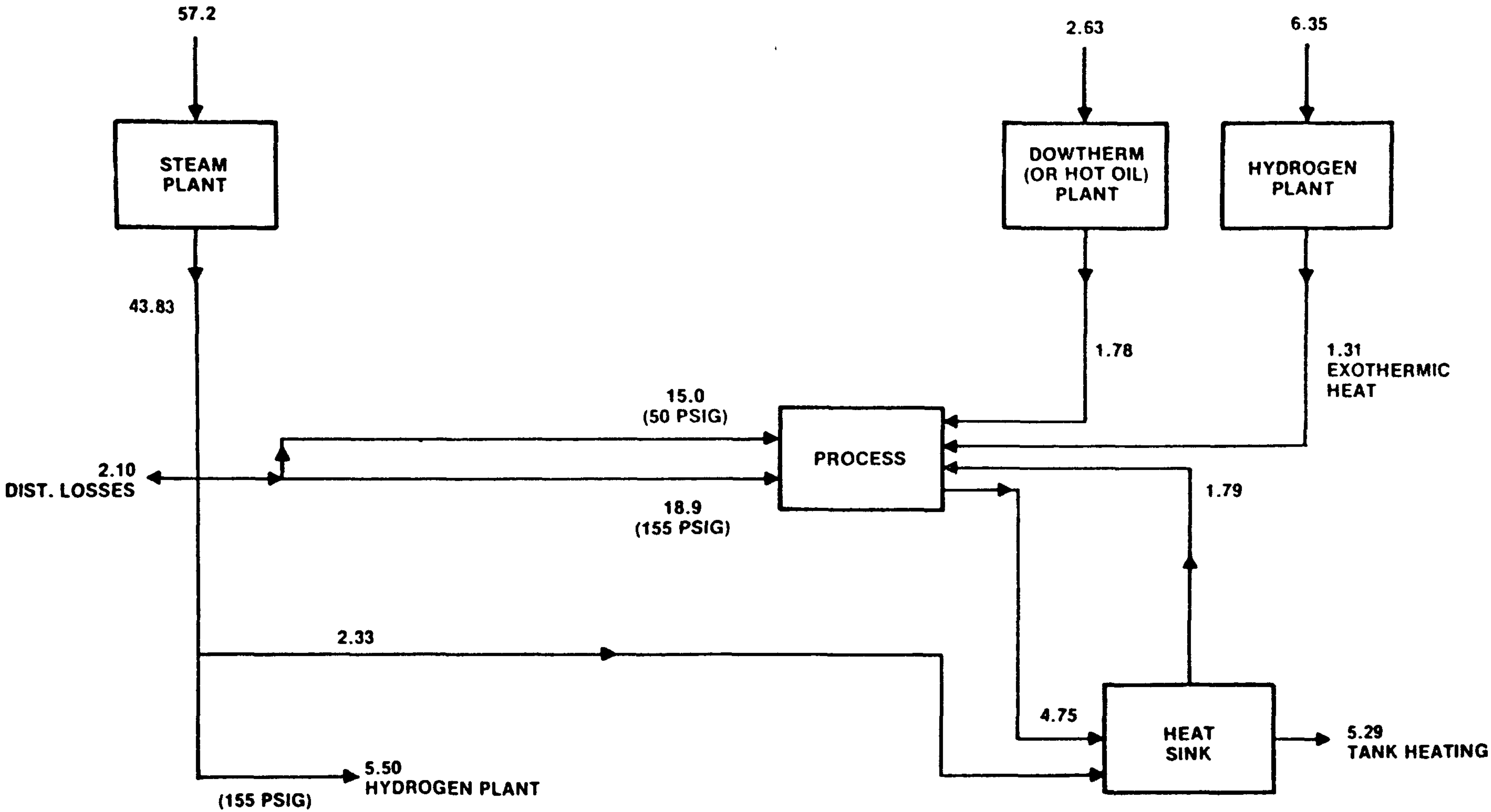


Exhibit D-1.26

Average Month Energy Balance for the Present Configuration of an Edible Oil Plant
(Units - 10^9 Btu/month)

EXHIBIT D-1.27

FUEL ENERGY REQUIREMENTS FOR LARGE CONVENTIONAL EDIBLE OIL PROCESSING PLANT

APPLICATION	MONTHLY REQUIREMENT
Process steam generation at 155 psig/50 psig	57.2×10^9 Btu
Dowtherm (hot oil) process heat	2.63×10^9 Btu
Hydrogen generation	6.35×10^9 Btu

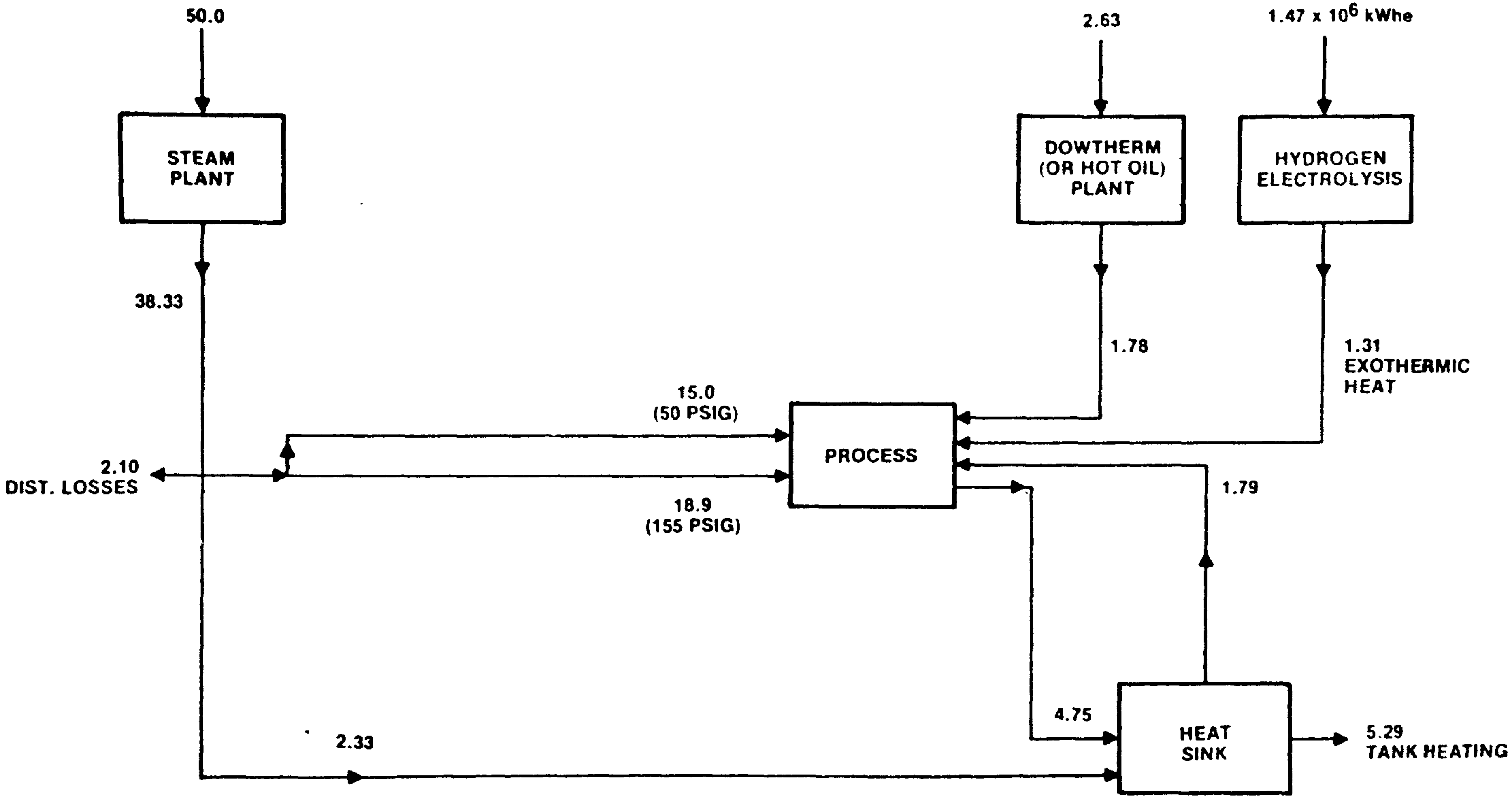


Exhibit D-1.28

Average Month Energy Balance for the Electrolysis (Purchased Power) Configuration
(Units - 10^9 Btu/month)

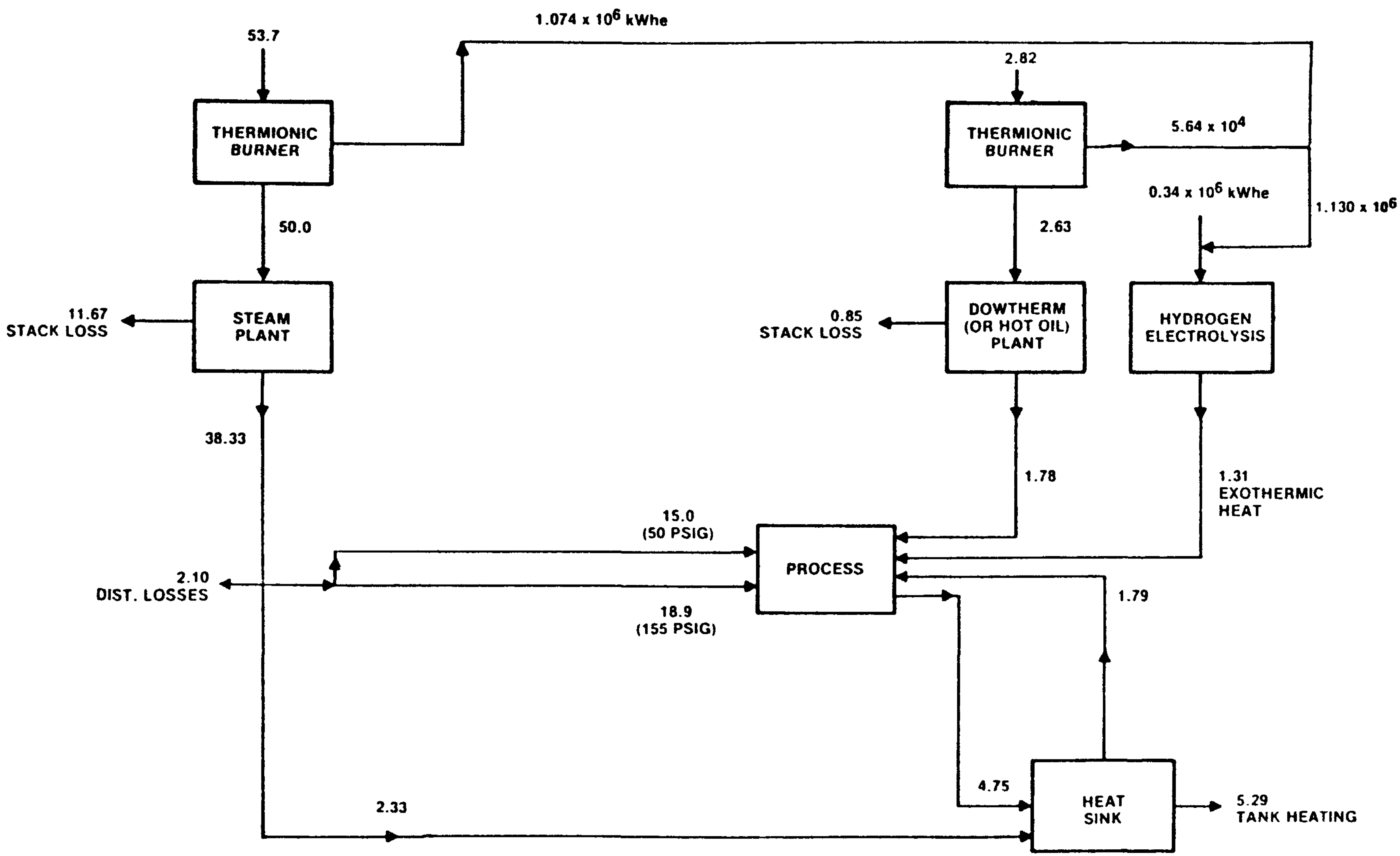


Exhibit D-1.29

Average Month Energy Balance for the Electrolysis (Output of Thermionic Burner Power Supplemented with Purchased Power) Configuration (Units - 10^9 Btu/month) balance resulting from incorporation of the thermionic burner into the steam boiler and the Dowtherm boiler (i.e., or hot oil heater) is given, with the thermionic electric output used directly for hydrogen production from electrolysis (i.e., based on 20 kWh/10^6 Btu fuel). With hydrogen electrolysis, application of the thermionic burner would generate 1.13×10^6 kWh each month, with a replacement value (e.g., at $.08/kWh) of $90,400/month, or $1,085,000/yr.

The particular plant on which the energy balances are given produces 312,000,000 pounds of oil annually with an annual energy requirement of $.794 \times 10^{12}$ Btu. Nationwide, production is 13×10^9 pounds annually, corresponding to an energy consumption of approximately 33×10^{12} Btu of fuel. In addition, auxiliary electric power for plant operation raises the total to 40×10^{12} Btu annually (7,000,000 bbls petroleum equivalent) or about 3,000 Btu/lb of edible oil produced.

SECTION 1 - REFERENCES

1. United Technologies Corporation, Cogeneration Technology Alternatives Study (CTAS) Final Report, Vol. V: Analytic Approach and Results, Report DOE/NASA/0030-80-5, NASA CR-159763, UTC FCR-133 (1980).

2. Mr. Dreyfuss, Midland Glass Company, personal communication, 1-12-82.

2. GAS TURBINE AND COMBINED-CYCLE TOPPING

2.1 Introduction

A continuing effort exists to improve gas turbine efficiency primarily through boosting turbine inlet temperatures by using high-temperature blades, special coating materials, or sophisticated blade cooling techniques. Presently available gas turbines operate with inlet temperatures near 1350 K even though flame temperatures of 2500 K are achievable. These lower gas inlet temperatures are produced by adding large quantities of excess air after combustion.

A thermionic combustor represents an alternative method for increasing gas turbine system efficiency without increasing turbine inlet temperatures. The thermionic converters can utilize the higher temperature gases directly to produce electric power. The energy so removed is compensated for by using less excess air to achieve the design turbine inlet conditions.

There are important differences between gas turbine and combined-cycle topping applications and the industrial cogeneration applications discussed in Part D, Section 1; these differences strongly influence the assessment of thermionic technology for the applications. The foremost of these differences relates to the financial motivation for the systems.

An industrial cogenerator must evaluate cogeneration -- regardless of technology -- as a potential investment which will ultimately be judged on some measure of return against alternatives ranging from other ways to reduce production costs of the product of interest, to investments in other products which may not be relevant to cogeneration, or even to non-production investments (e.g., real estate or subsidiary acquisition). Thus, simply demonstrating that cogeneration can reduce production costs may not be decisive in motivating adoption of the systems. On the otherhand, gas turbine or combined-cycle plants of the size discussed in this section (i.e., on the order of 100 MW or larger) are most commonly operated by regulated electric utilities for peaking power. In this financial environment, the bus bar cost of electricity produced by a technology is usually a sufficient financial measurement of its attractiveness, although factors such as reliability may also be important.

Because the industrial cogenerator is primarily concerned with producing a product, and not with generating electricity, there may be little incentive to adopt efficiency improvements that come as a cogeneration technology matures. Since the cogeneration system must normally be sized to fit process thermal demand, more electrically efficient systems may, in fact, result in negligible improvements in cogeneration economics if the extra efficiency means a poorer match to a facility's electrical-thermal demand ratio. Such improvements, of course, may open up new processes for cogeneration application.

In electric utility applications, however, any efficiency improvements which can be obtained at a reasonable capital cost are desirable since they will invariably reduce reliance on a utility's most expensive generating methods. Thus, in certain respects, electric utility applications represent a "friendlier" environment for the development of thermionic technology than do cogeneration applications. Counterbalancing these factors, however, are questions of scale and risk. As shown in Part D, Section 1, cogeneration applications exist at scales as small as a few kilowatts. Utility peaking stations, however, normally are no smaller than a few tens of megawatts, and may be a few hundred megawatts in size. Utilities must also be exceptionally concerned with reliability of supply, and, consequently, have a strong aversion to adopting any new technology until its operation has been thoroughly demonstrated at scales comparable to utility needs. Cogeneration applications of thermionics, therefore, will probably be commonplace before the applications discussed in this section are commercialized.

Nevertheless, the Program has investigated the design and economics of gas turbine and combined cycle topping applications at various times. Methods of calculation recommended by the Electric Power Research Institute (EPRI) were used to assess thermionic topping of a combined-cycle plant. Brown-Boveri Turbomachinery (BBT) and Stone and Webster Engineering Corporation (SWEC) were the major subcontractors for this study, and the combustor used a variation of the thermionic array module (TAM) design. A study with United Technologies Corporation (UTC) was also undertaken to examine the feasibility and cost of a thermionic gas turbine combined-cycle power plant where thermionic heat exchanger (THX) modules were located in the combustor of the turbine. An on-site coal gasifier was used with Rasor Associates providing the THX design and cost. UTC determined the cost of the remainder of the system.

These studies are not directly comparable to each other because of differing assumptions they involve. Similarly, because the economics of electricity generation has in recent years proven to be highly volatile as fuel cost, interest rate, inflation, and demand expectations have changed tremendously, the results of these studies cannot be regarded as reliable indicators of thermionic generation costs when thermionic systems of the required scale are actually available. Nevertheless, the studies are of interest because they both suggest that, could such thermionic systems be constructed and reliably operated today, they could lower the cost of energy produced by steam turbine and combined-cycle technologies.

The remainder of this section discusses these studies.

2.2 Thermionic Array Module (TAM) Design

The TAM design is conceptually very similar to the generalized cogeneration systems described in Section 1. The topping system consists of a compressor, a multizone thermionic combustor, a gas turbine, and a steam bottoming cycle. Ambient air would enter the compressor at an assumed temperature of 290 K; after compression at a 12:1 ratio, the air would be discharged to the collector cooling section of the thermionic combustor at 630 K. The air would be further heated by the collectors to 870 K and then split into two streams. One portion would go to the combustor first stage, while the other portion would be mixed with the combustor discharge gases. The proportion of air in the first stage and discharge streams would be determined by heat balance requirements.

Rich combustion would take place in the first stage of the combustor, producing a flame temperature of about 2600 K. Thermionic emitters, themselves at temperatures ranging from 1800 K down to 1600 K, would convert a portion of the flame energy to electricity, and the combustion products would fall to a temperature of about 1200 K before being mixed with the collector cooling air stream. Since the first stage of combustion would occur in a rich fuel-air mixture, unburned fuel would remain for burning in a secondary combustion zone. In order to minimize NO_x generation, water quenching and noncatalytic ammonia injection would be employed immediately prior to this secondary combustion.

After secondary combustion, the resulting gases would enter the turbine at approximately 1370 K. The turbine has been assumed to be a BBT Type II. Turbine exhaust gas would be at approximately 800 K when made available to the steam boilers. In the analysis a single-steam process pressure has been assumed although, in actual practice, the use of two or three pressure streams is more likely.

Exhibit D-2.1 is a schematic of the combustor. For thermionic topping a center body would be added to an existing BBT Type II turbine combustor design. The center body would greatly increase the surface area for mounting thermionic converters.

A medium-Btu gas, whose composition is shown in Exhibit D-2.2, has been selected as the system fuel for purposes of the analysis. It has been assumed that the fuel was supplied from the gasifier at a temperature of 480 K. The flame temperature and NO_x concentration of the combustion gases have been calculated by a thermodynamic equilibrium computer program with the results shown in Exhibits D-2.3 and D-2.4, respectively. These results have indicated that a combustion air temperature of 810 K is sufficient to provide a flame temperature of above 2480 K (4000°F) over a wide range of air-fuel ratios.

As the combustion gas flows through the combustor, heat would be transmitted (i.e., mainly by radiation) from the gas to the thermionic converters

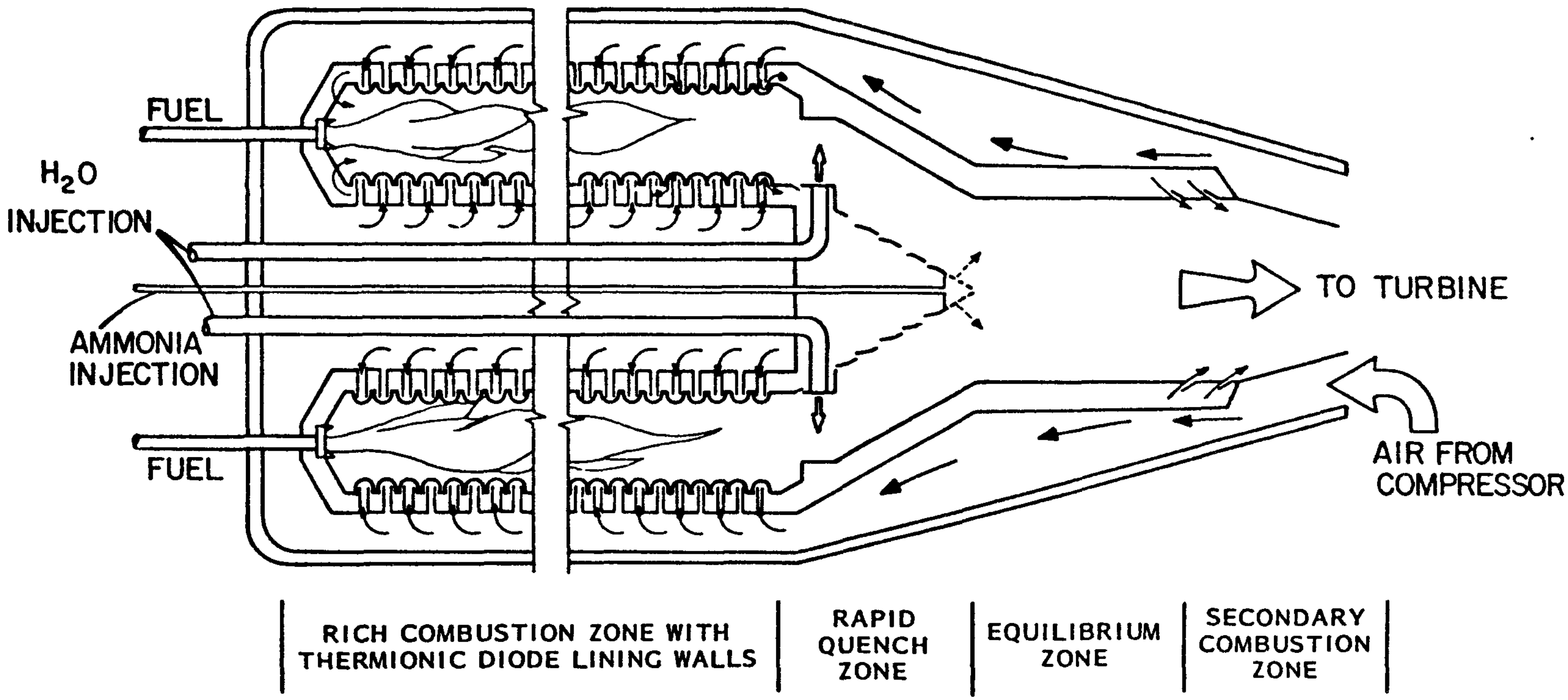


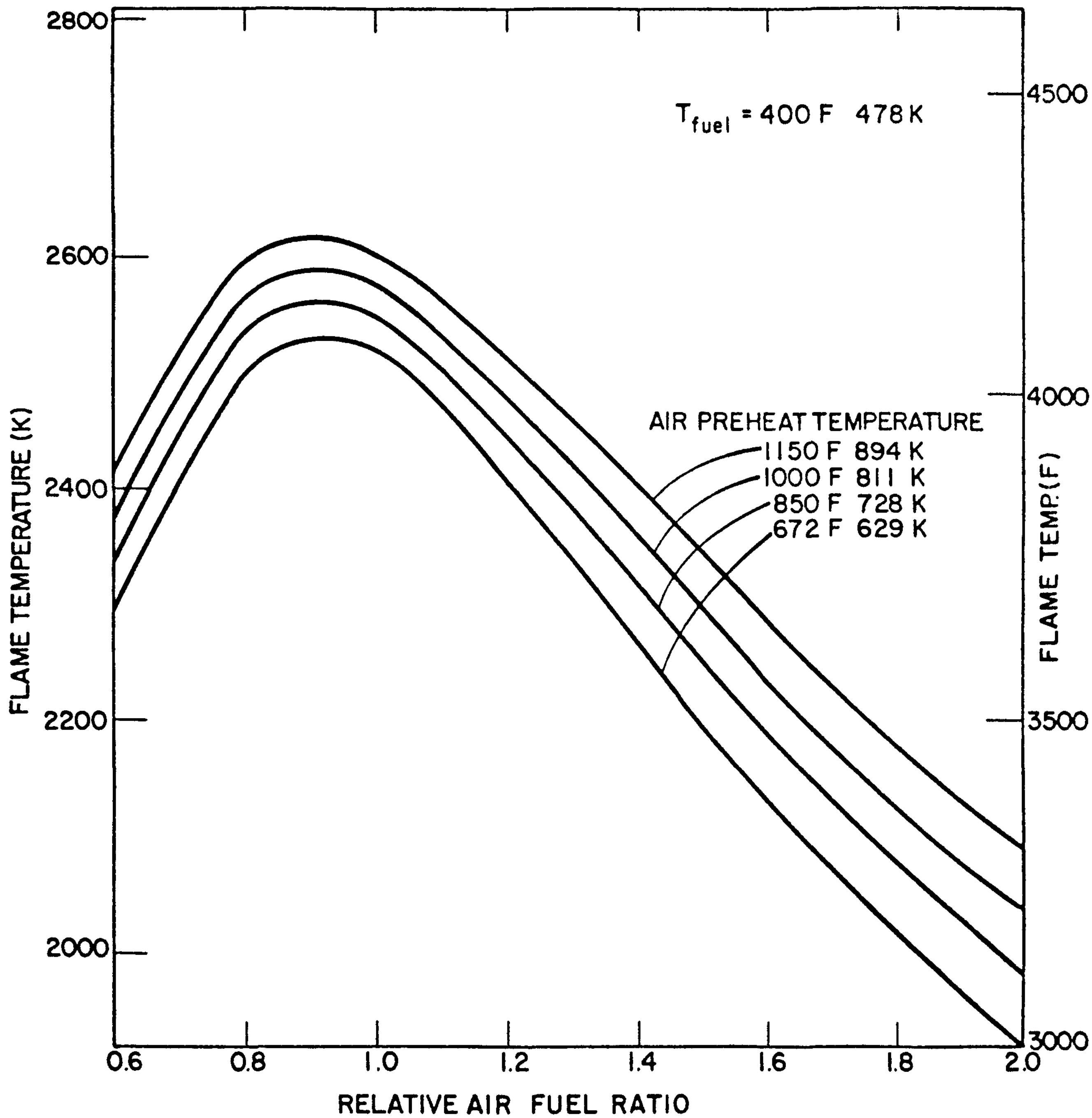
Exhibit D-2.1

Thermionic Combustor for Combined-Cycle or Gas Turbine Topping

EXHIBIT D-2.2

MEDIUM-BTU FUEL COMPOSITION

CONSTITUENT	VOLUME FRACTION
H	.3594
CO	.5151
CO_2	.1186
N_2	.0059
CH_4	.001



Fxhibit D-2.3

Flame Temperature Versus Air-Fuel Ratio in Thermionic Combustor for Gas Turbine and Combined-Cycle Topping

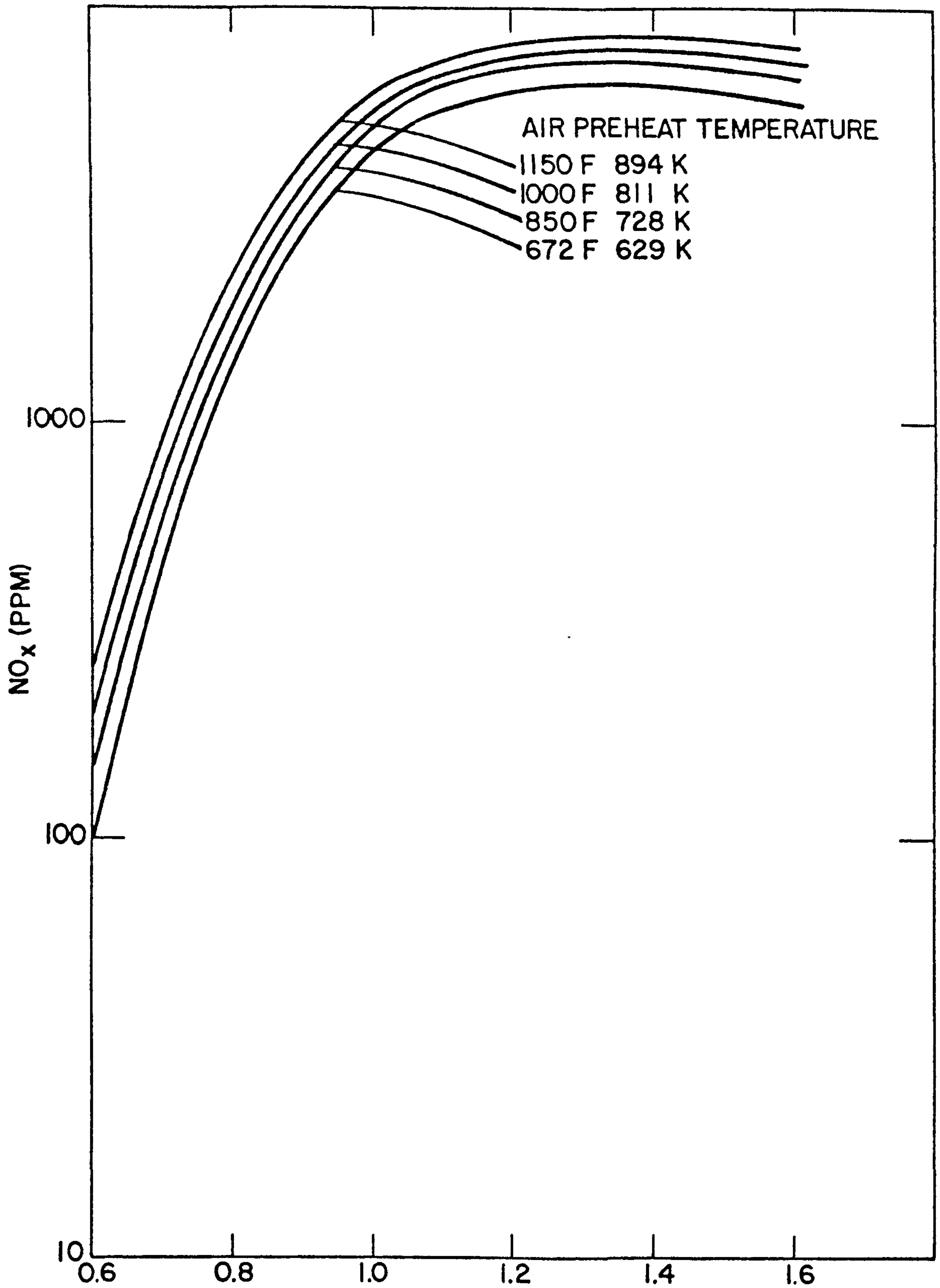


Exhibit D-2.4

NO_x Concentration as a Function of Relative Air-Fuel Ratio and Air Preheat Temperature in Thermionic Combustor for Gas Turbine and Combined-Cycle Topping and the temperature of the gas would be gradually reduced. The temperature -- hence the heat flux from the gas -- would change continuously from the entrance to the exit of the combustor. This would result in a change in the performance of the thermionic converters. In order to simplify the system assessment, an average performance of the thermionic converters has been calculated based on this temperature variation. The average thermionic performance has then been used as an input to the system heat balance calculations.

The nominal size of the untopped combined-cycle system has been assumed to be 100 MW.

2.3 Thermionic Array Module Performance and Economics

The performance of the thermionic converters in the system has been calculated according to the method suggested by Hatsopoulos and Gyftopoulos [1] with two modifications. The arc drop across the plasma was estimated from both experimental data and theoretical plasma analyses. The voltage drop across the emitter surface was calculated from the detailed design of the torispherical emitter, as was the cost, which included material, processing, and labor [2].

The rules which were used in the economic assessment are those published by EPRI [3] and have been widely used in utility practice. Costs for the combined-cycle turbine systems and the balance-of-plant have been prepared by BBT and SWEC, respectively. More specifically, the cost of the untopped plant was calculated based on a quotation for a typical combined-cycle plant supplied by BBT.

The cost of the thermionic converters and the required accessories have been estimated by combining Thermo Electron's estimates of converter manufacturing cost, vendor quotations for purchased parts [2], BBT estimates for modification to the standard combustor for accommodating thermionic converters, and the added cost of modifying the generator to be coupled to the thermionically driven acyclic motor. The estimates for the cost of the acyclic motor and the additional switch gear required for the thermionic portion of the plant were prepared by SWEC. A 15% contingency was added to all TAM-related costs.

The converter costs were calculated on a unit basis; the total cost was obtained by calculating the number of units needed to generate the required power. The cost or the thermionic balance of the plant was estimated for a reference 7.7 MW thermionic output. The costs were assumed to vary linearly with thermionic power output.

Exhibit D-2.5 compares the economics of the topped and untopped systems. In the untopped system, the gas-turbine output is 69 MW; the steam turbine produces 30 MW for a total of 99 MW. The overall efficiency is 44%, corresponding to a lower heating value heat rate of about 7700 Btu/kWh. The total plant cost is $38 million; the specific cost is $380/kW. If the fuel cost is assumed to be $\$5/10^6$ Btu ($5.3/kJ), then the levelized system cost can be calculated as a fuel cost of 73 mills/kWh, an operating and maintenance cost of 3 mills/kWh and a capital cost of 12 mills/kWh, resulting in a total bus bar cost of 88 mills/kWh.

This can be compared with the thermionically topped case, also shown in Exhibit D-2.5, which uses the same fuel and gas-turbine, steam-turbine system.

EXHIBIT D-2.5

COMBINED-CYCLE COMPARISONS

	UNTOPPED	TOPPED
DESIGN PARAMETERS:		
Fuel	Med. Btu	Med. Btu
Heating Value, kJ/kg	11.4	11.4
Gas-Turbine Inlet Temp., K	1367	1367
Emitter Temp. Range, K	-	1800-1600
Collector Temp., K	-	1000
Air Preheat Temp., K	628	936
Relative Air-Fuel Ratio	-	.9
Thermionic Barrier Index, eV	-	2.2
Current Density, A/cm^2	-	5
Avg. Power Density, W/cm^2	-	2.4
PERFORMANCE:		
Gas-Turbine Output, MW	69	69
Steam-Turbine Output, MW	30	30
Thermionic AC Output, MW	-	8
Total Output, MW	99	107
Efficiency, %	44	46
Incremental Efficiency, %	-	88
COSTS:		
Total Plant Cost, $\$ \times 10^6$	38	42
Total Plant Cost, $/kW	380	390
Incremental Cost, $/kW	-	530
Fuel Cost, $/kJ	5.3	5.3
Levelized Annual: mills/kWh		
Fuel Cost	73	70
O&M Cost	3	3
Capital Cost	12	12
Bus Bar Cost	88	85
Incremental Bus Bar Cost mills/kWh	-	55

The thermionic converters would be arranged in five temperature stages -- i.e., with the hottest stage near the burners of the primary combustor -- of 1800 K, 1750 K, 1700 K, 1650 K, and 1600 K. The thermionic collectors would preheat the combustion air to 940 K.

The current density of the thermionic converter is 5 Amperes/cm^2 with a barrier index of 2.2 eV, resulting in an average power output of 2.4 Watts/cm^2. The performance of the gas and steam turbines would be virtually the same as the untopped case; the output of the thermionic converters would be an additional 8 MW, resulting in a total topped plant output of 107 MW. The thermal efficiency of the plant would be 46% and the corresponding heat rate would be 7400 Btu/kWh. The incremental heat rate (defined as the ratio of the heating value of the additional fuel burned to the additional power generated) would be 3900 Btu/kWh. This would correspond to an incremental efficiency of 88%.

The economic effect of the thermionic system can be viewed as a decrease of three mills per kilowatt hour in the price of fuel (i.e., due to the higher efficiency with which electricity is produced) without assuming significant changes in operating and maintenance or in capital costs per kW. This would result in decreasing the bus bar cost of electricity from 88 to 85 mills/kWh.

A more realistic cost estimate has been obtained by considering the thermionic system in greater detail. The thermionic stage operating at an emitter temperature of 1800 K has been subdivided into 4 substages operating at different current and power densities. By operating at higher power densities, these stages of thermionics would provide a better match to the higher heat flux available at the combustor. The 1800 K thermionic stage has been divided into substages with current densities of 13, 11, 8, and 5 Amperes/cm^2. The results are shown in Exhibit D-2.6. Average thermionic power has increased to 3.7 Watts/cm^2 from 2.4 Watts/cm^2, the gas and steam turbine cycle is unaffected, and the overall system efficiency is unchanged. However, the cost of the topped system has decreased to $41 million, resulting in a further decrease in total bus bar costs to 84 mills/kWh. Indeed, the incremental cost of the generation supplied by the thermionic converters alone is only 53 mills/kWh.

Investigations were also undertaken to examine the effects of the principal thermionic and gas-turbine variables on system performance and cost. The cases just described were taken as a basis for varying the parameters one at a time to explore the sensitivity of the system.

In Exhibit D-2.7 the thermionic power and incremental bus bar cost are plotted against current density. For these calculations the thermionic current density has been assumed to be the same for all five emitter temperature stages. As the current density increases to 5 Amperes/cm^2, the power from the thermionic converter increases to 8.1 MW, then decreases as the current is further increased. At low currents, the efficiency of the thermionic converter is low, resulting in a lower overall output. As the current is increased, the heat flux required at the emitter becomes large, which requires high combustion gas temperature for heat transfer. The minimum incremental bus bar cost of 54 mills/kWh occurs at a higher current density than the maximum power. The cost reduction results from an increased average power output density and the lower unit thermionic cost. At still higher output, the reduced enthalpy extraction results in a slowly increasing cost. The cost at maximum power is not significantly different from the minimum cost, 55 versus 54 mills/kWh.

EXHIBIT D-2.6

COMPARISON OF COMBINED-CYCLE WITH AND WITHOUT THERMIONIC TOPPING: VARIABLE THERMIONIC CURRENT DENSITY

	UNTOPPED	TOPPED
DESIGN PARAMETERS:		
Fuel	Med. Btu	Med. Btu
Heating Value, kJ/kg	11.4	11.4
Gas-Turbine Inlet Temp., K	1367	1367
Emitter Temp. Range, K	-	1800-1600
Collector Temp., K	-	1000
Air Preheat Temp., K	628	939
Relative Air-Fuel Ratio	-	.9
Thermionic Barrier Index, eV	-	2.2
Current Density, A/cm^2	-	13-5
Avg. Power Density, W/cm^2	-	3.7
PERFORMANCE:		
Gas-Turbine Output, MW	69	69
Steam-Turbine Output, MW	30	30
Thermionic AC Output, MW	-	8
Total Output, MW	99	107
Efficiency, %	44	46
Incremental Efficiency, %	-	88
COSTS:		
Total Plant Cost, $\$ \times 10^6$	38	41
Total Plant Cost, $/kW	380	380
Incremental Cost, $/kW	-	447
Fuel Cost, $/kJ	5.3	5.3
Levelized Annual: mills/kWh		
Fuel Cost	73	70
O&M Cost	3	3
Capital Cost	12	11
Bus Bar Cost	88	84
Incremental Bus Bar Cost mills/kWh	-	53

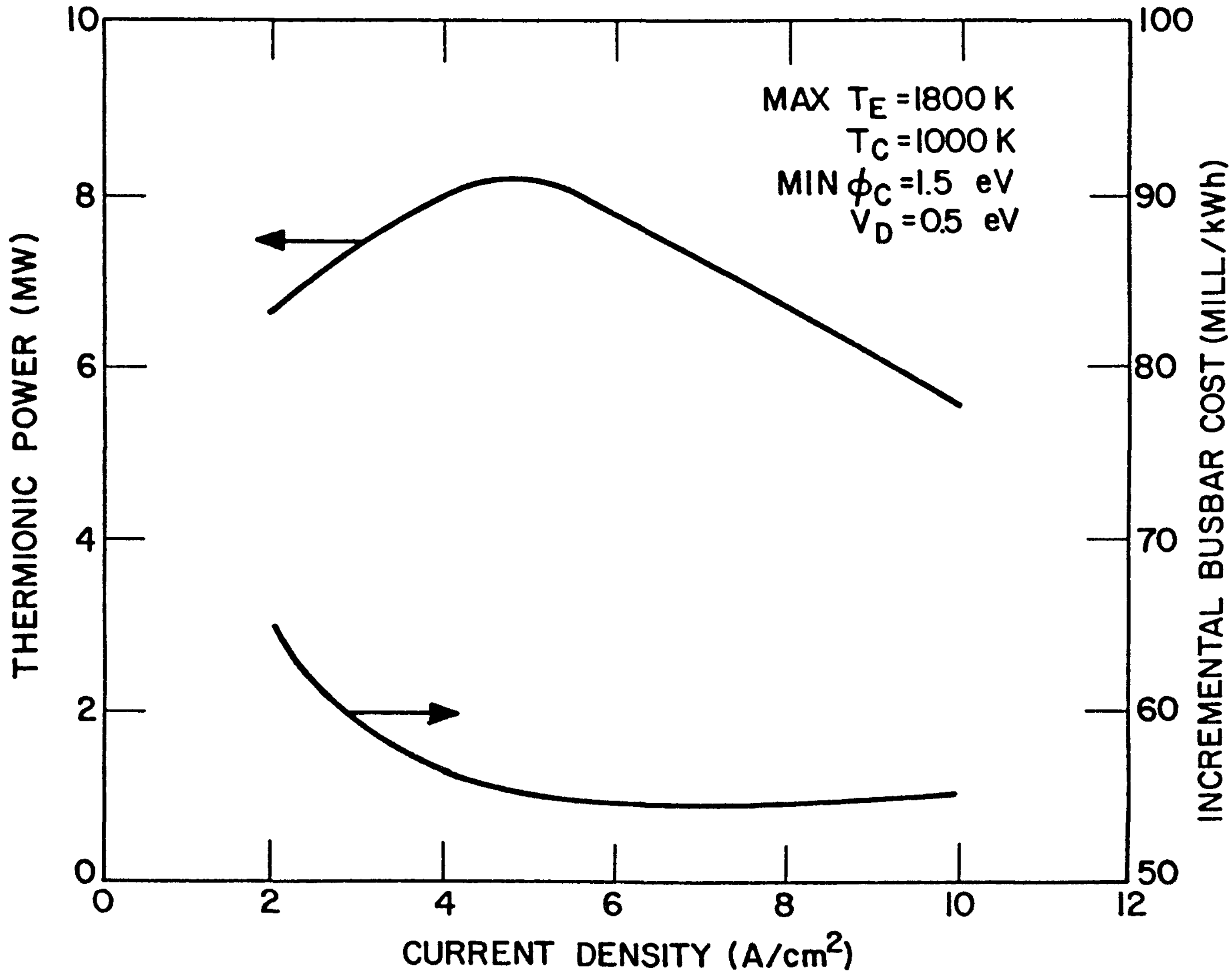


Exhibit D-2.7

Thermionic Power and Cost Versus Current Density for Steam Turbine and Combined-Cycle Topping

Exhibit D-2.8 shows the power supplied by thermionic topping as a function of the gas temperature (i.e., after secondary combustion) supplied to the gas turbine. As expected, power generation from the topping system declines as more energy is "reserved" for the turbine.

The performance of a thermionic converter improves with increasing emitter temperature and with increasing input power density. Therefore, it is desirable to operate most of the thermionic converters at the highest practicable temperature. To maximize the extraction of enthalpy from the hot combustion gases, the temperature of the emitters of the successive stages is lowered until the power output of the converter is below 1 Watt/cm^2. This occurs typically at 1600 K. It is uneconomical to operate thermionic converters with present-day performance below this temperature. Exhibit D-2.9 illustrates the effect of the emitter temperature of the hottest stage of thermionics. With a maximum emitter temperature of 1800 K, 8 MW are obtained. The power decreases to 7.4 MW as the maximum emitter temperature is lowered to 1730 K. This temperature corresponds to the actual converter emitter temperature in the 12,500 hour life test discussed extensively in Part C. Exhibit D-2.9 also shows the effect of reducing the arc drop in the converter. If the arc drop is reduced to .3 eV, the maximum power obtained can be increased to 11 MW.

The effect of collector temperature is shown in Exhibit D-3.10, where maximum thermionic power for various collector temperatures is plotted. The maximum thermionic power is generated at a collector temperature of 950 K with a minimum collector work function of 1.5 eV. If the collector work function is lowered to 1.3 eV, e.g., by using advanced collector materials, the optimum collector temperatures become lower.

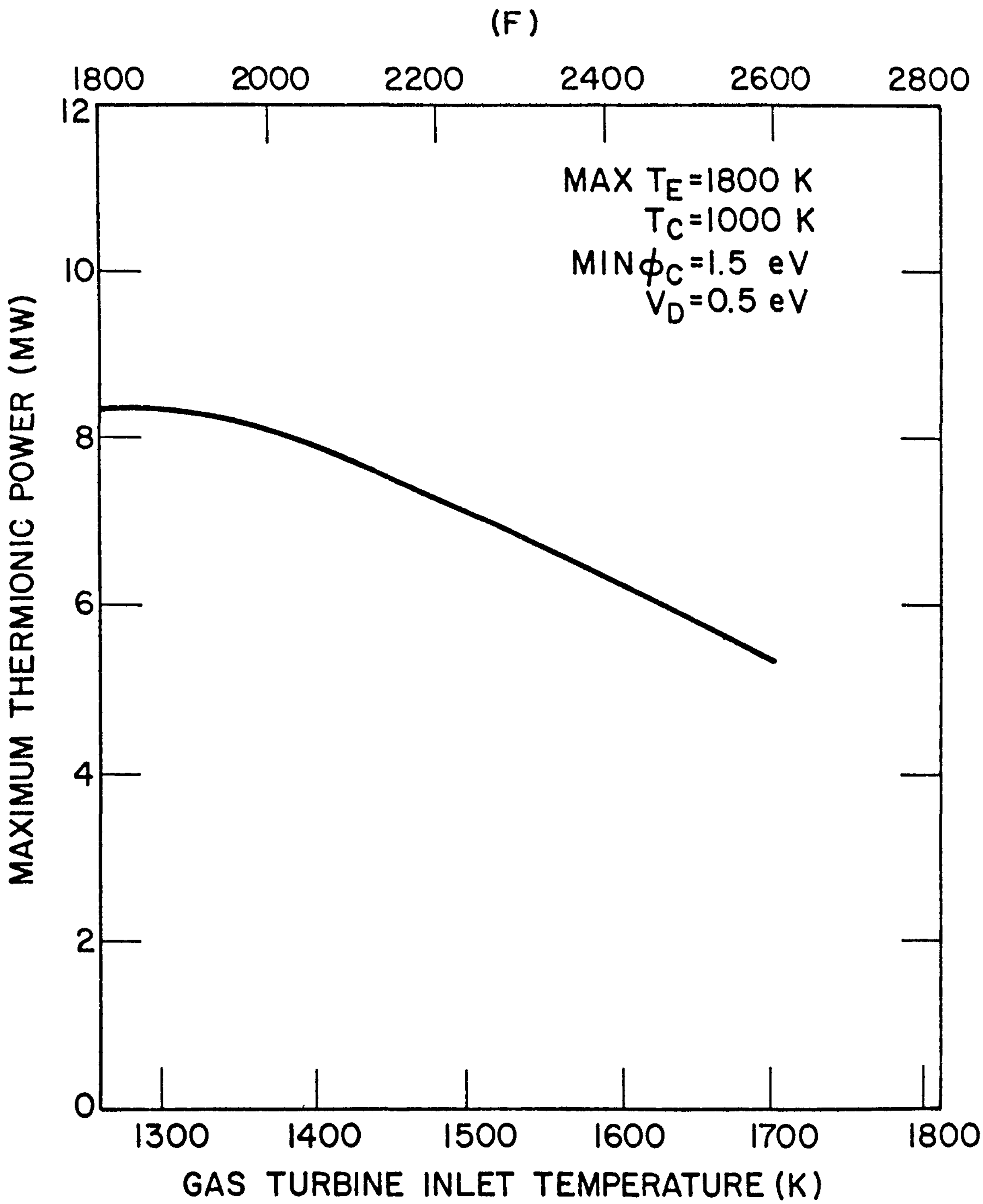


Exhibit D-2.8

Thermionic Power Versus Gas Turbine Inlet Temperature
in Gas Turbine and Combined-Cylce Topping

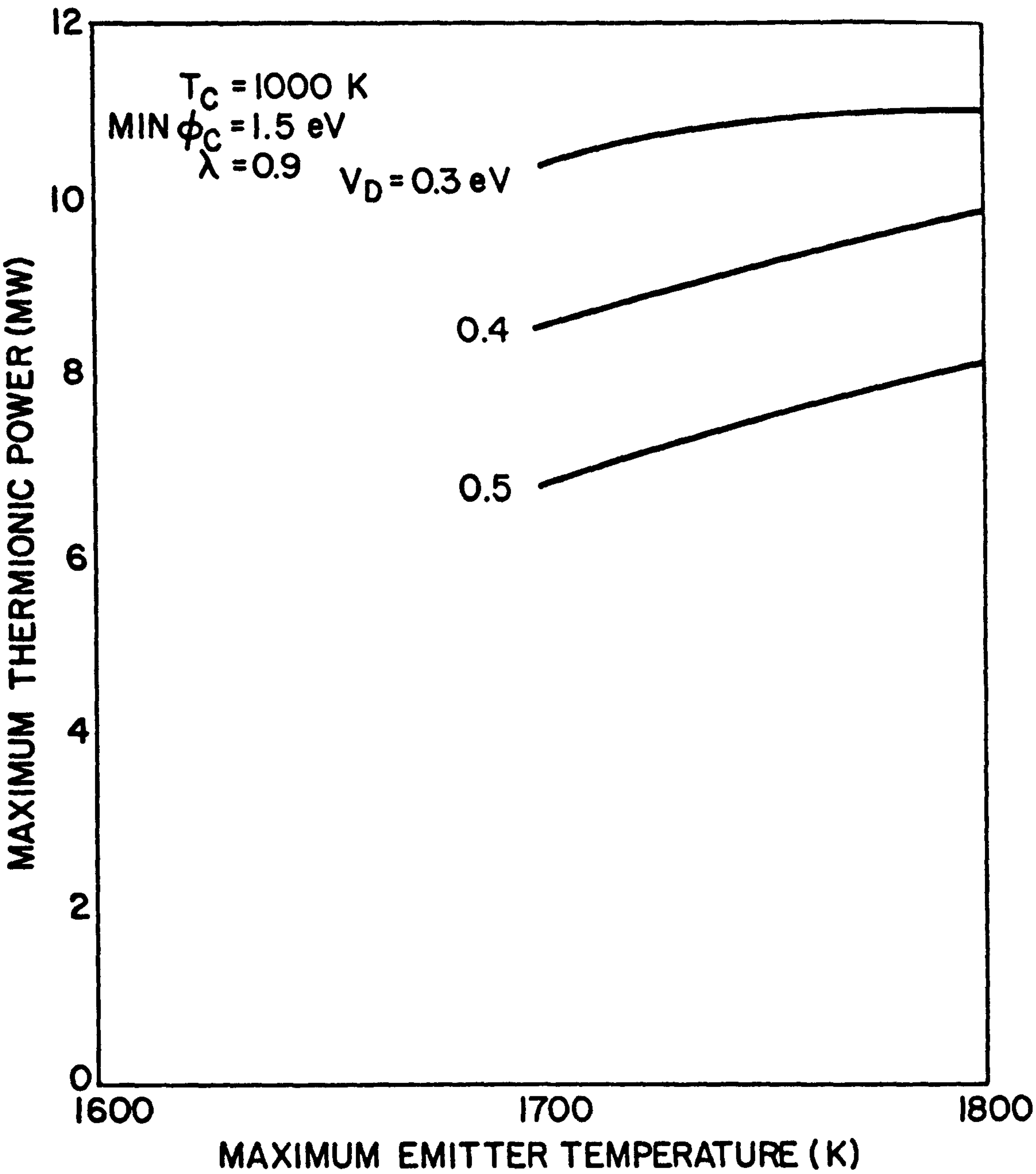


Exhibit D-2.9

Maximum Thermionic Power Versus Emitter Temperature for Gas Turbine and Combined-Cycle Topping

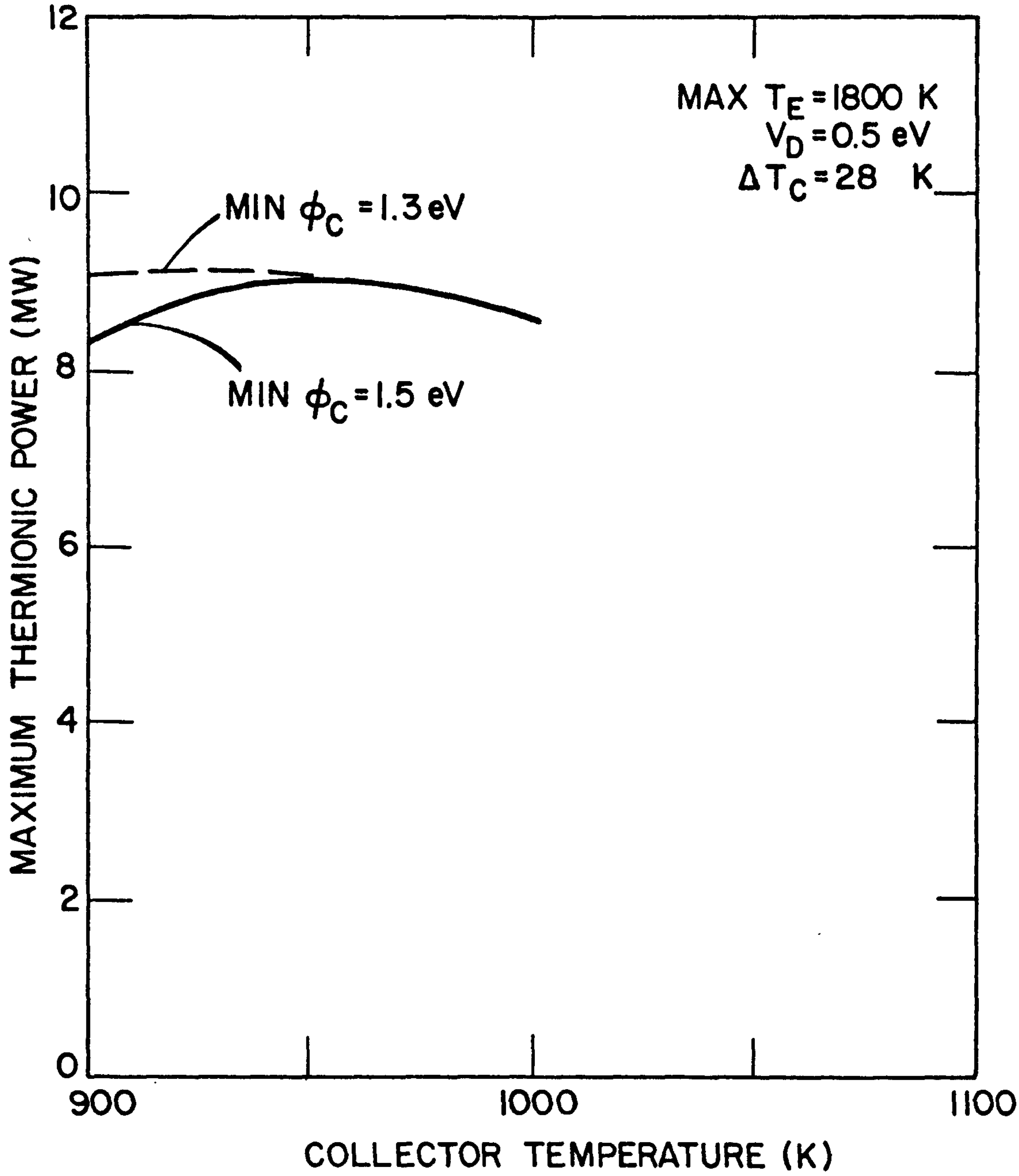


Exhibit D-2.10

Maximum Thermionic Power Versus Collector Temperature for Gas Turbine and Combined-Cycle Topping

The collector temperature in the system is itself dependent on the effectiveness of the heat exchange between the collector and the compressor discharge air. This effect is shown in Exhibit D-2.11, where the thermionic power is plotted against the temperature difference in the heat exchanger. As the temperature difference is increased, the thermionic power generated is decreased due to insufficient collector cooling capacity.

The incremental capital cost of the thermionic topping plant was calculated for varying emitter temperatures as shown in Exhibit D-2.12. For a maximum emitter temperature of 1800 K, with performance as already demonstrated (i.e., V_D = .5 eV), a cost of $475/kW is projected. If an improvement in the converter characteristics is assumed (i.e., V_D = .4 eV), this cost is reduced to $430/kW. A further improvement (i.e., V_D = .3 eV) would result in a cost of $375/kW; this cost is equal to or below that quoted for combined-cycle plants.

The incremental bus bar cost was also calculated for these cases and the results are shown in Exhibit D-2.13. The fuel cost was assumed to be $\$5/10^6$ Btu. The incremental bus bar cost for present day converters operating at 1800 K is shown to be 55 mills/kWh. For an improved thermionic converter (i.e., V_D = .3 eV), the cost could be as low as 50 mills/kWh.

The cost of the thermionic converter is strongly dependent on the amount of tungsten required in the emitter structure. As the emitter thickness is increased, the cost of the thermionic converter is increased, but the voltage drop in the emitter is decreased with a consequent increase of power output from the converter and a decrease of specific cost. This effect is illustrated in Exhibit D-2.14. As the voltage drop in the emitter is decreased (the emitter is made thicker) the power increases and cost decreases.

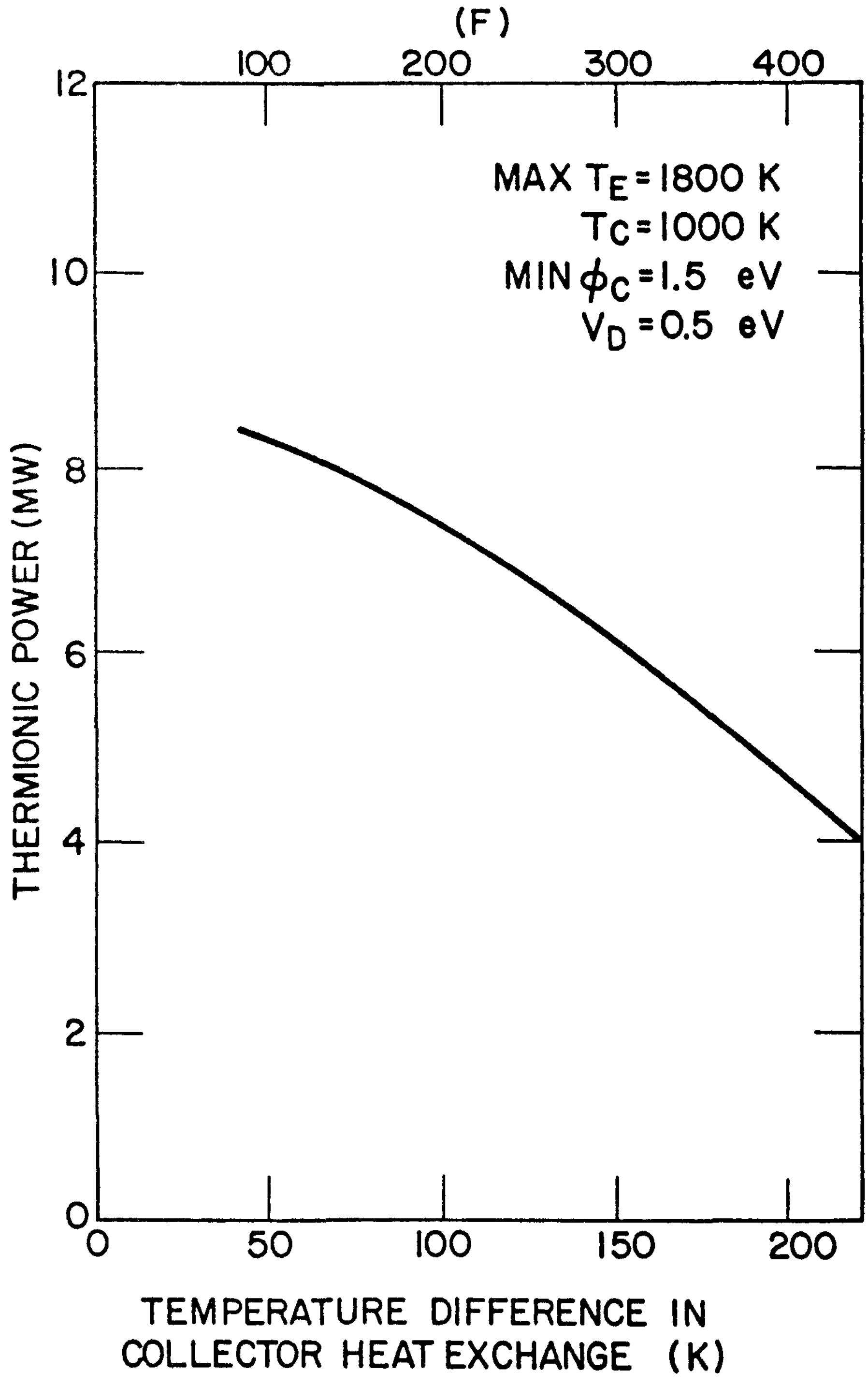


Exhibit D-2.11

Maximum Thermionic Power Output vs. Collector Heat Exchanger Temperature Difference for Gas Turbine and Combined-Cycle Topping

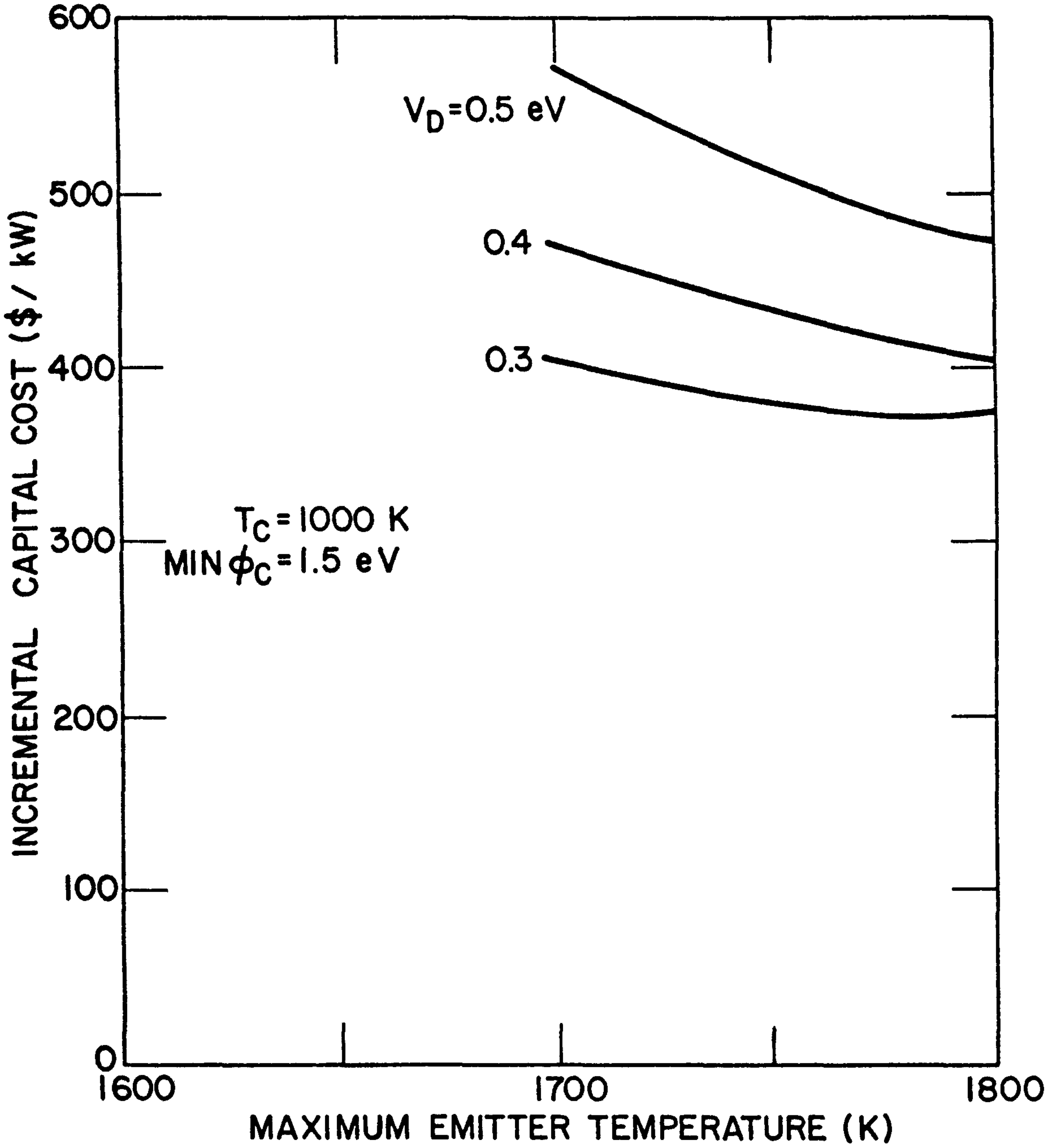


Exhibit D-2.12

Incremental Capital Cost Versus Emitter Temperature for Gas Turbine and Combined-Cycle Topping

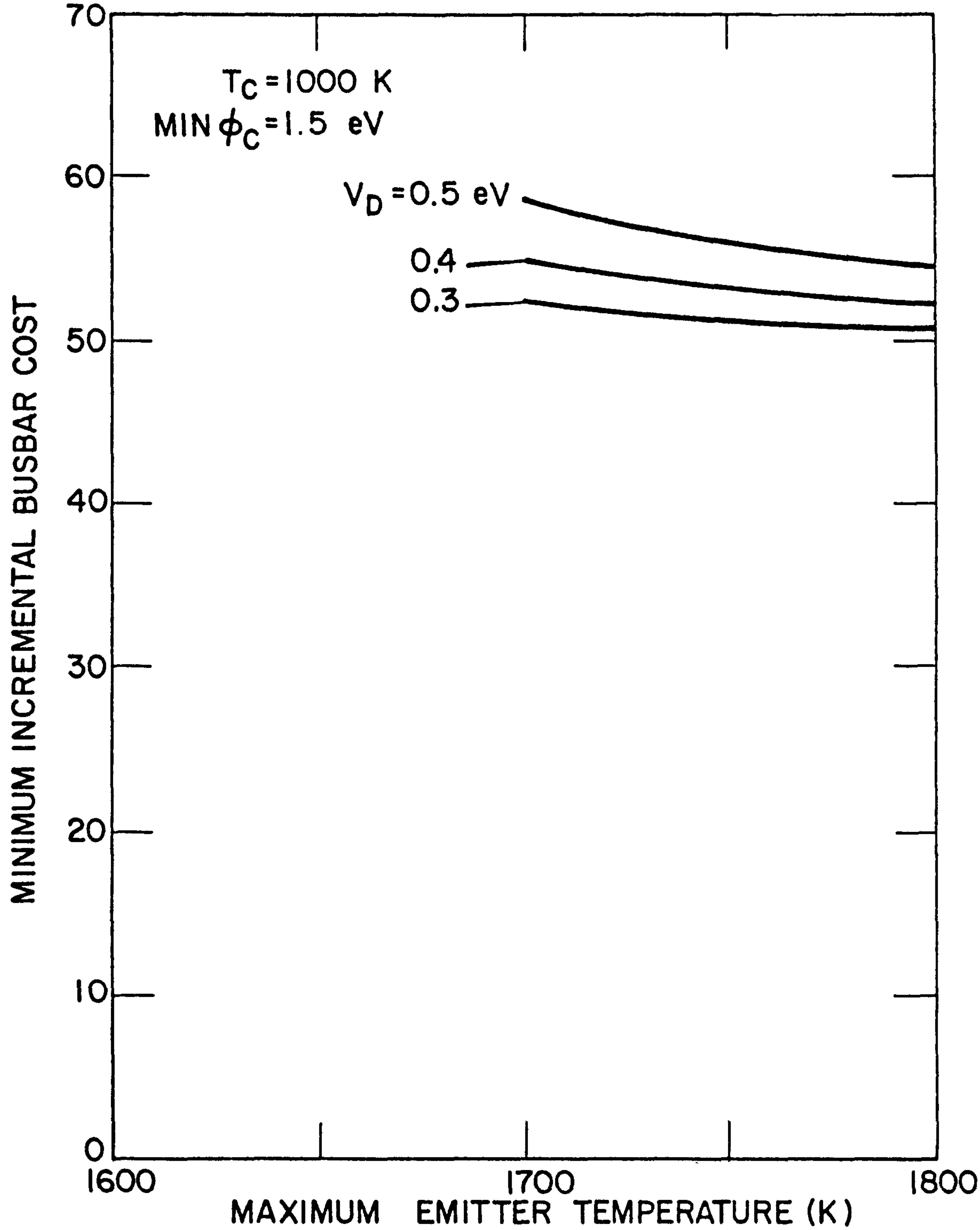


Exhibit D-2.13

Incremental Busbar Cost Versus Emitter Temperature for Gas Turbine and Combined-Cycle Topping

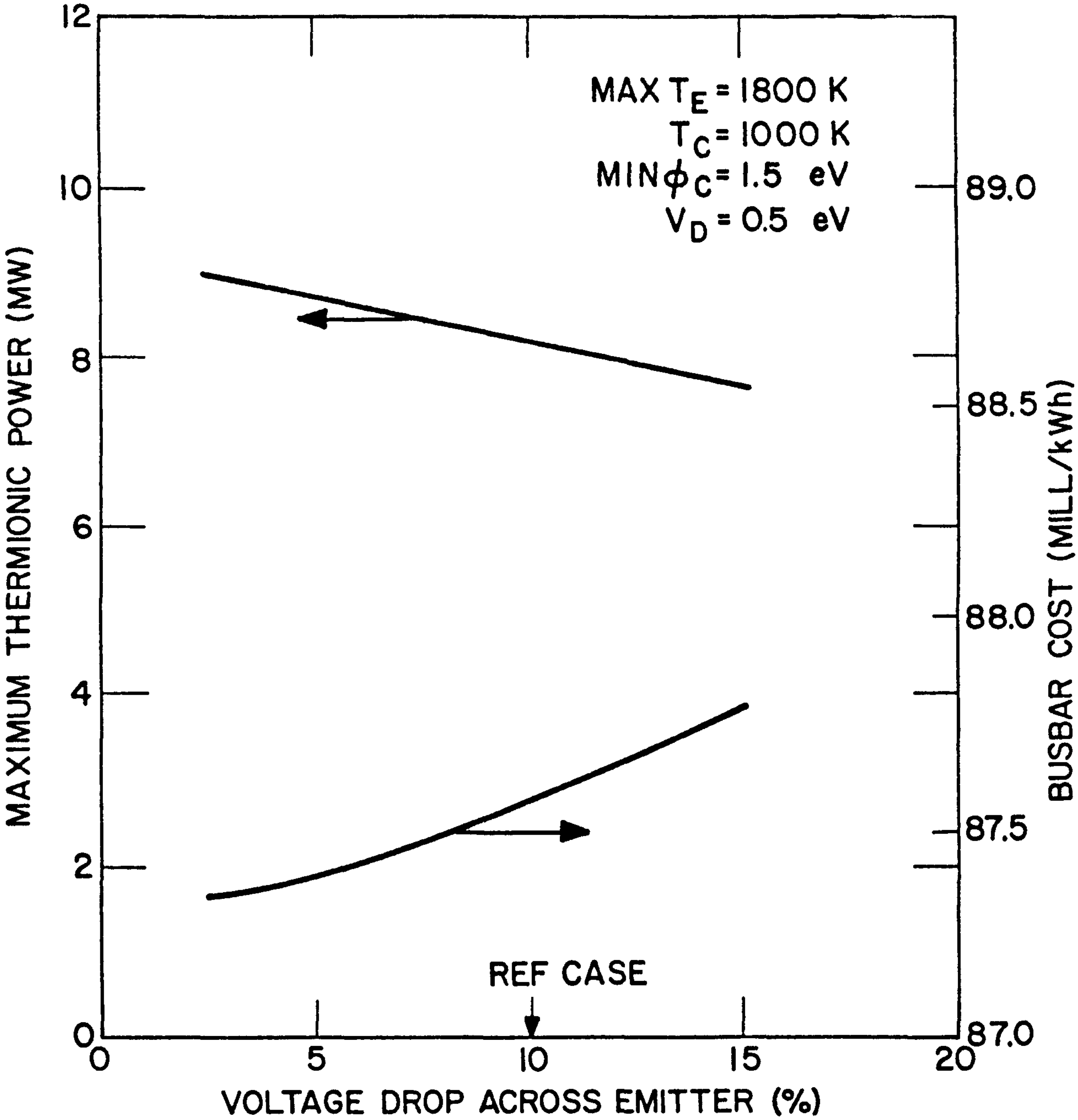


Exhibit D-2.14

Maximum Power Output and Busbar Cost of Electricity as Function of Emitter Voltage Drop in Gas Turbine and Combined-Cycle Topping

Overall, this study concluded that present-day thermionic converter performance is sufficient for use to top a combined-cycle or gas turbine power plant with a significant increase in power production. Thermionic topping of gas turbines appeared to be economically competitive.

These conclusions are of particular note in view of the fact that prior studies of thermionic topping of steam power plants showed such an application to be feasible only with a significantly improved thermionic converter performance. In placing the thermionic converters in a TAM combustor where the temperature and the heat flux are significantly higher than in a steam power plant boiler, the operating point of the thermionic converter is shifted to a region where power could be produced much more economically with today's thermionic performance.

2.4 Thermionic Heat Exchanger Design

A design concept for a thermionic topping cycle, a gas turbine cycle, and a steam turbine bottoming cycle is shown in Exhibit 2-D.15. It uses the THX as an integral design element. Air from the compressor is used to cool the collector side of the THX units. These heated gases then flow into the furnace for combustion with gasified coal. The heat pipes of the THX units draw energy from the combustion gases and convert it to DC power with thermionics. The gas turbine system generates electric power in the conventional manner. The turbine discharge gas passes into a steam system where a steam turbine creates additional electric power. Note that the only basic change from a standard gas turbine combined-cycle system is the thermionic combustor.

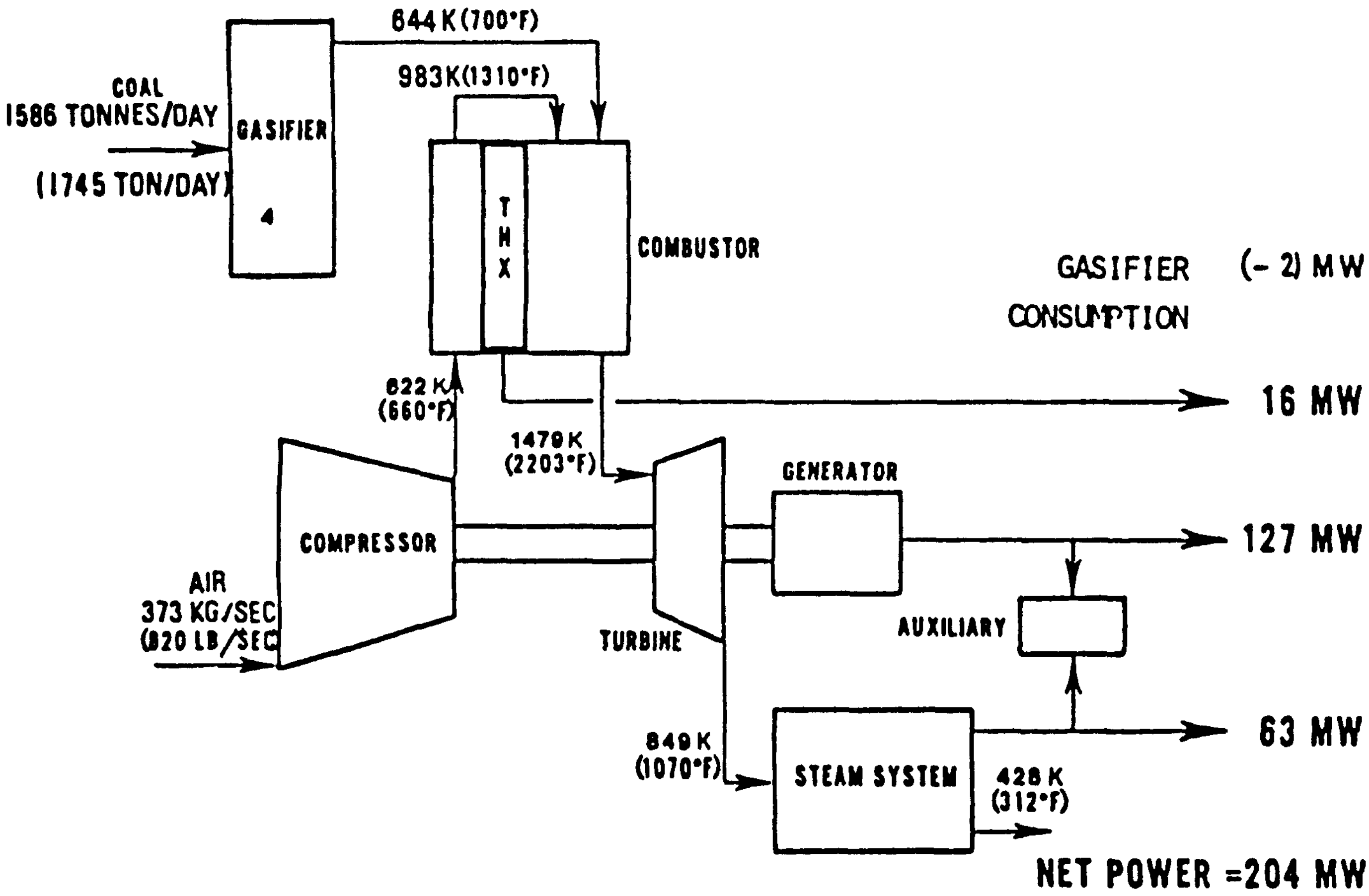


Exhibit D-2.15

Thermionic/Gas Turbine System Schematic

The gas turbine analyzed in this study is shown in Exhibit D-2.16. The large silo combustors used with this system are particularly suitable for thermionic modification. They provide the required physical space needed for the addition of thermionic converters. They also permit combustor changes without affecting the design of the turbine.

A rich-lean combustion process would be used because it is capable of producing low NO_x with all fuels, both nitrogen-bearing and nitrogen-free. More importantly, it can provide the high temperatures preferred for thermionics and still provide low NO_x production rates. Thus it avoids the cost and reliability penalties imposed by post-combustion clean-up.

Three alternatives for integrating the thermionic devices into the combustor were considered:

- Directly heated emitters (i.e., converters on the wall of the combustor).
- Indirect radial array (i.e., horizontal heat pipes with converters).
- Indirect axial array (i.e., vertical hanging heat pipes with converter).

The combustor configuration with the directly heated thermionic converters installed in the combustor wall is shown in Exhibit D-2.17. This design requires the combustor surface area to be approxiamtely four times as great as a conventional configuration.

The indirect axial array of heat pipes, shown in Exhibit D-2.18, permits a short rich combustor length; i.e., 1.68 m, as compared with the 8.1 m required by directly heated converters. Heat pipes integrate the heat flux variations

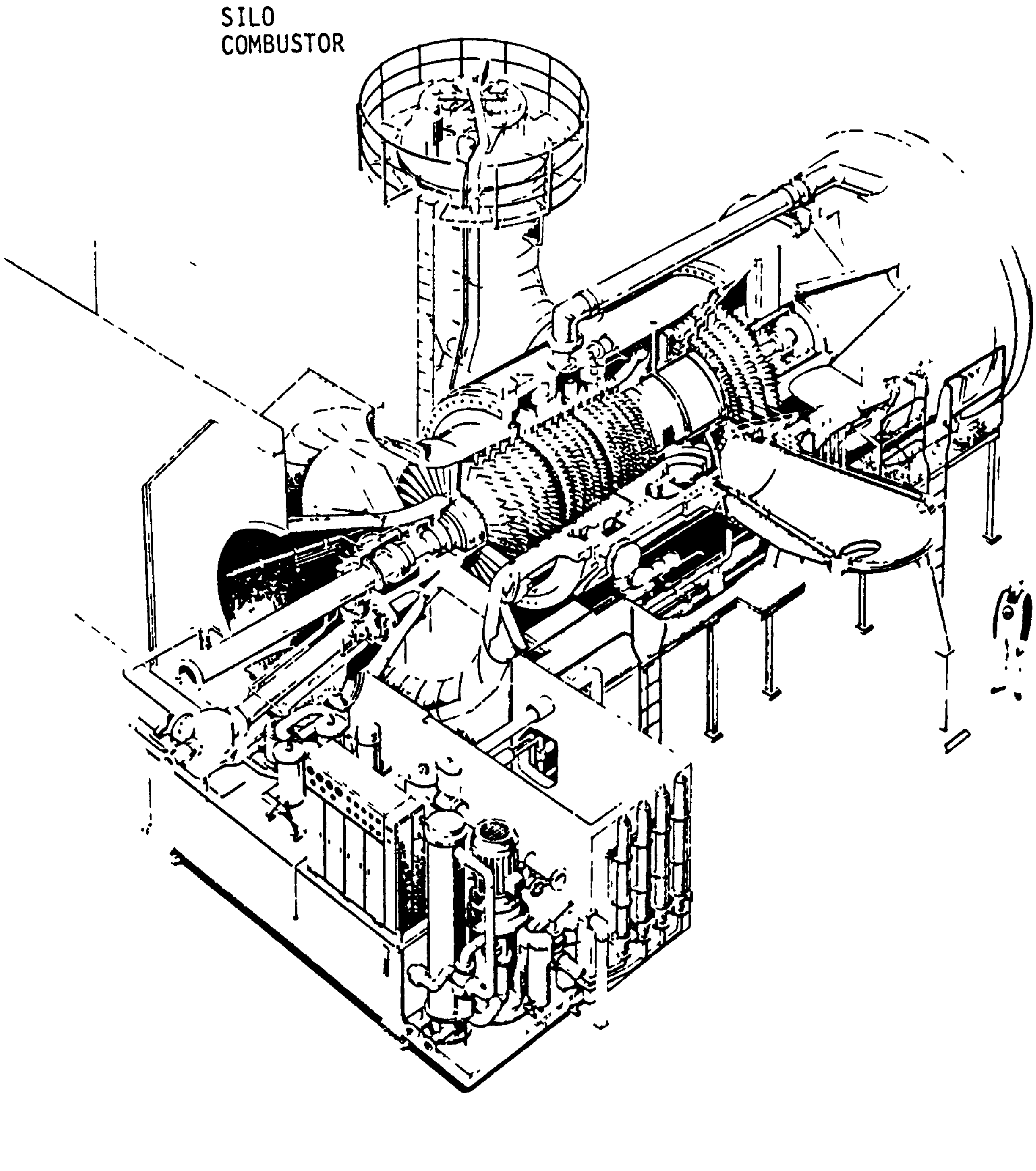
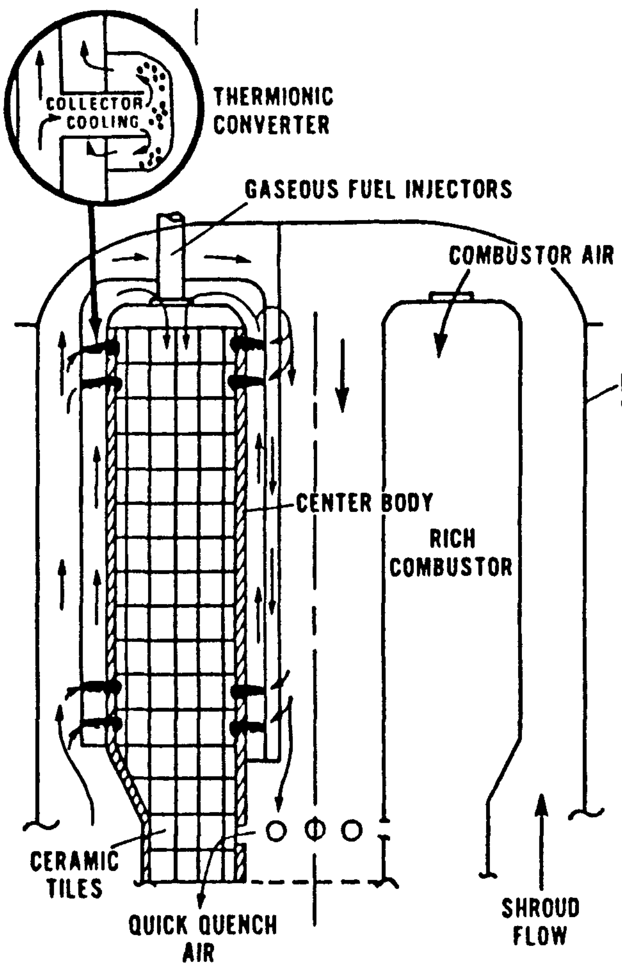
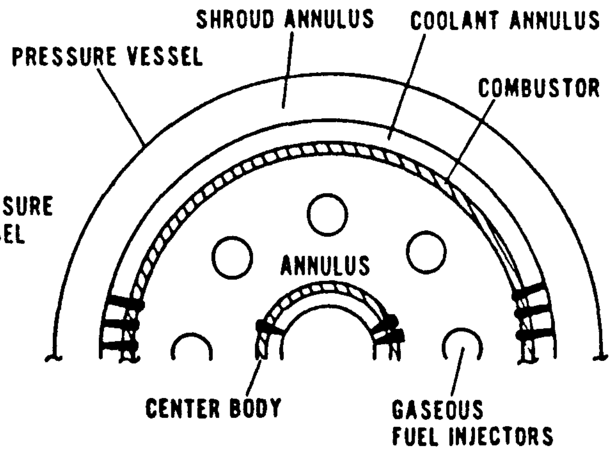


Exhibit D-2.16

UTC/KWt Gas Turbine





• FEATURES

- LONG COMBUSTOR LENGTH TO GET AREA
- VARIABLE HEAT FLUX WITH LENGTH
- SEGREGATED COOLANT PASSAGES
- HIGH COOLANT PRESSURE DROP

Exhibit D-2.17

Direct Thermionic Converter Combustion Chamber

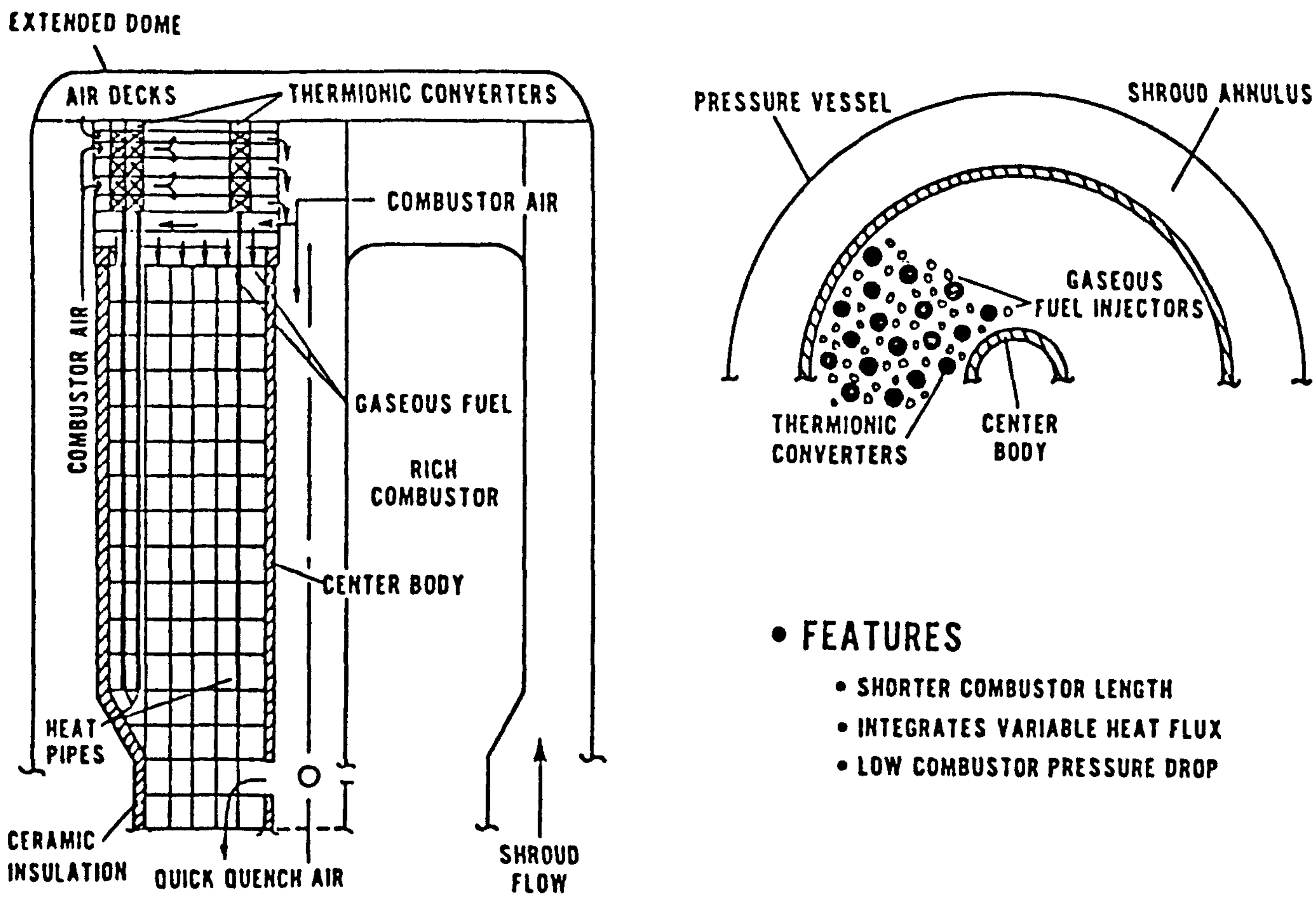


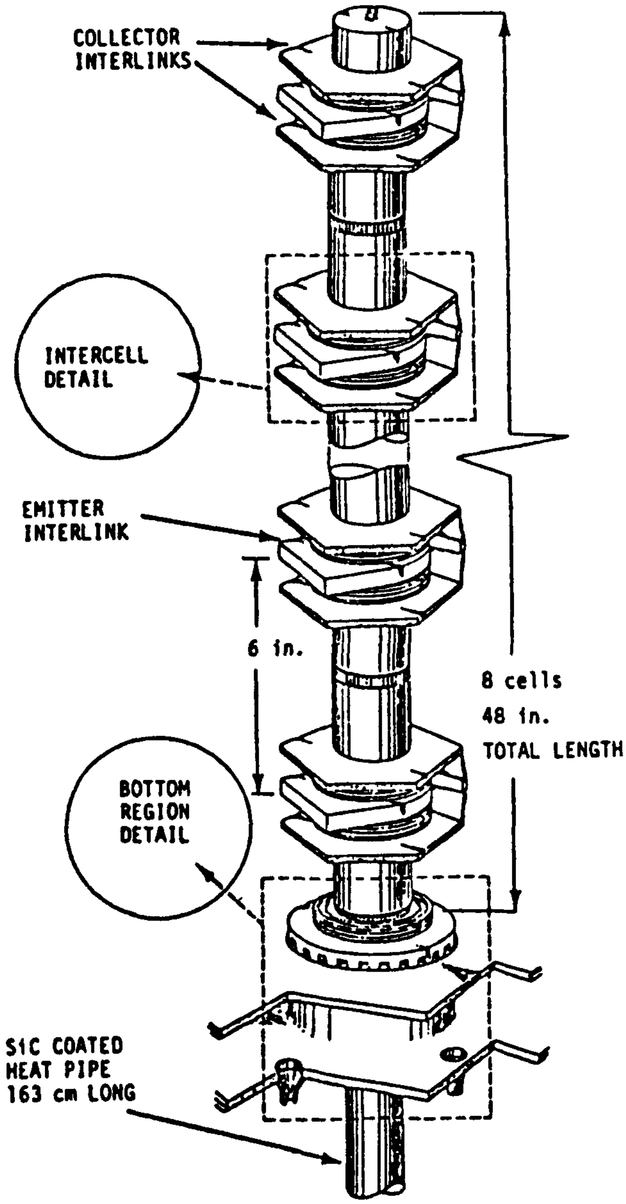
Exhibit D-2.18

Axial Indirect Thermionic Converter Combustion Chamber while protecting the combustor wall from the high combustion temperatures. Since a considerable variation in heat flux exists from the inlet to the exit of the rich combustion zone, the axial heat pipe thermionic system offers significant advantages over directly heated converters.

The thermionic converter/heat exchanger design which best fulfills the requirements is the axial arrayed configuration shown in Exhibit D-2.19. This configuration also accommodates a very important design feature; it facilitates the connection of converters into series-parallel networks, an option which greatly enhances reliability of the power unit. A silicon-carbide-protected Mo-Li heat pipe collects heat from the furnace and delivers it to the converter. Such heat pipe operation has been demonstrated at the thermal fluxes (143 Watts/cm^2 wall, 11.3 kW/cm^2 axial) and temperatures (1700 K) of interest for periods in excess of 15,000 hours. The predicted output of each THX is 7.4 kW at a lead efficiency of 12.9%. The electrical power output of each individual THX cell (i.e., 1 of 8) is 2100 Amperes at .44 Volts (924 Watts) delivered into a power inverter at 95% efficiency. Costs of these units when building 5 power plants per year was projected to be 460 $/kW. The power conditioning costs are estimated at 100 $/kW.

2.5 Thermionic Heat Exchanger Performance and Economics

The performance of the integrated system was parametrically studied by using a UTC Research Center SOAPP computer model, which accurately matches all key system components. Further parametric mapping was accomplished using a simplified Rasor Associates system model which closely approximates the results obtained with the more sophisticated SOAPP system. The effects of varying



Fxhibit D-2.19

Thermionic Heat Fxchanger Design

turbine temperatures, compressor pressure ratios, converter operating temperatures, operating current, and internal losses were all evaluated.

The key to the overall plant efficiency is the temperature at various points. This effect is graphically shown in Exhibit D-2.20 where plant efficiency is plotted against turbine inlet temperature. Also shown are the family of curves for two classes of THX devices. The design point was selected to be 1422 K. Plant efficiency increases from 37.7% to 39.1% with present generation thermionic devices. Note that second generation devices should increase efficiency by another 1.4% to 40.5%.

Exhibit D-2.21 shows a performance and cost summary applied to a 372 MW reference (i.e, untopped) combined-cycle plant. Addition of the thermionic combustor increases the power output of the overall system by 36 MW or 9.7%. Note, not all of this increase is generated by the THX converters. Additional power is generated by the gas turbine because of a favorable change in gas temperature, pressure and flow conditions. The price for this increased power is a small increase in coal consumption. The marginal efficiency (increase in power output divided by increase in fuel energy input) is 70%.

Supporting capital cost and electricity cost data are presented in Exhibits D-2.22 and D-2.23, respectively. The marginal specific cost of adding a thermionic combustor to the reference power plant (added capital cost divided by added power) is 22% higher than the capital cost of the reference plant (1110 $/kW versus 910 $/kW); however, second generation thermionics would reduce the marginal specific cost of the combustor to below that of the baseline plant while increasing plant capacity nearly 20%. The marginal cost-of-electricity

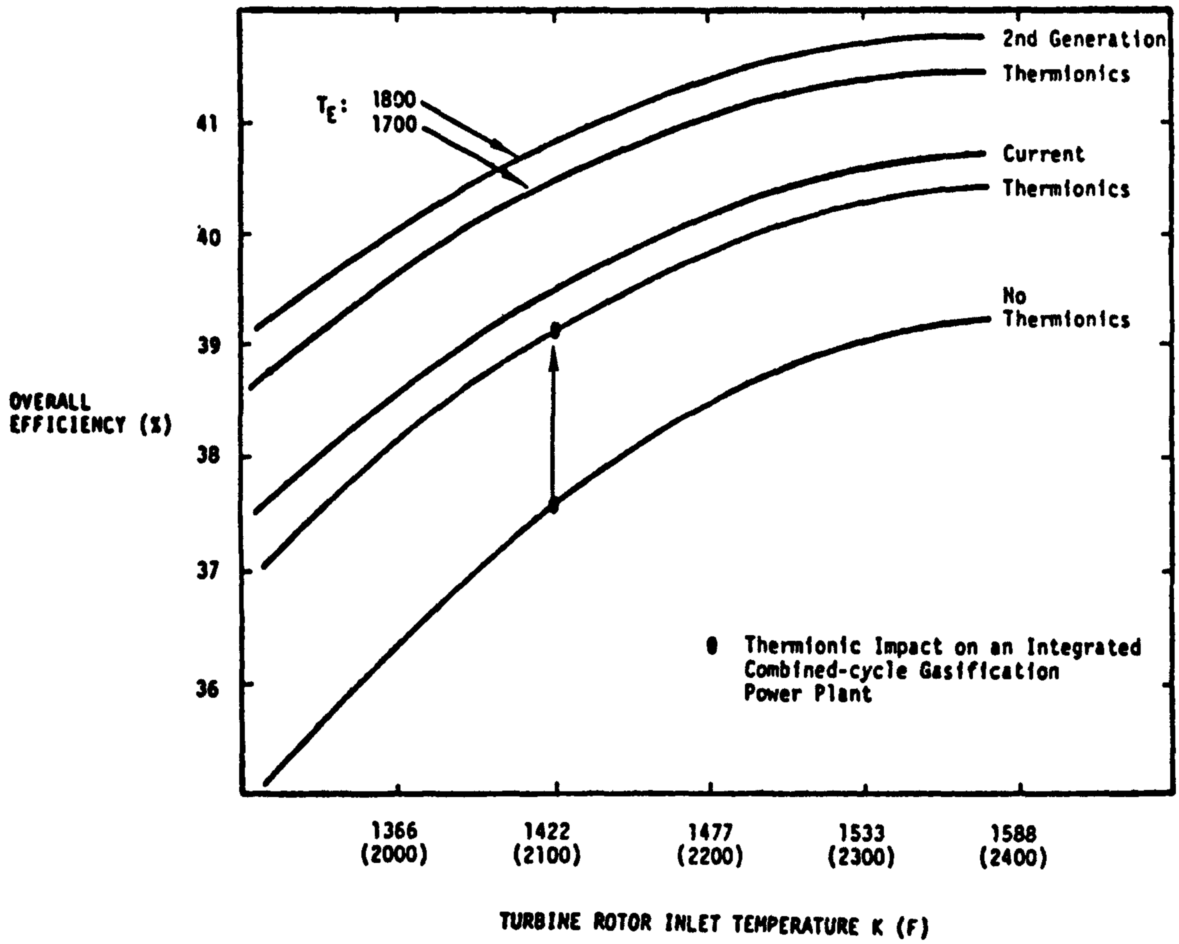


Exhibit D-2.20

System Efficiency Improvement with a Thermionic Combustor Employing a Thermionic Heat Exchanger

EXHIBIT D-2.21

PERFORMANCE AND COST COMPARISON OF SYSTEM
WITH AND WITHOUT THERMIONICS

OVERALL SYSTEM	WITHOUT THERMIONICS	WITH THERMIONICS	MARGINAL
Gas Turbine Power, MW	250	245	4
Steam Turbine Power, MW	126	126	0
Thermionics Power, MW		32	32
Plant Hotel Load, MW	-4	-4	0
Total Power, MW	372	408	36
Efficiency, %	37.7	39.1	70
Coal Consumption, Tons/Day	1655	1745	90
Capital Cost 1979, $M	908	926	1111
Cost-of-Electricity, mills/kWh (Fuel @ $1.35/MBtu)	71.0	70.1	58.8

EXHIBIT D-2.22

BREAKDOWN OF CAPITAL COST
FOR THX TOPPING SYSTEM

	372 MW W/O Thermionics		408 MW W/ Thermionics		36 MW Marginal
	($M)	($/kW)	($M)	($/kW)	($/kW)
Gas Turbine (2)	31.3	84.1	31.8	77.9	13.9
Thermionics	---	---	17.8	43.6	494.4
Steam System	41.8	112.4	42.1	103.2	8.3
Balance of Plant	24.3	65.3	26.9	65.9	72.2
Architect/Engr.	22.4	60.2	27.3	66.9	136.1
Contingency	14.6	39.2	17.8	43.6	88.8
Interest and Escalation	40.0	107.5	48.8	119.6	244.4
TOTAL POWER PLANT	174.4	468.8	212.0	521.0	1058.1
Gasification System	254.0	682.8	266.7	653.7	352.8
TOTAL SITE COST (1982 $)	428.4	1151.6	478.7	1174.7	1411.0
Mid-1979 $	337.6	908.0	377.7	926.0	1111.0

EXHIBIT D-2.23

BREAKDOWN OF COST-OF-ELECTRICITY
FOR THE THX TOPPING SYSTEM

	Baseline	Thermionic	Marginal
Capital	26.6	27.2	32.6
O&M	19.4	18.9	12.8
Fuel	25.0	24.0	13.4
(1.35 $/MBtu) (1979 $)			
TOTAL	71.0	70.1	58.8

with the addition of thermionic combustors is 58.8 mills/kWh, which is 17% below the baseline plant (71.0 mills/kWh), with coal costs at 1.35 $/MBtu.

The savings projected by this study are preliminary, and further optimization of both the THX design and the overall combustor design is possible. Reduction in THX and the combustor costs would likely result from more extensive design effort.

Thus, preliminary projections have suggested that currently demonstrated levels of thermionic performance could potentially improve the efficiency of a combined-cycle system as much as 2 percentage points, or the equivalent of raising the turbine inlet temperature by 111 K (200°F). The use of advanced converters could provide an efficiency increase equivalent to a 220 K (400°F) turbine inlet temperature increase.

SECTION 2 - REFERENCES

1. Hatsopoulos, G.N., and Gyftopoulos, E.P., Thermionic Energy Conversion, Vol. 1, Processes and Devices. The MIT Press, Cambridge, MA (1973).

2. Thermo Electron Corporation, "Manufacturing Cost of Flame-Heater Thermionic Converters," Report No. TE 4258-163-79 (April, 1979).

3. Electric Power Research Institute, Technical Assessment Guide, EPRI Special Report No. PS-1201-SR (July, 1979).

3. UTILITY BASE-LOAD GENERATION

3.1 New Thermionic Heat Exchanger Coal-Fired Power Plant

Steam power plant efficiency has been increasing since the turn of the century (see Exhibit D-3.1). Business pressures to reduce expenses and inflation of fuel costs have been driving factors, as have improvements in advancing technologies.

Perhaps the single most important means by which plant efficiency has been increased has been the use of higher peak cycle temperatures. The inability to achieve still higher efficiencies from this approach stems from several sources, including: increased steam pressure requirements, reduced material strengths at higher temperatures, the decomposition of water at temperatures over about 870 K, and increasingly stringent air pollution regulations. The steam cycle has clearly reached the point of diminishing returns with temperatures near 870 K. Yet the peak combustion temperatures available, even in conventional systems, are much higher. The flame temperatures in coal-fired furnaces are typically greater than 1900 K, for example. Clearly, an additional thermodynamic cycle which is compatible with the steam system and can take advantage of these higher temperatures is preferable. The thermionic converter provides such a cycle. It also has the distinct advantage of providing high conversion efficiency with relatively elevated heat rejection temperatures (i.e., 670 K to 1020 K). In fact, the heat rejection temperature is sufficiently high that the conventional steam cycle can further process this "waste" heat as originally designed to generate power. If a steam cycle operates between 310 K and 870 K

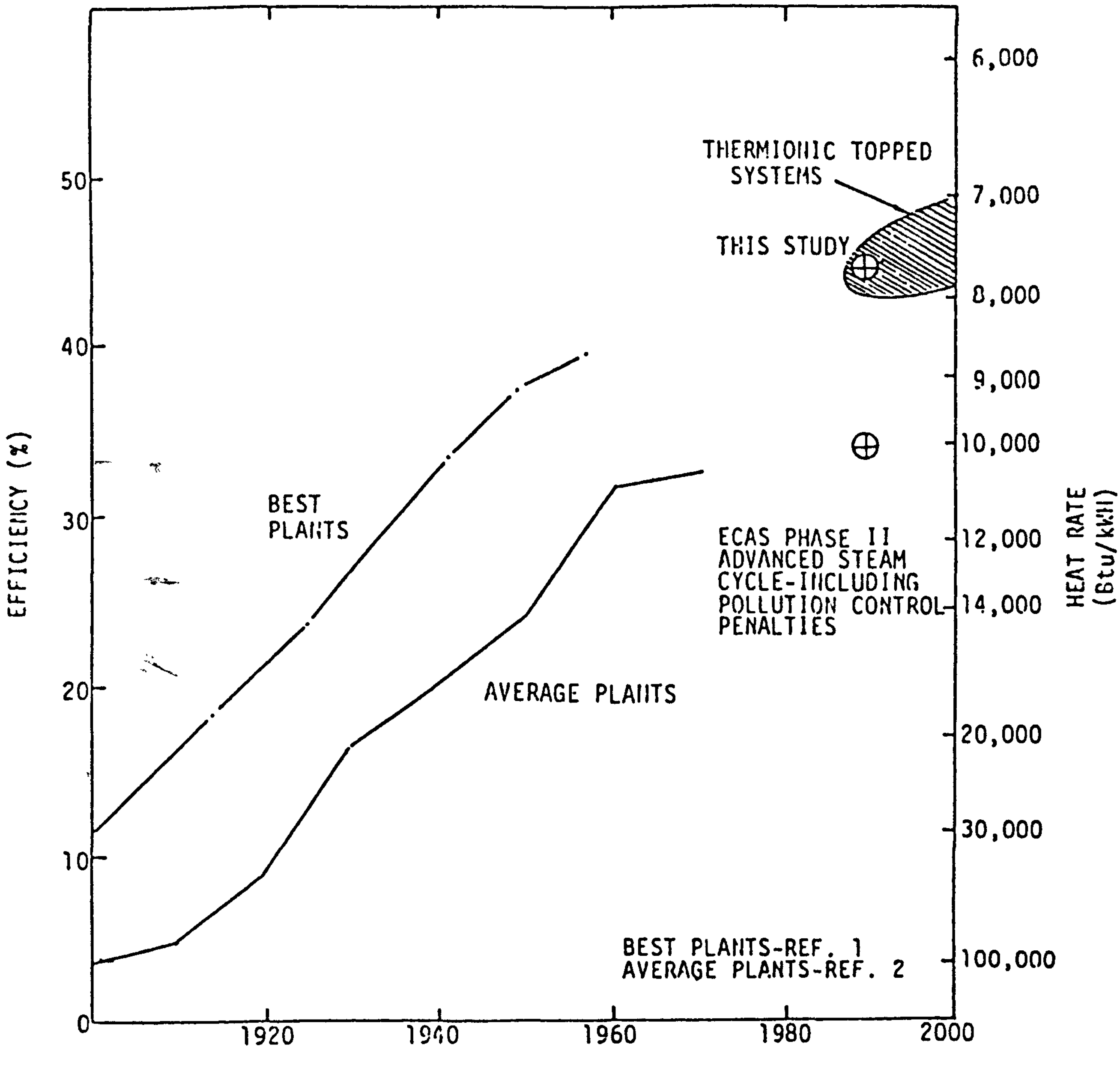


Exhibit D-3.1

The Historical Performance of U.S. Central Station Power Plants (i.e., 100 F and 1100 F), a maximum Carnot efficiency of 64% can be expected as seen in Exhibit D-3.2. However, if a thermionic topping cycle which operates between 870 K and 2200 K (i.e., 1100 F and 3500 F) is added to the steam cycle, a total Carnot cycle efficiency of 86% results. The maximum efficiency has been enhanced by a substantial 22 percentage points. Although Carnot efficiencies cannot be realized in practice, they do provide insight into the maximum theoretical limits.

During the Program several studies were made to evaluate the economics of thermionic converter topping of steam power plants. In most of those studies [1,2] the requirements for optimum overall performance were subordinated in favor of using conventional steam systems and furnaces. One purpose of the studies described in this Section was to perform a parametric analysis to select the operating point of a more nearly optimized, integrated thermionic power plant, using a furnace better adapted to thermionic conversion. Another objective was to compare its performance with other advanced energy conversion systems and with conventional systems. In the first of these studies, Foster-Wheeler Development Corporation designed the furnance and performed the heat-train integration. Rasor Associates Incorporated designed the thermionic heat exchanger and Bechtel National Incorporated designed the steam cycle and the balance-of-plant. All participants provided costs in their area, but Bechtel had the responsibility for the economic analysis of the plant.

One of the most important aspects of any study is the ground rules assumed. The ECAS Phase II advanced steam cycle [3] was taken as the standard for the bottoming advanced steam cycle. Constant 1975 dollars were used with escalation rates and interest rates identical to those used in ECAS. The

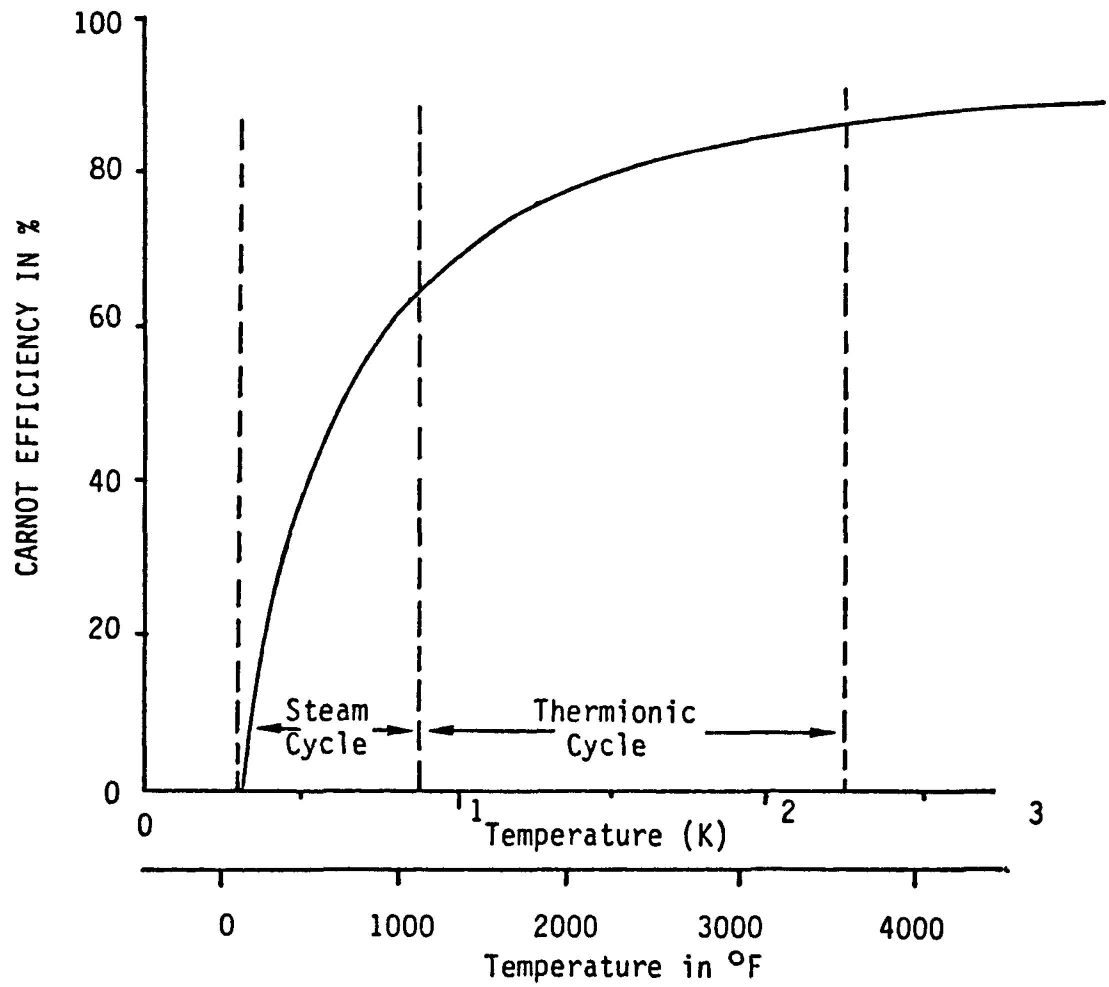


Exhibit D-3.2

Carnot Cycle Efficiency as a Function of Maximum Available Temperature. Rejection Temperature Assumed 100 F (311K) results are reported in 1975 dollars to permit direct comparison to the ECAS studies. Otherwise, the economic methods of the Electric Power Research Institute (EPRI) were used [4].

Exhibit D-3.3 shows a simplified schematic of the power system. Each THX produces between 200 and 400 kW of power while transferring heat from the furnace to the steam cycle. In the THX concept, thermionic converters are mounted on one end of a heat pipe which is used to remove heat from the furnace. Exhibit D-3.4 shows the conceptual design. Heat transfer, which takes place almost isothermally, is accomplished by lithium vapor which evaporates off the interior pipe walls of those portions of the heat pipes which hang inside the furnace. The vapor condenses at the upper end of the heat pipe, at the interior pipe walls, and flows by capillary action from there to the bottom to be evaporated again. Steam pipes incorporated into the collectors of the converters remove reject heat from the thermionic system and deliver heat to the steam cycle.

Several potential levels of thermionic converter performance were analyzed: fully developed, second generation, and first generation (i.e., 1980 performance). The distinction between the levels of performance is that the second-generation level assumes a modest increase in efficiency due to improvements in surface physics or reductions in plasma losses. For fully developed thermionic converters, a reduction of both surface and plasma energy losses in the converter to the lowest levels measured in the laboratory is assumed.

The coal-fired furnace for the system is shown in Exhibits D-3.5 and D-3.6. The concept combines the high-temperature gas heat exchanger, furnace,

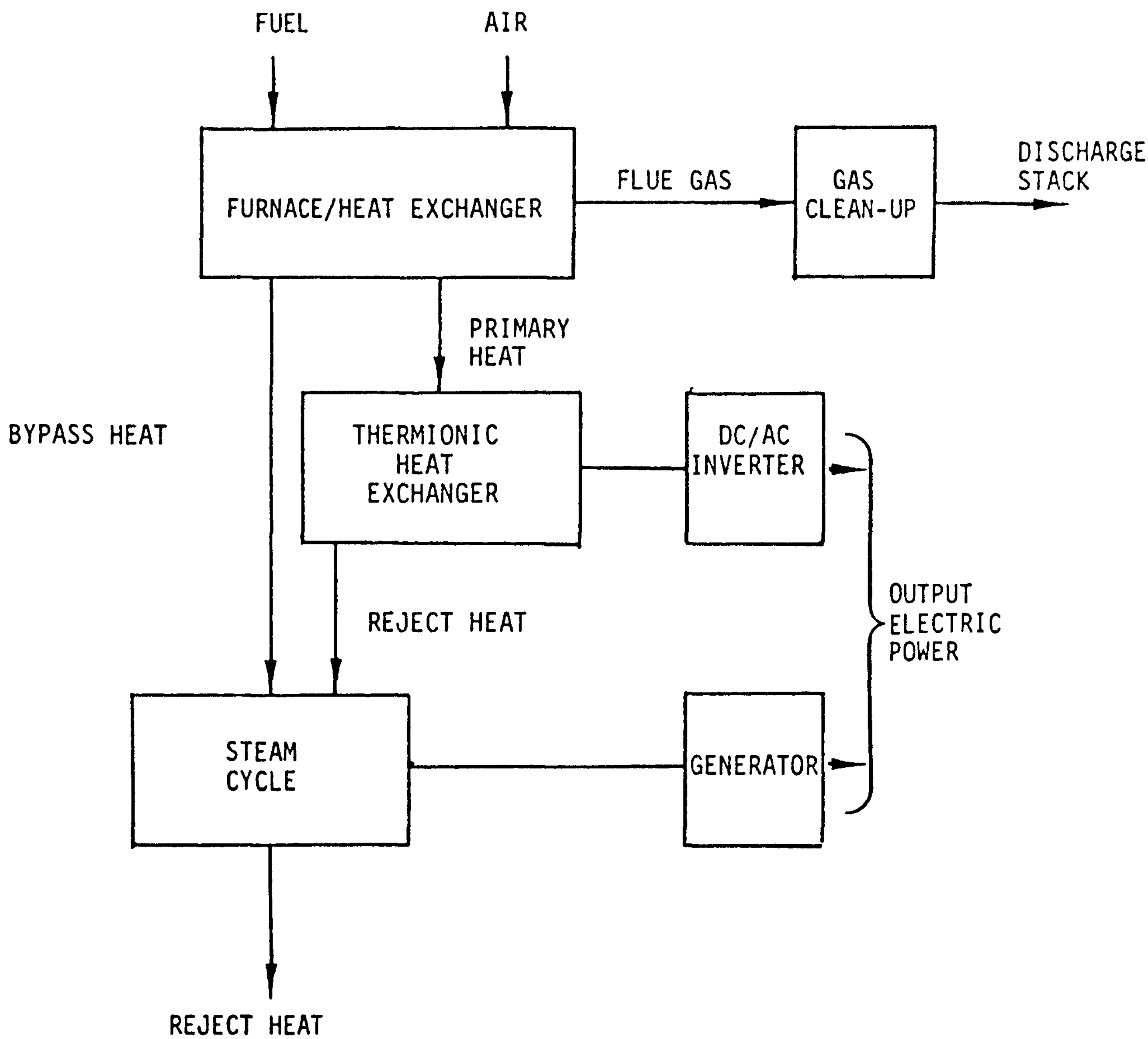


Exhibit D-3.3

Basic Cycle Schematic of A Thermionic Power Plant

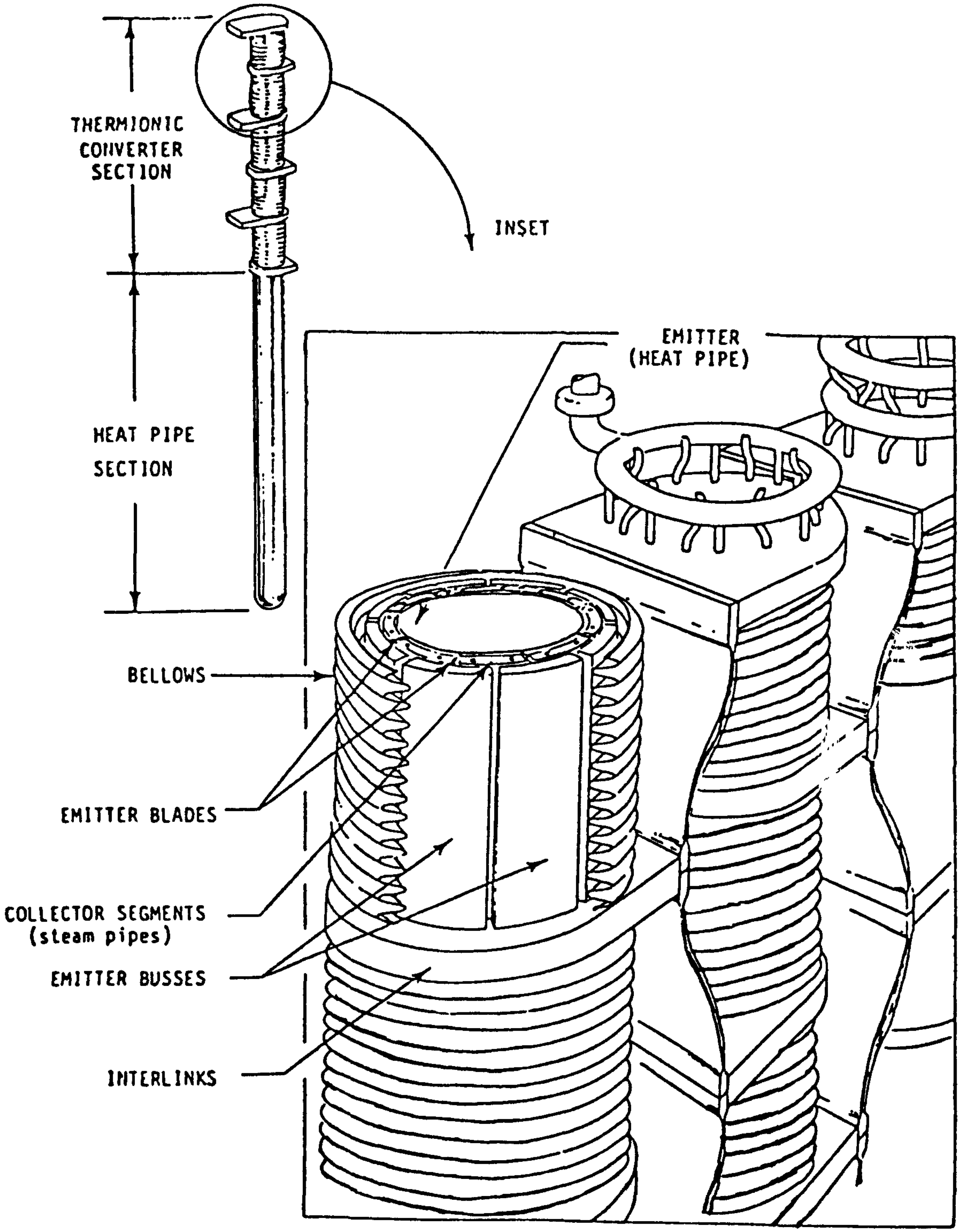


Exhibit D-3.4

Thermionic Heat Exchanger

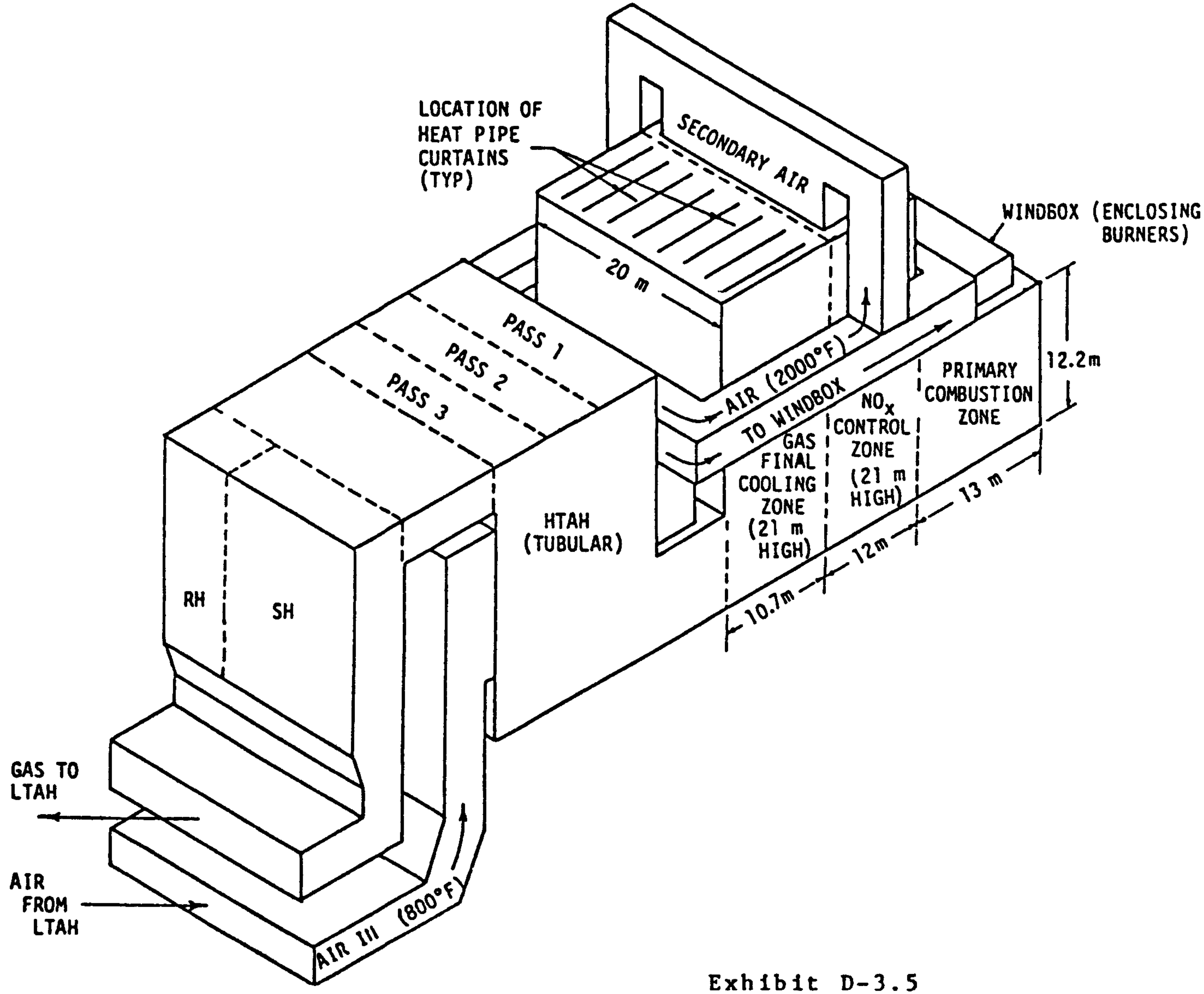


Exhibit D-3.5

Furnace/Heat Train Layout

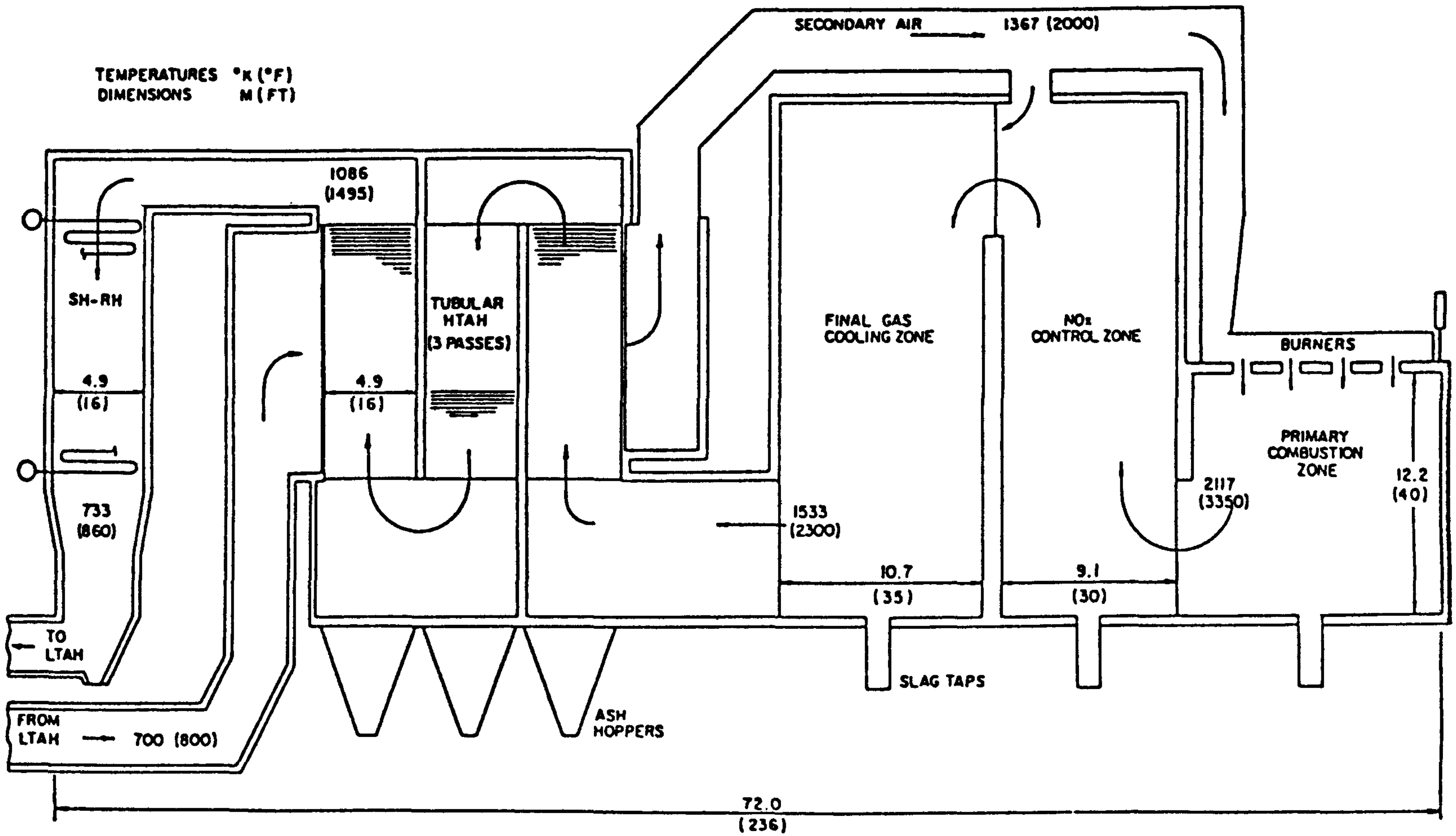


Exhibit D-3.6

Typical Elevation View of a Furnace/Heat Recovery Train for 1800 MW Thermal Input, 1367 K Secondary Air Temperature and converters as one system. The THX modules are mounted above the furnace and are aligned in banks or rows in close proximity, forming heat pipe "curtains" in the furnace. Details of the design, the engineering trade-offs, and input technical data are described in [5, 6]. The most demanding design requirement was compliance with the ECAS Phase II NO_x emission restrictions. These restrictions, .7 lbm/10^6 Btu, were followed so that the results of this study would be directly comparable with other ECAS study results.

A 3500 psig/1370 K (2000 F)/810 K (1000 F) steam cycle was selected as the most efficient, assuming state-of-the-art. Steam cycle parameters were not varied. Plant size was adjusted by specifying the fuel heat input to the furnace. A parametric analysis was performed using a cost and performance computer model. To model the performance and cost of the furnace and air preheat system, nine specific designs were developed. Air preheat temperatures of 1090 K, 1370 K, and 1640 K were each treated at three thermal power levels: 900, 1800, and 2700 MWt. These designs and the scaling relationships between them were used to model the power plant. A thermionic subroutine matches the design of the THX's and the furnace conditions. The main program models the overall plant and computes total costs and performance. Details of the computer code are included in [7].

A parametric analysis was performed to determine which of the large number of factors in the power plant designs have a significant effect and which have minimal effect on cost and efficiency. Those factors were then explored to determine what the optimum points were and how they were achieved.

The parametric analysis revealed that the lowest cost of electricity is obtained with a 2700 MWt plant, an air preheat temperature of 1090 K, and third-generation thermionics. The performance and cost of such a plant are summarized in Exhibits D-3.7 and D-3.8, respectively. Efficiencies of 46% were calculated for other air preheats. However, at $1/MBtu fuel cost, the 1090 K system yielded the lowest cost of electricity. The thermionic topped cycle results compare favorably with the ECAS steam plant [3]. Data from two thermionic generations and the ECAS steam plant are shown in Exhibit D-3.9.

Since the optimum design in terms of cost of electricity was achieved with the lowest air preheat temperature selected for the study, it is recommended that plant designs with air preheater temperatures lower than 1090 K be studied. The lower temperatures would permit the use of metallic, high-temperature air heaters instead of silicon carbide heat exchangers and thus offer further reductions in the cost of electricity. Another area for future study is higher efficiency, low-temperature thermionic converters. In a plant with a 1090 K air preheat temperature, some of the THX modules operate at a relatively low emitter temperature (1310 K). These THX's have low thermionic performance and therefore high specific cost (in dollars per kW). This suggests that further reduction in the cost of electricity may result from increasing efficiency or eliminating those units.

An increase in efficiency and a significant savings in plant operating cost could also be achieved if the steam used to reheat the flue gas after flue gas desulfurization could be reduced or eliminated. This could be accomplished by using a cyclic reheat system which extracts heat from the

EXHIBIT D-3.7

PERFORMANCE CHARACTERISTICS OF
COAL-FIRED THERMIONIC POWER PLANT

Fuel heat input, MW	2,700
Thermionic power output, MW	486
Turbine generator output, MW	817
Total output (gross), MW	1,303
Auxiliary power loss, MW	94
Net power output, MW	1,209
Efficiencies, %	
Thermionic power conversion	31.2
Steam cycle power conversion	43.1
Plant thermodynamic	54.7
Plant overall	44.8

EXHIBIT D-3.8

ECONOMICS OF COAL-FIRED THERMIONIC POWER PLANT

Plant construction time, years	5.6
Costs in millions of dollars	
Estimated plant cost (1975$)	608.6
Estimated plant cost in year of construction start	2,062.6
Escalation and interest during construction (IDC)	1,158.3
Capital cost in year 2000	3,220.9
Capital cost for plant completed in year 2000, expressed in 1975 dollars	667.1
Capital cost per installed kilowatt, $/kW	551.8
Levelized cost of electricity at 70% capacity factor in the year 2000 expressed in 1975 dollars, mills/kWh	
Capital	16.2
Fuel (a) (b)	15.3
Operating and maintenance (a)	4.8
Total	36.3

(a) Levelizing factor = 2.004

(b) Fuel cost = $\$1.00/10^6$ Btu (1975$)

EXHIBIT D-3.9

COMPARISON OF THERMIONIC TOPPED POWER PLANTS WITH STEAM PLANT

Plant (Electrical Output)	Efficiency (Percent)	Capital Cost ($/kW 1975)	COE (mills/kWh)
Thermionic (Fully developed) (1209 MW)	44.8	551.8	36.3
Thermionic (Next Generation) (1092 MW)	40.5	648.0	40.3
Steam (747.2 MW)	33.8	580.0	44.7

flue gas entering the scrubber and uses that heat to reheat stack gas. A Bechtel study reviewed an operating unit in Texas and concluded that an approximately 2.0 mill/kWh cost-of-electricity savings could be achieved in a 1000 MW plant by changing from the system used in this study to a cyclic reheat system. Although the increase in efficiency resulting from this change in stack gas reheat was not calculated, a rough estimate predicts an increase of more than one percentage point in overall plant efficiency. This savings would apply to both thermionics and the conventional steam cycle, but not to most other advanced plant designs because they do not use flue gas desulfurization systems.

Capital cost and the overall efficiency of the thermionic power plant are shown in Exhibit D-3.10 in comparison with other systems. The thermionic plant data represents the selected case in this study. The other data are taken from [3]. The thermionic plant has about the same capital costs as the conventional steam plant and has an overall efficiency that is about 12 percentage points (34% to 46%) higher. In addition, the capital cost of the thermionic plant is competitive with other advanced power plant concepts, and only open cycle MHD is shown with a higher efficiency.

The levelized cost-of-electricity as a function of plant efficiency is shown in Exhibit D-3.11 for the same nine concepts. The better plants will be positioned in the lower right-hand corner as in the previous figure. Again, we note that the thermionic system is second only to an open-cycle MHD plant.

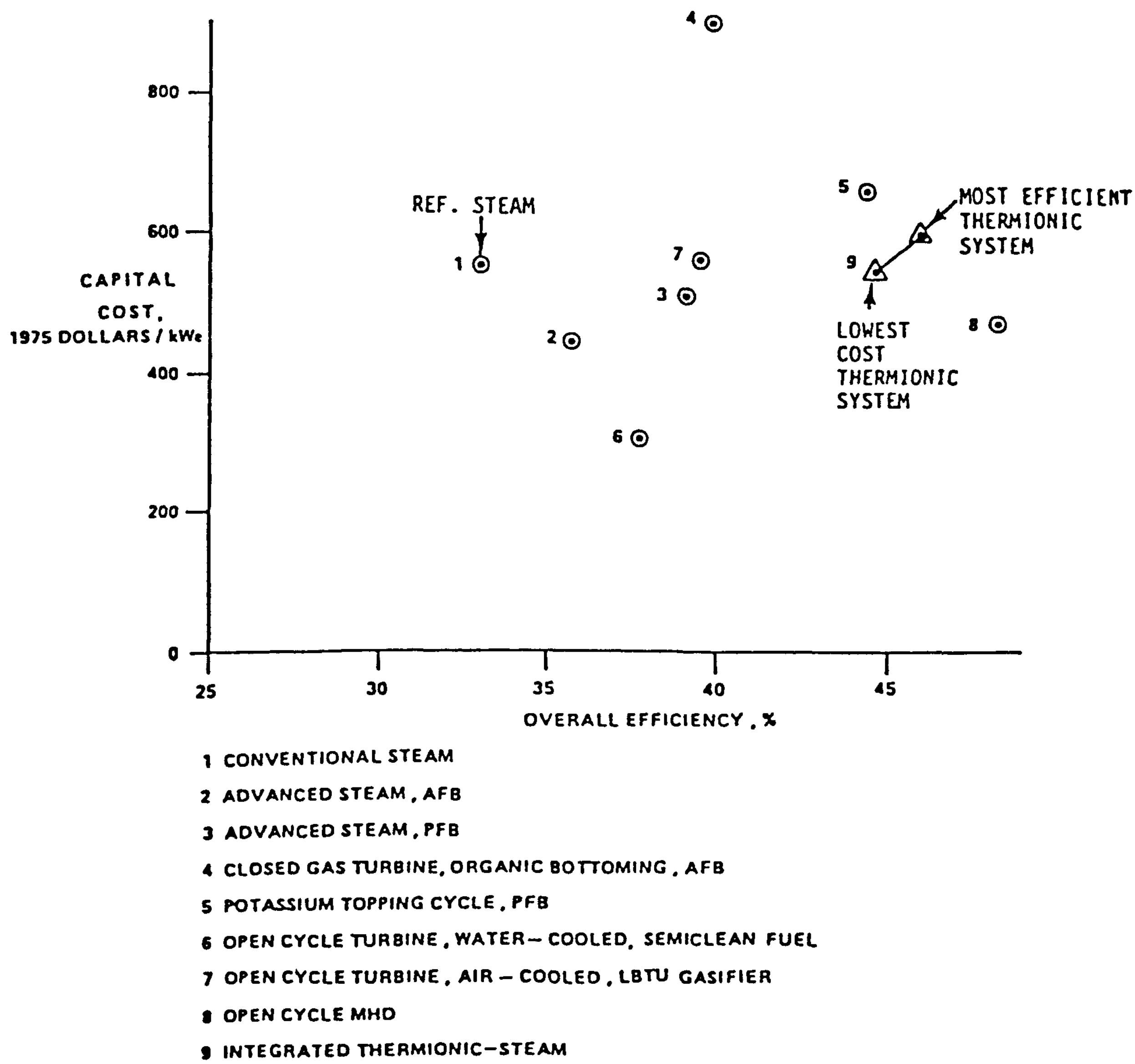
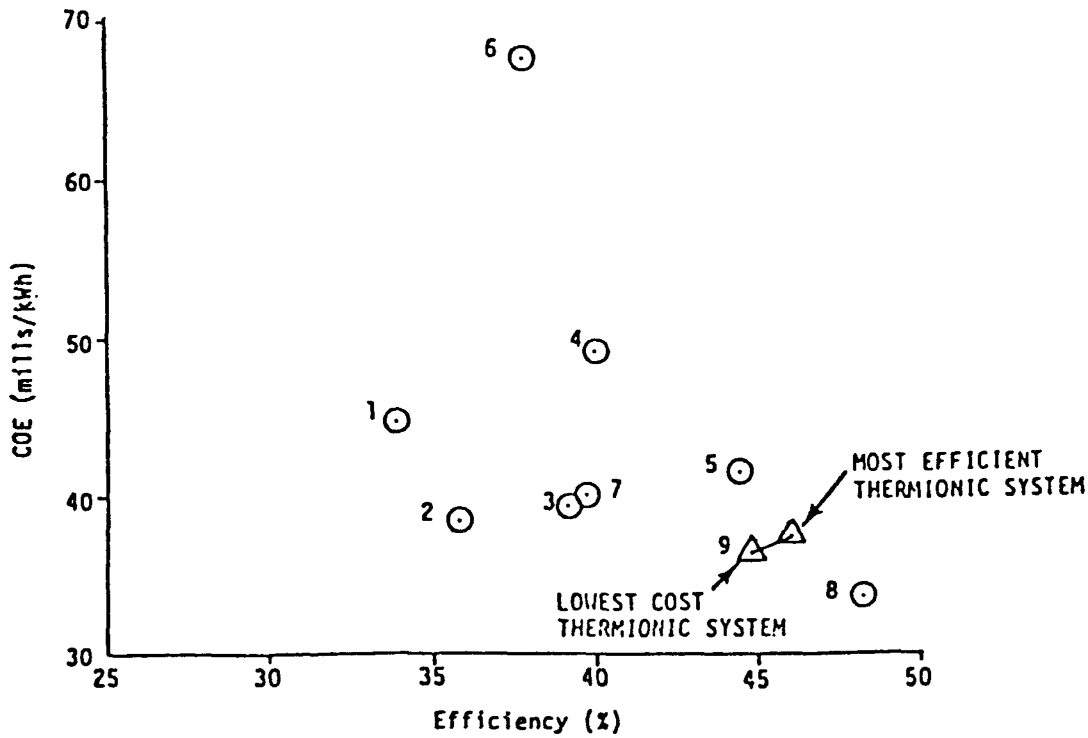


Exhibit D-3.10

Comparison of Thermionics with Other Power Plant Concepts in Terms of Capital Cost and Efficiency



1 CONVENTIONAL STEAM
2 ADVANCED STEAM, AFB
3 ADVANCED STEAM, PFB
4 CLOSED GAS TURBINE, ORGANIC BOTTOMING, AFB
5 POTASSIUM TOPPING CYCLE, PFB
6 OPEN CYCLE TURBINE, WATER-COOLED, SEMI-CLEAN LIQUID FUEL
7 OPEN CYCLE TURBINE, AIR-COOLED, LBTU GASIFIER
8 OPEN CYCLE MHD
9 INTEGRATED THERMIONIC-STEAM

Exhibit D-3.11

Comparison of Thermionics with Other Power Plant Concepts in Terms of Cost of Electricity and Efficiency

3.2 Retrofit Thermionic Heat Exchanger Coal-Fired Burner

The feasibility of retrofitting existing power plants with presently available converters was also considered in a preliminary study. The power plant selected for this study was a pulverized coal-burning system [8]. Only physical and energy balance constraints were considered. No detailed cost estimates were made.

In the late 1960's when it became increasingly obvious that lower emission of nitrogen oxides from power plants and other large industrial furnaces would be necessary, furnace manufacturers created low NO_x-controlled flow burners to meet this need. A ground rule in this study was that the retrofitted thermionic burners exhibit comparable features to the normal combustor. The new burners must be retrofitted without altering the overall system design. The keys to a retrofitted design are, therefore, the furance wall, the low NO_x coal burner nozzle, and the surrounding wind box assembly.

A typical two-stage coal burner is shown in Exhibit D-3.12. It fits between the furance wall containing the steam pipes (water wall) and the outer wall. This two-stage burner concept reduces NO_x emissions to acceptable levels. Burner assemblies are typically two feet in diameter. Work space behind the furnace outer wall was taken at a maximum of 15 feet. Hence, the sizing of each thermionic burner is limited by these dimensions.

A thermionic burner design concept which meets these needs is shown schematically in Exhibit D-3.13. The coal nozzle would be removed completely and replaced by the cogeneration thermionic burner. The change in the overall

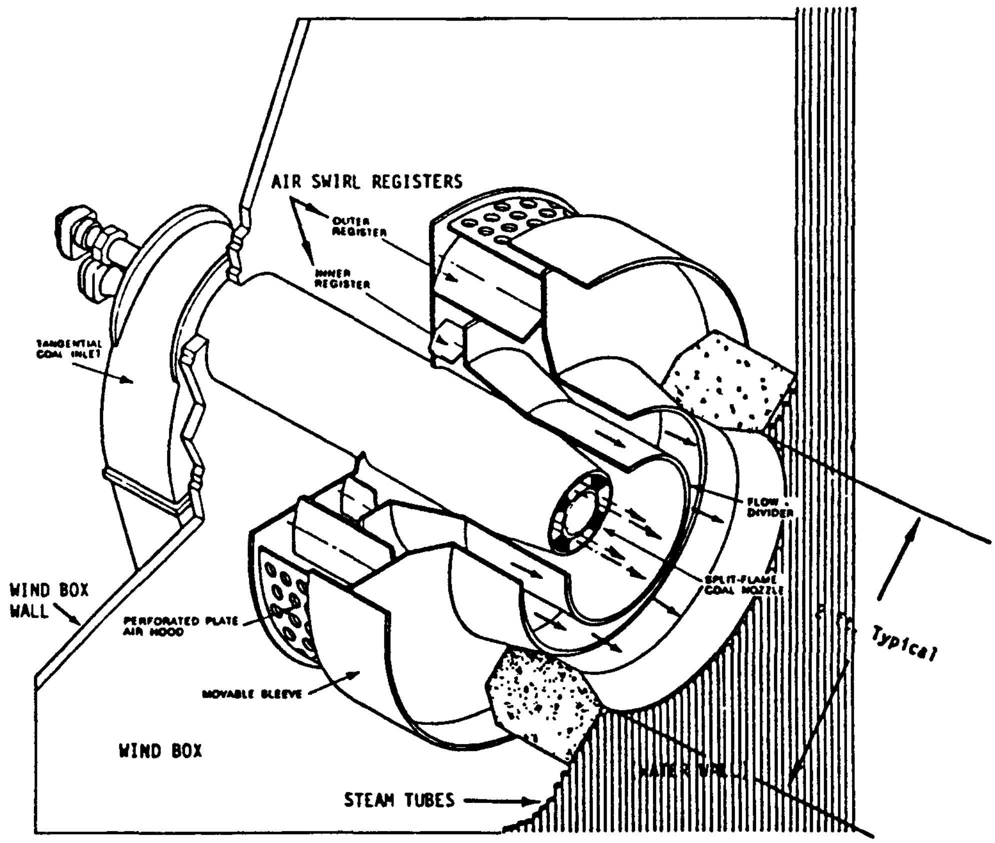


Exhibit D-3.12

Controlled Flow Low NO_x Coal Burner with Split Frame Flame Features and Water Wall Steam Tubes Deformed to Provide Opening

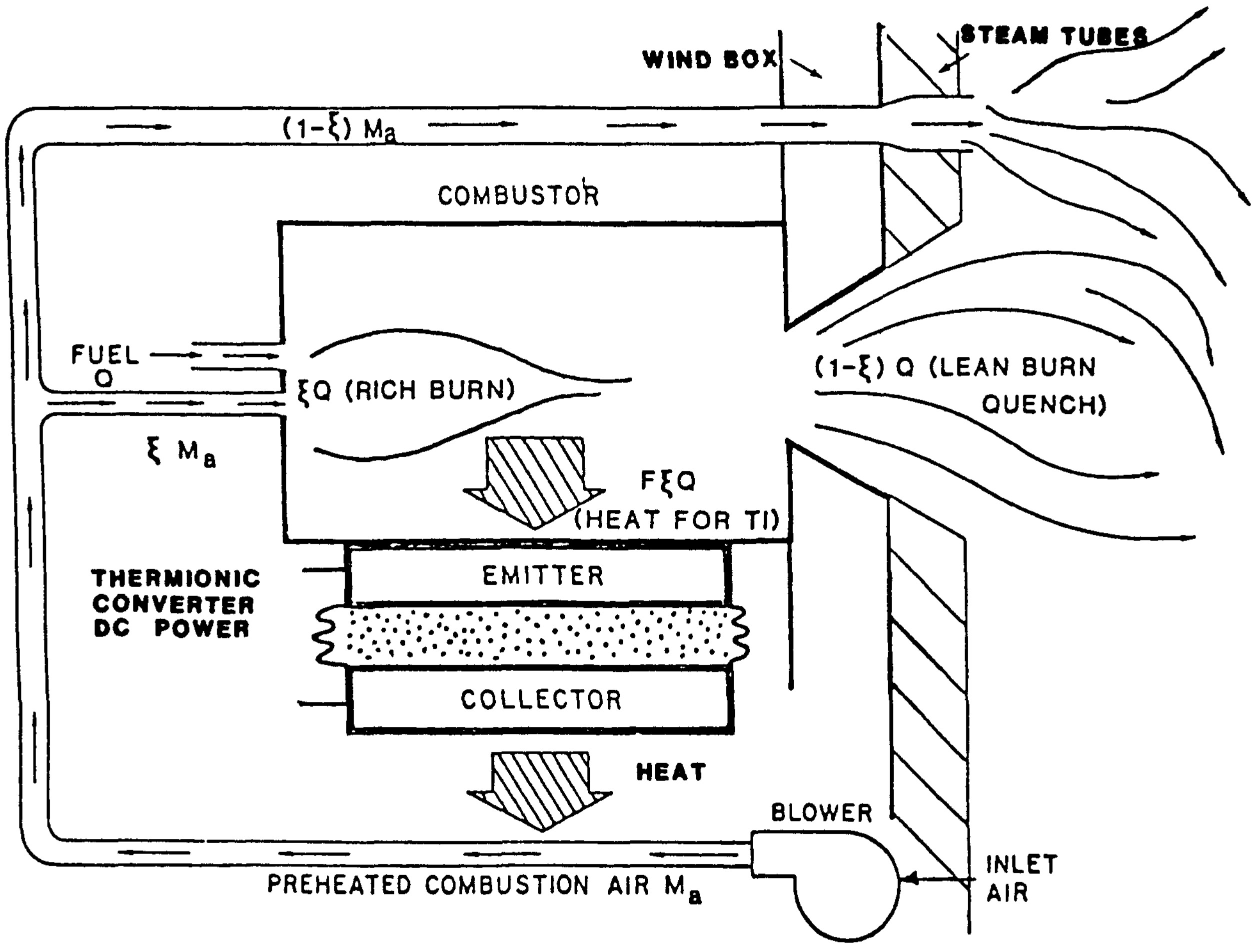


Exhibit D-3.13

Staged Combustion Retrofit Thermionic Burner for Coal Fired Central Station Boiler.

furnace operation is as follows. The rich burn takes place in the thermionic burner outside the main furnace. This burner then discharges hot combustion gases into the furnace where a lean burn takes place as before. The converter removes energy from the hot gases and cools them in the process. A gas blower passes air over the collectors to cool them, simultaneously preheating the air prior to combustion. Only a portion of this air passes through the thermionic rich-burn combustion region. The remainder is bypassed into the furnace.

The combustor was designed to use present generation thermionic diodes. The DC output power could be used on-site or conditioned by DC/AC power inverters to obtain 60 cycle AC.

The retrofit assembly is shown in Exhibits D-3.14 and D-3.15. This assembly acts as a plenum, off-board combustion chamber, and electric generator. It mounts onto the existing, typically two-foot diameter, nozzle mount on the water wall and, additionally, onto the wind box wall. Heater air is drawn into the inlet plenum from the wind box by two auxiliary air pumps. From within the inlet plenum, air enters the combustor cans via ports located on the periphery of the cans (see Exhibit D-3.16). Once in the combustor can, gas flows past the collectors, cooling them. The heated gas now enters the top of the combustion can by a duct located at the top of the can to be mixed with pulverized coal injected into the combustion chamber. Combustion takes place within the interior of the combustion chamber and flows downward through the can into the discharge plenum. The combustion gases heat the emitters of the converters. The converters generate the DC power output of the power plant. These flame-heated thermionic converters are shown in Exhibit D-3.17.

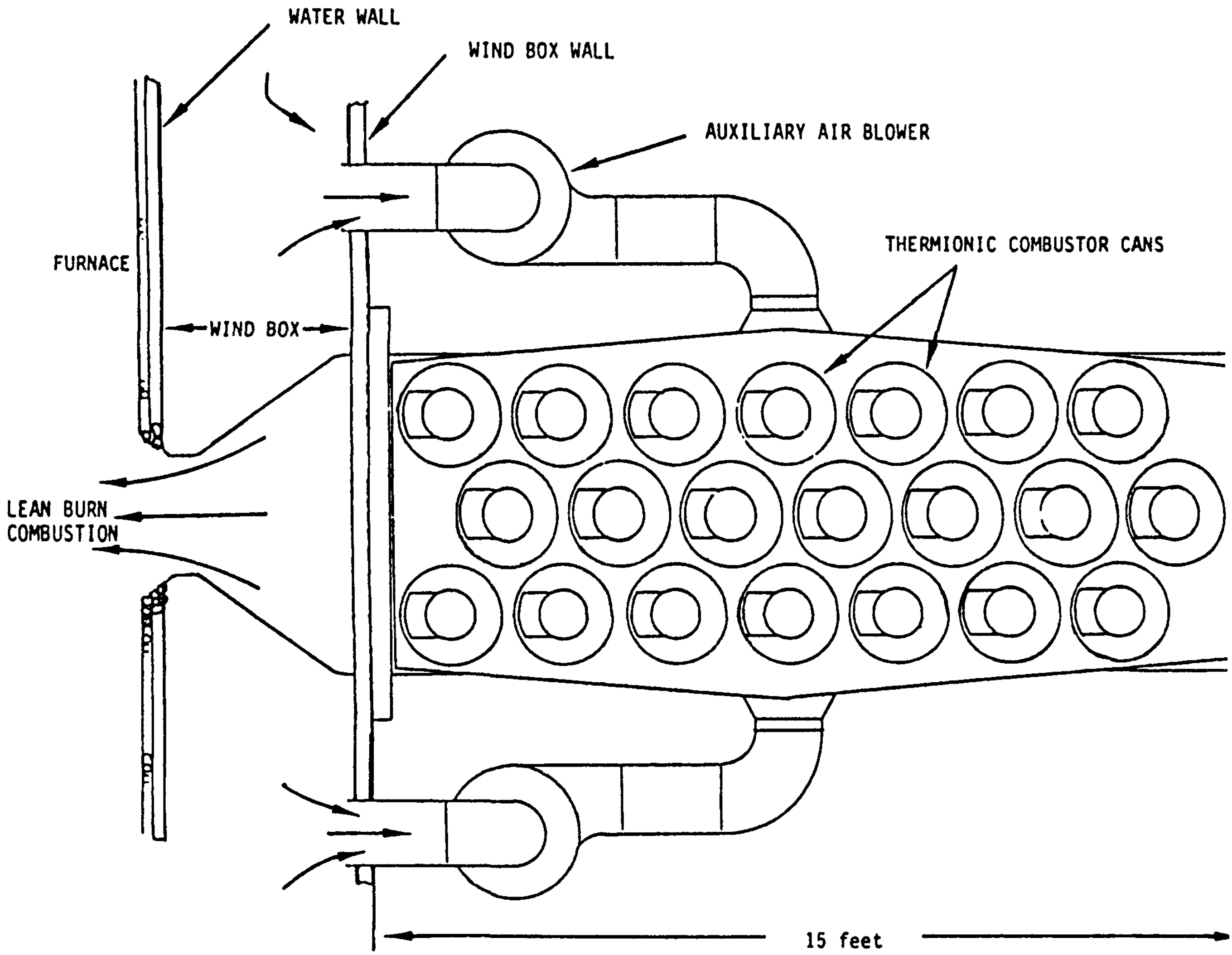


Exhibit D-3.14

Topview of Retofit Thermionic Combustor Showing 21 Combustor Cans

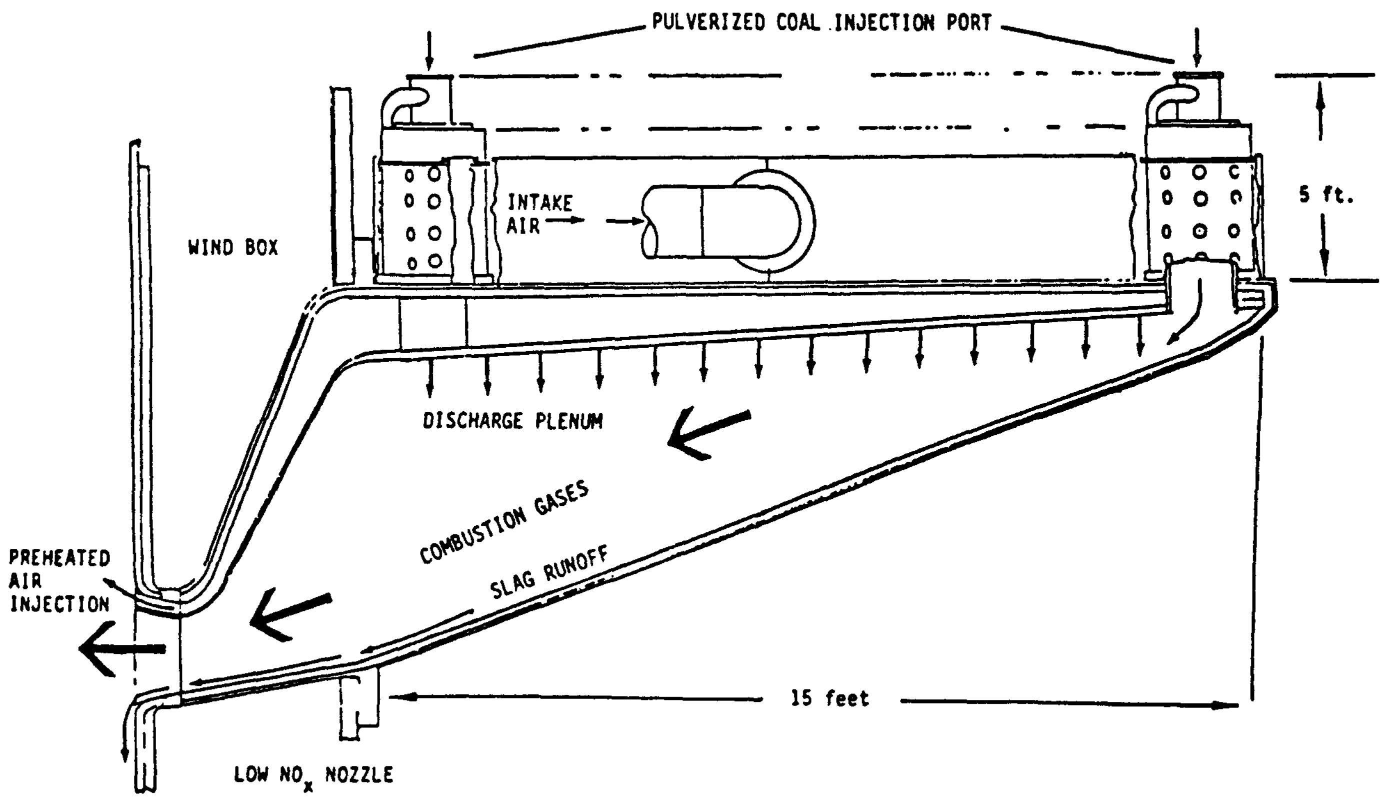


Exhibit D-3.15

Sideview of Retrofit Thermionic Combustor Showing a Cutaway View of the Plenum and Thermionic Combustor

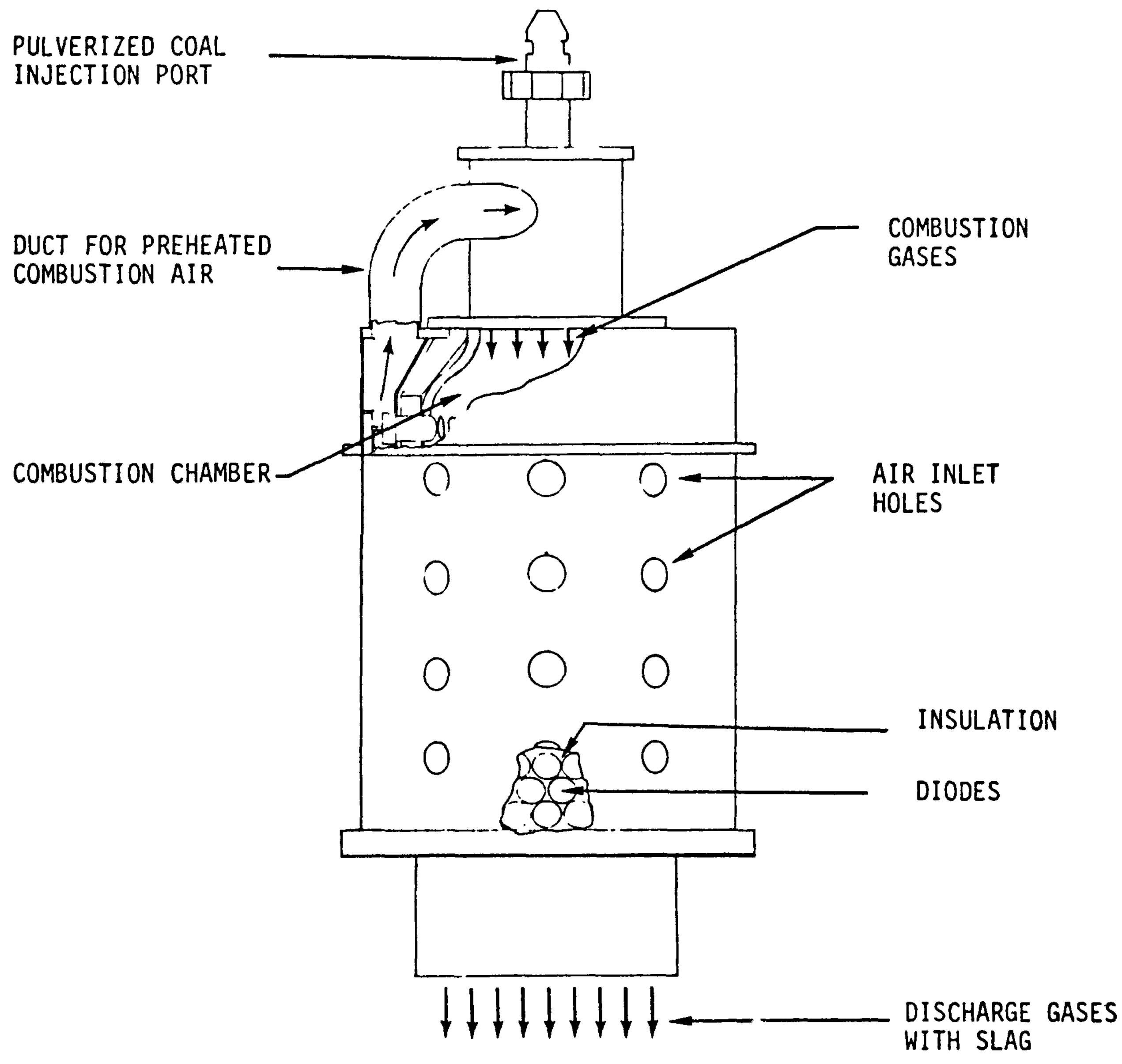


Exhibit D-3.16

Combustion Can Containing 384 Flame Heated Thermionic Converters

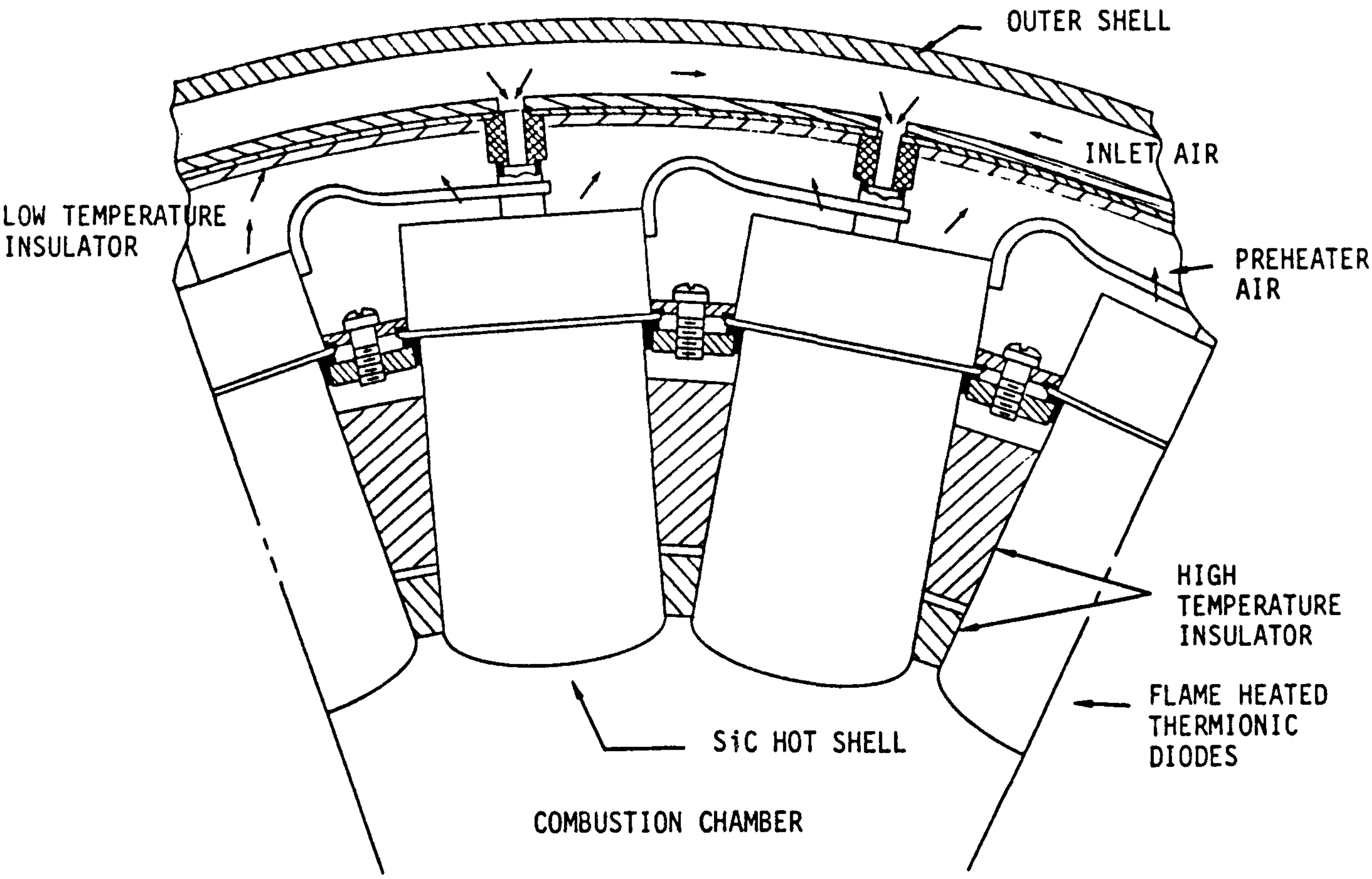


Exhibit D-3.17

Cutaway of Combustion Can Assembly Showing Flame Heated Thermionic Converters

Having passed the converter, the combustion gases exhaust into the discharge plenum. This plenum collects both the gases and the slag. The discharge plenum is sloped to permit slag to flow out of the plenum with the combustion gases. The combustion gas flow-rate is high enough to guarantee that the low NO_x nozzle remains open. The rich-burn gases along with the bypass gas provides a lean burn within the large furnace.

The retrofit thermionic converter system was designed for a 100 MBtu/hr pulverized coal furnace. The retrofit devices replace the basic burner. These replacement devices contain 21 combustor cans. Each can contains 384 thermionic flame-heated diodes whiich create 918 kW of thermionic power with a lead efficiency of 13.7%. Because 30% of the heat flows through the thermionic converters, the net conversion efficiency is 4.1%. The steam power plant alone generates 10.3 MW, assuming a 35% efficiency. Hence, the fractional increase in the output of the power plant with thermionic burners is 8.9%. This additional power is generated with an efficiency near 85%.

While this study has not examined costs, it is probable that the capital cost ($/kW) for the retrofit burners will be substantially below the cost of a new power plant.

3.3 Thermionic Array Module Coal-Fired Power Plant

Still another study of the use of thermionics to top a major coal-fired generating station was performed in which the thermionic design made use of a TAM combustor. In this "advanced boiler" concept, the largest possible number of thermionic converters are located in the hottest sections of the boiler and numerous small burners are used to obtain an essentially isothermal temperature profile in the combustion region. Both direct-coupled and air-coupled converters are used in the design, and the performance of a fully developed, or so-called "third generation," converter is assumed for the study.

The base steam system in the design is the commonly used 2400 psi/810 K/810 K (i.e., employing the conventional nomenclature for steam pressure/superheat temperature/reheat temperature). The steam turbine plant components are sized for 590 MW, as listed in Exhibit D-3.18, while the fuel- and draft-related equipment is sized for 680 MW, reflecting the additional fuel requirements of the converters. This power plant will generate a gross electrical output of about 850 MW. About 260 MW will be drawn directly from the thermionic converters, and 590 MW will be generated from the steam-turbine power cycle. About 50 MW will be required for plant auxiliary electrical loads so that the net plant output will be 800 MW. The reference fuel envisioned is pulverized Illinois No. 6 Coal.

In determining the station heat rate the auxiliary station loads were estimated to be nine percent of the turbine output based on typical current large pulverized coal units. The power-conditioning system efficiency used for the thermionic generation was 92% (i.e., a 4% loss in DC-to-AC inversion and no greater than a 4% loss in the external bus).

Since thermionic converters would be installed in boiler furnaces, it was considered important to maintain a thermionic converter configuration that would have minimum impact upon conventional steam boiler designs. The walls of most large, fossil-fired steam generators consist of tubes with an

EXHIBIT D-3.18

DESIGN SUMMARY FOR PULVERIZED
COAL-FIRED THERMIONIC TOPPED POWER PLANT

Coal: Illinois No. 6 - Heating Value 26 MJ/kg (11,132 Btu/lb)

Turbine Power Output	590 MW
Thermionic Power Output	260 MW
Net Station Heat Rate	8,300 Btu/kWh
Overall Boiler Dimensions	9.1 m x 44 m x 49 m high
Six Compartments Each	7 m x 9.1 m
Heat Release	50,000 Btu/ft^3

outside diameter generally in the range of 50 to 76 mm (i.e., 2 to 3 inches). These tubes are attached longitudinally with a membrane, about 6 mm (¼ inch) thick by 13 mm (½ inch) wide, making a gas-tight seal.

To simplify the attachment of converters, a boiler tube with a square outside and a round inside is used. The flats on the extrusion allow the attachment of the converters. On the boiler waterwalls, where only one side of each tube is exposed to the flame, the converters can be mounted on one side only. In locations where the tubes are exposed on two sides, converters can be attached on both of the exposed sides.

The advanced boiler is divided into separate compartments in order to increase the surface area for converters exposed to high temperatures. A number of small burners, with a heat release of 50 million Btu/hr, are placed along about half the height of each compartment. The spacing between division walls is 7.3 m. Air-coupled converters surround the burners, thus eliminating the need for running water-cooled tubes around the burners. The rest of the side walls consist of water-cooled tubes with converters attached. The incoming air for combustion is heated by a conventional air preheater to 590 K then by the converter preheater to 700 K. It is believed that a relatively constant combustion zone temperature of 2030 K can be achieved with these conditions. The emitter temperatures of converters vary from 1870 K to 1670 K.

In the thermionic air preheater, the converter assemblies are mounted panels, on 99 mm (3.9-inch) triangular pitch, which results in a density of about 120 converters per square meter (11 converters/ft^2) of wall surface area.

Each burner wall panel has 7180 air-preheater finned-tube converters. The 12 burner panels provide a total of 86,170 finned tubes, with converters on the furnace side of the wall. With each finned tube providing .16 m^2 (1.7 ft^2) of heat transfer surface, a total of about 13,610 m^2 (146,500 ft^2) is available to heat the air.

The use of small burners, i.e., about one-third the size of conventional burners, helps to maintain an essentially isothermal temperature profile in the combustion region of the boiler. Above the burner region of the boiler, the gas temperature will decrease until the required heat has been added to the evaporator section of the boiler. At that point, 1370 K, the gas will be directed to the superheater/reheater section.

An additional constraint on a pulverized coal-fired boiler, as compared with those using clean fuels, is the presence of ash in the combustion products. This ash must be removed periodically from the combustion chamber. In conventional boilers this is done in one of two ways: either the combustion air temperature at the furnace exit is kept below the fusion temperature point of the ash (so-called "dry bottom" boilers), so that the ash falls to the bottom of the furnace where it collects as a powder, or; the ash collects at the bottom of the boiler in a molten condition ("wet bottom") and is removed in a molten condition. In a dry bottom operation, the combustion zone temperature must be kept below 1770 K. However, this low temperature would make thermionic topping an unattractive option.

The estimated cost of such an 800 MW thermionic-topped plant was developed and is shown in Exhibit D-3.19. These estimated costs are arranged by the

Exhibit D-3.19

ORDER OF MAGNITUDE OF PLANT COST ESTIMATE

Block		800-MW Plant (in thousand dollars) Conventional Plant Coal-Fired	800-MW Plant (in thousand dollars) Plant with Advanced Thermionic Boiler Coal-Fired
1	Land and Land Rights	By Client	By Client
2	Yard Work	$ 13,122	$ 10,205
3	Main Powerhouse	43,133	38,800
4	Administration Building	1,719	4,830
5	Miscellaneous Buildings		
6A	Boiler Equipment by Vendor	60,011	36,354
6B	TEC Panels, consisting of thermionic converters and boiler tubes		116,796
6C	M.G. Set, Switchgears, Bus Bars and Accessories		21,945
7A	Balance of Boiler Plant	27,735	21,008
7B	Ash Handling	7,608	6,424
8A	Coal Handling	13,020	11,021
8B	F.O. Equipment and Structures		
8C	Startup Oil (Light oil tank and equipment)	630	630
9	Stack (concrete w/liner)	4,201	3,690
10	Precipitator	11,140	8,365
11	Scrubber — SO_2	51,845	57,040
12	Turbine Generator by Vendor	37,120	27,178
13	Balance of Turbine Generator Plant	7,370	5,404
14	Circ. Water System		
A	Screenwell Structure	18,760	14,203
B	Condenser System		
C	Intake and Discharge System		
D	N.D. Cooling Tower		
15	Water Treatment	1,530	1,402
16	Waste Treatment	7,450	6,764
17	Accessory Electrical Equipment	15,280	11,740
18	Miscellaneous Powerplant Equipment	3,770	2,900
19	Main Transformers, Including Foundations and Accessories	1,800	3,100
20	Transmission Lines		
	Total Direct Accounts	327,244	409,799
	Distributable and Indirect Cost	$ 78,539	$ 98,352
	Total, Construction and Indirect Cost	405,783	508,151
	Allowance for Indeterminates	40,587	50,840
	Total, Estimate	446,370	558,991
	Escalation	86,330	106,281
	Total, Including Escalation	532,700	665,272
	Interest During Construction	81,700	101,928
	Total, Including Interest During Construction	$614,400	$767,200
	Cost/kW	$ 768	$ 959

Federal Power Commission block method of cost accounting and include interest at 10% and escalation at 6.5% during construction.

SECTION 3 - REFERENCES

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