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CONCEPTUAL ENGINEERING DESIGN STUDY OF
THERMIONIC TOPPING OF FOSSIL POWER PLANTS

FOR

THERMO ELECTRON CORPORATION
WALTHAM, MASSACHUSETTS

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I. INTRODUCTION AND SUMMARY

A. Background

The objectives of the program, of which this study is a part, are to examine the engineering aspects of topping a steam power station with thermionic energy converters (TECs). This study is a continuation of previous work which was directed chiefly at the study of thermodynamic options in the design of topping systems.

B. Purpose and Scope

Primary objectives of this study are to investigate alternative design concepts of thermal coupling of TECs to the steam cycle and the mechanical and electrical aspects of integrating TEC design into the steam power station. The specific tasks include:

1. Evaluate design concepts of TEC topping of solvent refined liquified coal-fired steam power plants, with main emphasis devoted to thermal, mechanical, and electrical design aspects.
2. Develop preliminary conceptual design of a modular TEC assembly.
3. Based on results of the above tasks, develop preliminary cost estimates of the design modification to a liquified coal-fired steam power plant with TEC topping.
4. Provide support to Thermo Electron Corporation in planning TEC hardware testing.

C. Summary of Results

1. A preferred design configuration of thermionic converter/boiler tube assembly was developed for thermal coupling of the TECs to the steam cycle. It consists of panels of flat thermionic converters bonded to waterwall and platen boiler tubes with square exterior and circular bore.
2. The feasibility and desirability of topping a conventional boiler with thermionics were evaluated. The improved plant thermal performance as compared with a conventional plant is:

	<u>Conventional Plant</u> <u>(without Thermionics)</u>	<u>Conventional Plant</u> <u>(with Thermionic Topping)</u>
Station Heat Rate (Btu/kw-hr)	9,123	8,274
Plant Thermal Efficiency (%)	37.4	41.2

This improvement in plant performance is limited by the fact that current design boilers constrain the location for installing the major portion of thermionic converters to relatively low temperature

regions of the boiler, thus preventing the advantage of greater thermionic converter efficiency. Therefore, a conceptual design of an advanced thermionic boiler was developed.

3. The design of the advanced thermionic boiler incorporates the following features:
 - a. Waterwalls are replaced by TEC panels.
 - b. Furnace is modular 8 ft x 30 ft cross-section with staged burners along its height so that maximum combustion temperature (typically 3,200 F) and short residence time (~ one second) are achieved, consistent with high thermionic device efficiency and low NO_x generation.
 - c. The thermionic converters are so arranged in series-parallel strings that their electrical output is compatible with d-c to a-c conversion equipment. Several fault protection features have been included in the conceptual design.
4. Thermal performance of an 800 MWe conventional power plant and an 800 MWe advanced thermionic topped power plant were compared.

TABLE I-1

COMPARISON OF POWER PLANTS WITH AND WITHOUT THERMIONICS

	<u>Conventional Plant (Without Thermionics)</u>	<u>Thermionic Topped Plant (With Advanced Design Boiler)</u>
Total Plant Power Generation	800 MWe	800 MWe
Steam Plant Power Generation	800 MWe	500 MWe
Thermionics Power Generation	-	300 MWe
Station Heat Rate (Btu/kwhr)	9,123	7,235
Thermal Efficiency (%)	37.4	47.2*

*Higher thermal efficiencies are attainable with additional thermionic converters but were not included in this design because they were considered not to be cost-effective.

5. Preliminary cost estimates indicate that a TEC topped liquified coal-fired plant (800 MWe total) using an advanced thermionic

boiler can be constructed for \$618 per KWe (1977 dollars), compared to a cost of \$503 per KWe for a conventional 800 MWe liquified coal-fired plant. This estimate was based on the FPC block method of cost accounting and includes interest (10 percent) and escalation (6.5 percent) during construction.

Using solvent refined liquified coal at a levelized cost (assuming a 30-year period) of \$4.92 per million Btu and a fixed charge rate of 18 percent, it was shown that the capitalized savings in fuel cost is \$217 million and that a total capital savings of \$156 per KWe could be achieved by using TEC topping.

6. An approach was developed for testing a thermionic converter/boiler tube test panel in an operating boiler.

II. THERMIONIC TOPPING OF CURRENT TECHNOLOGY BOILERS

A. Description of Fossil Plant Selected for Evaluation of Thermionic Topping

1. Basis for Selection

As has been stated in an earlier Stone & Webster Engineering Corporation (S&W) report (Reference 1), direct thermal coupling of thermionics to boiler waterwalls would yield attractive heat rate improvements over conventional steam cycles. Since the fuel specified to be used in this study is liquified coal, oil-fired type boilers were selected for evaluation of thermionic topping. In addition, a 2,400 psig/1,000 F/1,000 F* steam cycle was selected because it was shown in Reference 1 to have the best potential for thermionic topping. The plant selected for study has a net station output of 800 MWe.

2. Description of Conventional Oil-Fired Type Boiler

The oil-fired type boiler selected for conversion to a thermionic topped unit has a furnace width of 82 ft and a furnace depth of 44 ft. The furnace height not including the ash hopper is about 115 ft as shown in Figure II-1. All 32 burners are located on the front wall. Each burner has an input of 245 million Btu/hr for a total boiler input of 7.8 billion Btu/hr. The furnace wall area covered by the burners is approximately 68 ft wide by 34 ft high. The distance between the roof and the upper part of the arch at the furnace exit is 28 ft.

The furnace has seven steam-cooled platens located above the burners, extending from the front wall up through the roof. Except for some superheat pendant surface located above the arch, the remaining superheat and reheat surface is located in the back convection section of the boiler.

3. Description of Conventional Oil-Fired Type Boiler with Thermionic Topping

Figure II-1 shows the general configuration used in establishing the quantities and location of thermionic covered boiler tube surfaces. A relatively large number of platens covered on both sides with thermionics (configuration 6B in Figure II-2, Section IIC) have been located in the upper furnace. In addition, all furnace walls are covered with thermionics (configuration 6A) from the top of the ash hopper to the furnace roof, except for the front wall in the burner area, the furnace exit on the screen tubes, and the furnace roof. None of the superheat and reheat surfaces are covered with thermionics due to the low heat flux, the relatively high sink temperature, and resultant low thermionic efficiency. Thermionics were not installed on the front wall in the burner area because of the difficulty in applying thermionic devices to odd-shaped wall tubes, nor were they installed on the screen tube area, because of the relatively low thermal flux at this location, nor the roof area,

*Conventional nomenclature is used in this report: steam pressure/
superheat temperature/reheat temperature.

FIGURE II-1
OIL-FIRED BOILER

because of the many penetrations through the roof by the platens and superheater tubes. The platens were located in the upper furnace to avoid flame impingement from the high heat input burners.

B. Comparison of Thermal Performance of Plants (With and Without Thermionics)

The boilers described in Sections I-A.2 and I-A.3 (conventional boiler and the same boiler modified with thermionic topping in the evaporator section) have been compared (see detailed calculations in Appendixes D and E). Both boilers burn liquified coal and supply a 2,400 psig/1,000 F/1,000 F steam cycle with a turbine heat rate of 7,829 Btu/kw-hr. The thermionics are coupled to the watercooled surfaces of the boiler to provide the lowest practical collector temperature.

In determining the station heat rate, the auxiliary station loads, exclusive of the force draft fans, were estimated to be 4.5 percent of the turbine output based on typical current large oil-fired units. The forced draft fan power was calculated separately and included in the overall evaluation. The conversion efficiency used for the thermionic generation was 92 percent (4 percent loss in d-c to a-c inversion and no greater than 4 percent loss in the external bus duct). The basis for these values is discussed in Sections II-D and III-D.

Since the addition of thermionic topping to a conventional boiler was originally envisioned as a retrofit, the steam plant size was held constant (800 MWe) and additional surface area was added to the boiler to compensate for the significant reduction in heat flux to the evaporator section of the boiler caused by the addition of thermionic converters. Current boiler design required that this additional surface area be added via platens which are located in a relatively low temperature region of the boiler. The resulting location of the major portion of the thermionic converters in this low temperature region of the boiler prevented the maximum cycle efficiency improvement from being realized and led to the development of the advanced boiler concept discussed in Section III of this report.

Since the steam plant size was held constant, the thermionic topped cycle has a greater net output than the conventional boiler. It does, however, require greater fuel and combustion air input even though its efficiency is higher. The plant cycle comparison is summarized below:

TABLE II-1

COMPARISON OF PLANT CYCLE WITH AND WITHOUT THERMIONICS

	Conventional Plant (without Thermionics)	Conventional Plant (with Thermionic Topping)
Station Heat Rate (Btu/kw-hr)	9,123	8,274
Efficiency (%)	37.4	41.2
Total Water Cooled Surface with Thermionics (%*)	-	218
Total Generation with Thermionics (%*)	-	114
Total Heat Input with Thermionics (Fuel Consumption) (%*)	-	103

*As a percent of that with no thermionic topping.

C. Selection of Boiler Tube/Thermionic Converter Configuration

1. Basic Configuration Requirements

Since thermionic converters would be installed in boiler furnaces, it was considered important to maintain a thermionic converter configuration which would have minimum impact upon conventional boiler designs. The walls of most large, fossil-fired steam generators consist of tubes with an outside diameter generally in the range of 2 to 3 in. These tubes may be attached longitudinally with a membrane, about 1/4 in. thick by 1/2 in. wide, making a gas-tight seal. Alternatively the tubes are positioned within 1/16 in. of each other and a continuous weld is applied to the outside between tubes, again producing a gas-tight seal. A third configuration consists of a longitudinal fin extruded with the tube and welded to the adjacent tube. In all cases, the outside wall tubes are arranged to produce a leak-free box or furnace in which to burn the fuel.

Within the furnace confines, water-cooled tubes may be installed in the form of panels or platens. These platens generally consist of tubes tangent to one another and supported from headers at the top. In platens the tubes are generally not attached to each other as they are in the case of the furnace walls because a gas-tight seal is not required. The plane of the platens normally runs from front to back of the furnace, rather than from side to side.

2. Description of Alternatives Considered

Alternate design configurations shown in Figure II-2 were evaluated. The above basic design criteria have been retained in all arrangements shown.

Configuration concepts 1A, 1B, and 1C were developed to provide a flat surface on which to mount the thermionics. Configuration 1B would enable a larger thermionic converter to be used and possibly reduce fabrication costs. In configuration 1C liquid sodium was used to improve heat transfer and provide a more nearly uniform temperature across the collector.

The elliptical boiler tube configurations (2A, 2B, and 2C) were developed to provide a substantially flat surface for the thermionics with a more symmetric tube shape. Elliptical shapes 2B and 2C provide more extensive thermionic surface at the expense of a partially rounded thermionic device.

Pipe strap configurations (3A, 3B, and 3C) were considered for application to conventional boiler tube shapes. All three are similar, except for the thermionics. Configurations 3A and 3B would require membranes between tubes, but 3C would only require a continuous longitudinal weld.

Arrangements 4A, 4B, and 4C would apply primarily to platens in which tubes are tangent but not attached along their lengths. These configurations require a heat input completely around the tube and are thus not suitable for waterwall application.

FIGURE II- 2

CONFIGURATION CONCEPTS FOR THERMAL COUPLING
OF BOILER TUBES AND THERMIONIC CONVERTERS

Configurations 5A, 5B, and 5C were developed for application to existing units or to conventional waterwall designs. In case 5A an extrusion would be mounted between tubes, 5B is similar except that the extrusion cavity is filled with liquid metal, and 5C utilizes liquid metal to provide intimate thermal contact with the tube.

The last tube cross-sections 6A and 6B resulted from a review of the previous arrangements. The first would apply to the furnace walls and the second to the platens.

3. Evaluation of Configuration

Configurations 1B, 1C, 2A, 2B, and 2C were eliminated because the boiler tube would present extrusion difficulties. Semicircular thermionics are required for 3A, 3B, and 3C. They are difficult to manufacture and, therefore, these shapes were rejected. 6B was selected over 4A and 4B because 6B, which is narrower, must absorb the same amount of heat through a smaller cross-section, thus increasing the heat flux and increasing thermionics output.

Cross-section 5A was rejected because of potentially high thermal contact resistance due to poor fit between the extrusion and the tubes, and Configurations 5B and 5C were dropped from consideration because of the need for liquid metal, which is potentially hazardous in the presence of high temperatures.

Tube Section 6A is an outgrowth of 1A. In evaluating potential extrusions, it was discovered that 1A is difficult or perhaps impossible to manufacture because the distribution of metal is not symmetric about the tube centerline.

4. Conclusions

As a result of the above analysis, 6A and 6B have been selected as the best configuration of those presented. Configuration 6A would be used on outside furnace walls and 6B in platens. The 6A tubes would be mounted side by side and seal welded, thus eliminating the need for a membrane. This shape of boiler tube can be readily extruded by using a special die with conventional tube extrusion machinery.

D. Electrical Concepts

1. Thermionic Device Configurations

The basic thermionic device configuration was selected as 6A or 6B in the preceding section. This section discusses the rationale used to set the size (surface area) of the individual converter and the number of converters that would be connected in parallel and series to form a string. The following electrical design factors were considered in the evaluation.

- a. For a given power output, an increase in d-c voltage will decrease the power losses in connections and bus work and will reduce the cost of d-c to a-c conversion equipment. Since each thermionic converter develops a potential difference of the order of 1 V, it is therefore desirable to connect a large number of devices in series.
- b. The maximum permissible voltage is constrained by the dielectric strength of the creepage path across the material used between the device collectors and the grounded boiler tubes to which they are attached. The thickness of this material must be minimized to facilitate heat transfer.
- c. The current density of each converter (amperes per square centimeter of emitter surface) is a function of the difference in temperature between its emitter and its collector. Where converters at different temperatures are connected in the same series string, their individual surface areas must be varied along the temperature gradient to produce the common current which will flow through all of the converter.
- d. It is desirable that the two terminals of each series string be close together at the same elevation to reduce the length of bus work required to connect the string to conversion equipment.

Thermo Electron Corporation has recommended that the voltage constraint, Item (b) above, be 200 V under conditions expected in the boiler. To achieve the objective outlined in Item (a) above, it is proposed that the electrical mid-point of each series string be grounded through a resistor. The string voltage can then be as high as 400 V without violating the constraint.

Thermo Electron Corporation also has indicated that there is an upper limit to the size (surface area) of converters that can be manufactured at reasonable cost and maintained, while at high temperature, within required dimensional tolerances. For this reason, a maximum converter length, parallel to boiler tube, of 20 cm was adopted. In lower temperature zones of the boiler, where greater lengths would be required to meet the criterion described in Item (c) above, it is proposed that two or more shorter converters, adjacent along the tube, be connected in parallel to obtain the required surface area.

Following this rationale, strings of converters were configured to cover the entire area available, with each string generating a current of approximately 1,200 amp at a voltage as near 400 V as possible. For "current technology" boilers, the resultant configurations are as shown in Figures II-3 and II-4.

FIGURE II-3

ILLUSTRATION FOR A "STRING" TRAVERSING
THE WATERWALL FOUR TIMES, IN THE FIREBOX REGION

FIGURE II- 4
ILLUSTRATION OF A "STRING" TRAVERSING THE
WATERWALL SIX TIMES, IN THE PLATEN REGION

2. Boiler Zones Selected for Application of Thermionics

The electrical performance of thermionics as applied to current technology power plants was analyzed using two "averaged" sets of operating parameters, corresponding to a two zone model of the current technology boiler. It is believed that by investigating performance on the basis of the two boiler zones:

- a. The variation in boiler conditions experienced by thermionic devices would be accounted for; and
- b. The differences between the planar geometry in the platen regions and the planar geometries of the waterwall and firebox regions would be adequately formulated.

Thus the "firebox region" and the "platen region" were identified as separate operating zones. By matching the thermal conditions (heat flux available) for each zone, with the family of characteristic curves for thermionic converters, the following operating points for each zone were derived. (The detailed calculations are shown in Appendixes A and E.) The thermionic converter characteristics (e.g., efficiency, output voltage, etc.) were extracted from Reference 3. These characteristics are for third generation converters as described in Figure 1-4 of Reference 4.

TABLE II-2

CONVERTER PARAMETERS IN EACH ZONE OF CONVENTIONAL BOILER

<u>Converter Parameter</u>	<u>Zone 1 Firebox</u>	<u>Zone 2 Platen Region</u>
T_E ($^{\circ}\text{K}$)	1,500	1,460
T_C ($^{\circ}\text{K}$)	670	660
P_{in} (W/cm^2)	13.5	10.5
P_{out} (W/cm^2)	4.4	3.3
V_{out} (volt)	1.1	1.1
J (amp/cm^2)	4.0	3.0

3. Series and Parallel Connection of Thermionic Converters

Parallel connections between adjacent devices do not have to satisfy conflicting requirements of electrical and thermal conductivity because the electrodes being interconnected will be at essentially the same temperature. In addition, parallel connections between converters on adjacent tubes are "equalizer" connections and will carry little current

under normal operating conditions. Where adjacent converters along a single tube are connected in parallel, the connectors carry the full current of the string.

The series connectors must be optimized, accepting a reasonable voltage drop with corresponding loss of electric power in order to reduce thermal conduction from a hot emitter to a cooler collector. In the optimization, it should be recognized that the electric power loss in the series connectors, though it subtracts from the gross d-c output of the thermionic converters, is not a loss to the combined cycle. That heat becomes part of the total used for evaporation.

The composite connector, which parallels converters on adjacent tubes and connects them in series with a similar pair, adjacent along the tubes, is essentially a double-width series connector and must be optimized as such.

Output voltages shown in Table II-2 are net values, allowing for the voltage drops in optimized series and composite connectors.

Proposed uses of the three types of connectors in building up strings having the desired current and voltage output are shown in Figures II-5 and Figure II-6.

A device length was initially selected which provides a resultant string current of approximately 1,200 amp. This converter length was increased by 5 percent and divided into the physically determined string length. The resultant figure represented the number of units per string. Its multiplication by $(V_{out} \text{ less } 60 \text{ mV})$ would represent the string output voltage. A minimum potential of 300 V (terminal to terminal) was sought. If the string potential value was acceptable, then the converter configuration was accepted; otherwise, an iterative procedure was followed, adjusting the converter length, bringing about decreases in current for increases in string voltage. Other options would be to increase string length - however, such an increase would be restricted to fixed increments to maintain the preferred location of termination points - or increase the number of devices in parallel (this would be constrained by current magnitude considerations and the decision to cover only two tubes per string segment).

4. Electrical Performance of Interconnected Thermionic Converters.
(Calculations for assessing the electrical performance of strings in each of the zones are found in Appendix I.)

The firebox zone is estimated to be 17.5 m (57 ft) high for the conventional 800 MWe boiler. If each string traversed the wall four times, then a string would cover eight boiler tubes and be 70 m in total length. This string, based on the parameters of the firebox zone discussed earlier, would have the following performance characteristics. Power output would be 418 kW, voltage output 346 V d-c. This zone would accommodate 85 strings, thus having a capacity to generate approximately 35.5 MWe.

FIGURE II- 5
CONCEPTUAL ILLUSTRATION OF SERIES - PARALLEL CONNECTOR
IN STRING ASSEMBLY (TWO CONVERTERS IN PARALLEL)

FIGURE II-6
CONCEPTUAL ILLUSTRATION OF PARALLEL AND SERIES -
PARALLEL CONNECTORS IN A STRING ASSEMBLY
 (FOUR CONVERTERS IN PARALLEL)

The waterwall zone in the platen region of the boiler above the firebox would be approximately 16 m in height. Each string is designed to traverse these waterwalls six times, and 12 tubes would be covered by a string with a total string length of 96 m. The "per string" performance parameters of this zone are voltage output of 357 V, power 49 strings; thus a total generating capacity for the waterwall of 21 MWe would exist.

For the platens, two of the four sides of individual tubes will have mounted thermionics. The average length of a platen tube is estimated to be 15 m, and a single string of thermionic converters will traverse a platen surface six times. With such a configuration, the output voltage of a string is 346 V and the power, per string, 415 kW (assuming each device is 7.5 x 13.3 cm). Utilizing the surface requirements determined in calculation Appendix E, there would be 21 platens and each platen would consist of 60 tubes (based on the estimated average length of 15 m). There would be 10 strings per platen (there are two surfaces available per tube), thus 4.15 MW per platen. The total power generating capability of these converters on the platens would be 87 MWe.

The power available at the terminals of all strings, based on electrical performance calculations above and including the connection losses, would be 143.5 MWe.* The design of the interconnecting bus work between the boiler and conversion equipment is based on copper bar sized to carry 800 amp/in.² The losses for interconnecting the string outputs with the d-c to a-c conversion equipment should not exceed 4 percent, thus 137 MWe would be available at the converter input terminals. D-c a-c conversion efficiency of 96 percent (see section below) would result in 131 MWe available at the generator output.

5. Conversion of Direct Current to Alternating Current

Two general methods of converting d-c to a-c have been investigated. The first of these employs solid state devices, silicon diodes, thyristors, etc., in inverters. The second method implements a special form of d-c motor, the acyclic motor, to drive a conventional a-c generator. Both methods are viable and in both cases, the technology is well developed. The choice must be made on the basis of cost, efficiency, and reliability.

In this application, Thermo Electron Corporation estimates that maximum acceptable d-c voltage to ground may be of the order of 200 V. This limit is set by the electrical insulation between the thermionic collector and the boiler tube. By grounding the electrical midpoint of a string, it is then possible to design for an output of 400 V. This value is important in the comparison of alternatives, since the equipment costs for either solid state inverters or acyclic motors are sensitive to voltage level, being high at lower voltages.

*The difference between this figure and the results of calculation Appendix E reflect the inclusion of required spacing between devices and connection losses within the string.

Best estimates for "mature product" prices of acyclic motors, with accessories, at this voltage are of the order of \$22 per kW for 1,800 rpm large machines. To this must be added the price of conventional a-c generators with excitation and cooling accessories, estimated at \$35 per kW. Each acyclic motor would have an associated accelerating motor estimated to be approximately 100 hp. The attendant control equipment (load centers and starters) has not been included in the estimate. The overall efficiency of the motor-generator set would be approximately 96 percent.

Solid state inverters at this voltage level would be built in small modules of, perhaps, 400 kW each for a cost of approximately \$156 per kW and would have an efficiency of the order of 92 percent. While the relatively small size of the modules is advantageous in minimizing the length of d-c bus required (which also minimizes electrical losses in transmission), the large number of modules suggests higher maintenance cost.

Addressing reliability, no comprehensive comparative data are available for either alternative. The acyclic motor is a relatively simple machine, but there are very few of them in service. A-c generators of the size contemplated have a long record of consistent reliability. Low voltage inverters are still experiencing the kinds of troubles that are characteristic of the "learning curve" period in product development.

Despite the absence of hard data to use in the comparison, it seems apparent that the cost and efficiency figures indicate a significant advantage for the acyclic motor - a-c generator alternative in this application.

E. New Boiler vs Retrofit Applications

It is apparent from the discussion in Section II-B that there is a significant change in furnace waterwall surface in boilers with thermionic topping as compared to those without. Therefore, the application of thermionics to an existing unit would seem at first glance to be very difficult, if not completely impossible. While this might be true if the retrofit of a complete thermionic topping cycle is considered, there are operating boilers which cannot meet performance guarantees because of evaporator/superheater-reheater surface area mismatches. One problem that this mismatch causes is that of low superheat and reheat temperatures. If the low steam temperature is due to excessive evaporator surface resulting in low combustion gas temperatures at the furnace exit, the proper application of thermionics to furnace walls could both improve the performance of the boiler and steam cycle and add power generation that could not be obtained in any other manner.

On these units an increase of 25 F (975 F vs 1,000 F) in superheat and reheat steam temperatures results in an increase of approximately 0.8 percent in station heat rate.

For an 800 MWe unit with 60 percent availability, at 100 percent load, burning oil that costs \$2 per million Btu, a 0.8 percent improvement in station heat rate could mean over \$650,000/yr in fuel savings alone. This would apply to a unit with a cycle efficiency of 10,000 Btu/kWh.

It would depend on boiler construction, but it is also possible that thermionics can be added to existing boilers without design deficiencies. In these cases, water-cooled as well as superheat and reheat surfaces would be altered as necessary. The extent of such a modification would depend on each boiler and thus would have to be evaluated on a case-by-case basis.

F. Materials and Fabrication Considerations

1. Boiler Tube Configuration

Babcock and Wilcox Tubular Products Division has indicated that a boiler tube of rectangular cross-section is feasible for standard production. The basis for the feasibility study was the standard 2 1/2 in. OD tube modified to include a 2.95 in. wide flat as the surface intended for the thermionic device attachment. (See dimensional sketch on page 2-8 of Appendix G.) This rectangular tube design was selected for the waterwall and platen application. The available cost estimate is \$.95/lb for quantities of 10,000 lb or more.

The rectangular boiler tube design is advantageous for the thermionic converter application, but it requires additional material and a more complicated stress analysis. Extruded material mechanical properties should be similar to the round tube extrusion. Minimum mechanical properties per ASTM A-209 or A-210 would be specified for the square extrusion.

2. Electrical Insulation/Thermionic Tube Bond Material

One requirement is to electrically insulate the thermionic converter from the boiler tube. However, the insulating material must not reduce heat transfer to the boiler tube to the extent that excessively high collector temperatures are experienced. Consequently, the desired characteristics of the insulation material are good thermal conductivity and high electrical resistivity. A compromise must be reached since good thermal conductors (metals) usually display good electrical conductivity (low resistivity).

Preliminary candidate materials are magnesia, forsterite, boron nitride, alumina-oxide, alumina-columbium, and alumina-chromium. Physical property data are given in Table II-3.

Magnesia is an attractive candidate on the basis of its coefficient of thermal expansion and other experimental data. Recent experiments conducted at Thermo Electron Corporation showed that a 0.006 in. thick plasma arc sprayed MgO layer on low carbon steel displayed a breakdown

voltage in excess of 500 V at 550 c (1,022 F). Alumina-chromium is also an attractive candidate on the basis of thermal conductivity, electrical resistivity, and bonding experience. The thermal conductivity of the alumina-columbium cermet could be increased by increasing the Cb content.

Cost considerations show that the alumina ceramic would be the most economical choice, closely followed by either cermet. The cost of the boron nitride would be at least one magnitude greater than the other candidates.

Thermo Electron Corporation has successfully bonded a thermionic device to a metallic surface using a cermet of an alumina-columbium mixture. Integral bonds have been produced by autoclaving at 1,650 C under pressure. Magnesia has been successfully plasma-arc sprayed and thermal cycled to evaluate bond strength.

TABLE II-3

PHYSICAL PROPERTY DATA FOR ELECTRICAL INSULATION MATERIALS

<u>Material</u>	<u>Thermal Conductivity (Btu/hr/ft²/F/ft)</u>	<u>Volume Resistivity (microhm-cm²/cm)</u>	<u>Coefficient of Thermal Expansion (x10⁻⁶/F)</u>
Boron Nitride	16.6 (570 F)	2.3×10^{10} (900 F)	4.3 (60-1,800 F)
Alumina (99.5 per- cent Al ₂ O ₃)	8.5 (650 F)	$10^9 - 10^{10}$ (932 F)	3.7 (68-932 F)
Magnesia	9.5 (800 F)	2.1×10^{10} (932 F)	7.0 (70-1,300 F)
Fosterite	2.92 (572 F)	1.2×10^{10} (932 F)	6.2 (70-1,300 F)
Alumina/Columbium 70 Al ₂ O ₃ /30 Nb	15-20 (est-600 F)*	10^8 (est.)	4.0-4.5 (est.-68- 1,000 F)
Alumina/Chromium	29.0 (500 F)	10^8 (est.)	4.7 (32-1,832 F)

*This estimate is based on the assumption that the thermal conductivity of a composite is linearly proportional to the percentage of the constituents.

Although the alumina-columbium would appear to be a likely candidate at this point in time, it is recommended that other cermets and the magnesia be pursued. The alumina-chromium or alumina-chromium-molybdenum would be slightly more economical, and the effect of chromium and/or molybdenum additions to the boiler tube base material, via diffusion at the bond interface, has been more widely documented than that of columbium. Chromium will increase oxidation resistance. Molybdenum will increase bond strength and enhance elevated temperature properties. There is some question about the corrosion performance of the columbium addition

since columbium metal will oxidize rapidly above 750 F. There is no known mechanical or metallurgical prohibition that would disallow the use of the alumina-chromium cermet, and the increased thermal conductivity, over the alumina-columbium cermet, would be beneficial.

3. Thermionic Converter/Boiler Tube Fabrication Process

Fabrication methods presently available for attaching the thermionic converter to the boiler tube are brazing, autoclaving, and plasma-spray.

Brazing would be the process selected for fabrication if a solid ceramic (alumina) was chosen for the insulation/bond material. Brazing would be preceded by a metallizing operation to prepare the ceramic surface for bonding. Coors Porcelain Company has used a manganese-molybdenum braze extensively for high temperature applications.

Autoclaving would be the fabrication process if a cermet slurry was used as the insulation/bond material. Hot-pressing will be accomplished by the application of pressure at 1,650 F. Time-at-temperature will be substantial to allow the diffusion bond to form.

Plasma-spray would involve the deposition of a ceramic or cement to either bonding surface, followed by metallizing and/or brazing to achieve the necessary bond strength. Such bonds could be formed by brazing, autoclaving, swaging, or explosive forming.

Prior to final selection of the fabrication process, a detailed analysis should be conducted on the resulting bond strength and the effect of the fabrication process on the mechanical properties of the boiler tube material. Long time at temperatures in excess of 1,600 F may lead to a grain-coarsened structure resulting in unacceptable mechanical properties. A subsequent heat treatment may be desirable.

4. Typical Shop Fabricated Modules

This section describes waterwall tubes and panels in a conventional boiler and waterwall tubes, panels, and platens in a conventional boiler modified with thermionic topping.

a. Conventional Wall Tubes

Material -

Typically low carbon steel such as ASTM SA 210A-1, SA 210C, SA 213-T2, and SA 106 GR B.

Outside Diameter -

Typically circular tubes 2 in. to 3 1/4 in. in 1/4 in. increments with an outside diameter tolerance of:

2 in. to 2 1/2 in. ≤ 0.010 in.

2 1/2 in. to 3 in. $\leq$ 0.012 in.
3 in. to 4 in. $\leq$ 0.015 in.

Wall Thickness -

Typically 0.203 in. to 0.300 in. depending on tube outside diameter for a 2,400 psig cycle. In accordance with ASTM A450, thickness can vary 22 percent over minimum wall.

Tube Center-to-Center Tolerance -

A noncumulative $\pm 1/16$ in. manufacturing tolerance is typically allowed. The total panel width of approximately 27 tubes (6 ft to 8 ft) has a tolerance of $\pm 1/8$ in.

b. Conventional Wall Panels

Furnace wall panels consist of tubes welded tangentially along their length or tubes longitudinally welded to 1/4 in. by 1/2 in. carbon steel membranes to an overall width of 8 ft to 10 ft. The length of the panel varies but is normally in the range of 20 ft to 60 ft long.

c. Thermionic Wall Panels

These wall panels will consist of tubes, square on the outside and having a circular bore. For the walls, thermionics will be mounted on one outside face and the tubes will be welded longitudinally. It is expected that the base tube material will be carbon steel; that the tolerance will be about the same as for circular tubes; the center to center distance will also be about the same; and the panel length will be limited by handling requirements of the thermionics. It is likely that a membrane will not be required.

d. Platen Tubes

The platen tubes will have the same shape as the wall tubes. Thermionics will be applied to two faces rather than one. Tubes are not attached along their length as in the case of the wall panels. Tube length is limited either by the fabrication method or by handling requirements of the thermionics.

III. THERMIONIC TOPPING OF ADVANCED DESIGN BOILERS

A. Description of Physical Features

1. Introduction

The configuration of the advanced design thermionic boiler is shown in Figure III-1. This boiler will be designed to burn about 22 percent less fuel than a conventional unit producing 800 MWe station output because of the higher efficiency of a thermionically topped cycle.

Since approximately 500 MWe will be generated by the conventional steam cycle and the remainder by thermionics, the quantity of air preheater, superheat, and reheat surface will be more closely related to a conventional 500-600 MWe unit. The furnace surface, however, will be significantly greater than that of an 800 MWe unit in order to accommodate the thermionics.

2. Major Components

- a. Regenerative air preheater
- b. Thermionic air heater
- c. Thermionic furnace
- d. Burners
- e. Superheater
- f. Reheater

3. Major Component Description

a. Regenerative Air Preheater

The regenerative air preheater would be a standard Ljungstrom or Rothemuhle unit perhaps with no more surface than that required in a conventional 500 MWe fossil-fired boiler.

b. Thermionic Air Heater

To increase the flame temperature to 3,200 F, a thermionic air heater is used. The thermionic air heater would be located on the front and rear walls of the furnace, covering as much surface as required to preheat the combustion air from 600 F to 800 F or higher. The outlet air temperature would be designed to maintain a combustion zone temperature of 3,200 F.

The thermionic air heater would consist of a number of heat pipe cooled thermionic devices. The emitter of each device would be exposed to the 3,200 F furnace and each collector attached to a heat pipe located in the 600 to 800 F combustion air duct.

FIGURE III-1
ADVANCED THERMIONIC
BOILER

c. Thermionic Furnace and Burners

To maintain a constant 3,200 F in the lower half of the furnace, it is envisioned that the furnace will have to be divided into separate compartments, i.e., division walls running from the front to the rear furnace wall and extending from the roof down through the floor at approximately 8 ft intervals are conceived. The 8 ft spacing is set by the burner size.

Normally in a large utility boiler, burners of 200 to 250 million Btu/hr heat input would be used. Because the temperature must be maintained essentially constant over a large portion of the furnace, the flame must be spread over this area. Rather than a few high heat input burners, many small burners, say 50 million Btu/hr each, would be spread along the thermionic air heater wall, dispersed between the heat pipes.

For approximately the lower half of the furnace height, the thermionic air heater forms the front and rear walls. The side walls of each compartment in this lower zone and the remainder of the furnace (all walls) are covered with water cooled thermionics.

Between the lower furnace, operating at a constant 3,200 F, and the furnace exit, the gas temperatures would drop to approximately 2,400 F. This transition zone would have thermionics covering all water-cooled walls. It is likely that some superheat and reheat surface would be located in the upper furnace.

4. Superheat and Reheat Surfaces

Economizer surfaces are normally located in a low temperature zone of the boiler. To obtain as much benefit as possible from water-cooled thermionics surface, the economizer has been relocated from its normal position and equivalent surface placed at the bottom of the furnace in the form of water-cooled furnace wall.

Functionally, in addition to preheating the feedwater, in conventional designs, the economizer reduces the flue gas temperature to a level of between 600 and 800 F. Since the flue gas temperature must be reduced in some manner, and since economizer service has been relocated, superheat and reheat surface must be used to absorb this heat. Typically low temperature superheat surface is only about one-half as effective in absorbing heat as economizer surface because of the steam film resistance. For this study, it therefore has been assumed that the economizer surface will have to be replaced with about twice as much low temperature superheat surface in the convection section of the boiler.

B. Basis for Selection of Features

1. Potential Boiler Fuel

The fuel specified to be used in this study is liquid fuel derived from coal. There are four second-generation schemes:

- a. Direct hydrogenation
- b. Solvent extraction
- c. Pyrolysis
- d. Indirect liquefaction

One particularly attractive process is the direct hydrogenation process. The Department of Energy (DOE) will fund a program centered about a pilot coal-liquefaction plant in Catlettsburg, Kentucky. The low-ash, low-sulfur fuel will be suitable for clean electric power generation and would be a definite candidate for thermionic boilers.

Based on a March 1977 EPRI study prepared by S&W (Reference 5), it is believed that, in view of increasing petroleum depletion cost effects, the possibility of coal conversion process commercialization is good, predicated on available data. According to the same report, under certain stated assumptions, the price of petroleum and coal-derived fuel would be about the same in 1997. It is expected that this cost will be over \$5 per million Btu at that time.

2. Boiler Temperature Profile

In order to operate the thermionic converters at a high efficiency and high output power density, a high input heat flux and hence high gas temperature is necessary. A design goal of the advanced boiler is to maintain the highest allowable gas temperature over as large an area of the boiler as possible. An upper limit of 3,200 F was placed on the gas temperature, since above this temperature NO_x formation is greatly accelerated and the stack emissions would not be environmentally acceptable.

3. Basis for Burner Size and Configuration of Boiler Modules

To maintain this maximum temperature over as large an area of the boiler as possible, numerous small (approximately 50×10^6 Btu/hr) burners are used 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 (vaporization duty) has been added to the evaporator section of the boiler. At that point, approximately 2,345 F, the gas will be directed to the superheater/reheater section. Since the 50×10^6 Btu/hr burners require only 4 ft clearance on either side to prevent flame impingement on the waterwall, the use of these burners permits the boiler to be divided into several modules each approximately 8 ft wide. This modular construction maximizes

the waterfall surface to furnace volume ratio and thus leads to a smaller overall boiler. The burner selection also dictates the width of the boiler. These burners develop a flame approximately 15 ft long and, since front and back firing is employed, the boiler width is set at 30 ft.

With the boiler temperature profile defined, it is possible to calculate the amount of energy and heat transfer area of any temperature zone, based on the thermionic converter performance at that temperature. (See detailed calculations Appendix F and G.)

The required heat transfer area coupled with the cross section dimensions established by burner size define the height of any temperature zone of the boiler. Once the total required height of the 3,200 F isothermal zone was established, it was divided into a sufficient number of modules to limit the average gas residence time at 3,200 F to approximately 1 second in order to limit NO_x formation.

The total number of burners required depends on the plant size, station heat rate, and burner size. In this case, 800 MWe, 7,235 Btu/kW, and 50×10^6 Btu/hr, respectively; thus, a total of 5.78×10^9 Btu/hr or approximately 14 burners for each of the 8 boiler modules.

C. Thermal Performance of the Advanced Design Boiler

The thermal performance of the advanced design boiler is detailed in Appendix F. Application of the thermionic topping cycle yields a plant with an overall efficiency of 47.2 percent. This compares to the 41.2 percent efficiency for the conventional boiler with thermionic topping and 37.4 percent for the conventional boiler with no topping cycle. Of the 833 MWe (gross) plant output, 524 MWe are generated by the turbine and 309 MWe by the thermionic converters.

The scope of this study did not permit optimization of this boiler on other than a thermal efficiency basis; however, in the initial evaluation it was judged uneconomical to place thermionics in the lowest temperature region of the evaporator. When the cost estimates are analyzed, it may prove economically attractive to remove the thermionics from the 2,600 F to 2,400 F zone of the evaporator; this would result in a small penalty on efficiency but may provide a significant direct cost savings.

D. Electrical Concepts and Performance

1. Thermionic Converter Configuration

There are two basic converter geometries for the advanced boiler design, corresponding to the waterwall panels and the air-cooled panels. For the waterwall panels, the converter area and converter interconnections will be adjusted to satisfy constant current constraints and to complement tube design. For the air-cooled thermionic panels, the configuration will be coordinated with the required distribution of burners.

2. Boiler Zones Selected for Application of Thermionics

There are five zones that have been described in the conceptual model of the advanced boiler. Zone 1, which is the fuel firing zone, has uniform conditions. The region above the fuel burning zone sees a continuous gradient of decreasing combustion gas temperature. This gradient in operating conditions is represented by the definition of four zones with constant conditions, but varying from one to another. Thermionic device operating parameters have been determined for each zone. Since each "string" of interconnected converter(s) will, in general, extend through all the zones, converter(s) geometry in each string will vary from zone to zone. The parameters for converter(s) are delineated, per zone, below:

TABLE III-1

CONVERTER(S) PARAMETERS IN EACH ZONE OF ADVANCED BOILER

Converter(s) Parameter	Zone 1 Waterwalls	Zone 1 Air Preheat	Zone 2	Zone 3	Zone 4	Zone 5
T_E ($^{\circ}\text{K}$)	1,600	1,600	1,575	1,525	1,450	1,400
T_C ($^{\circ}\text{K}$)	670	700	670	670	660	660
P_{in} (w/cm^2)	22	22	19.5	15	11.8	7.5
P_{out} (w/cm^2)	7.93	7.59	6.88	4.95	3.77	2.21
V_{out} (volts)	1.3	1.3	1.3	1.2	1.1	1.1
J (amp/cm^2)	6.09	5.83	5.29	4.12	3.43	2.01

3. Interconnection of Thermionic Converter(s)

Converter(s) are connected in parallel and series combinations to form strings, similar to the description in Section II-E. However, the model of the advanced boiler is more refined, in that it addresses gradations in internal boiler conditions. This is reflected in the determination of surface areas for series connected units, since all units shall be constrained to constant current for a given string. This requires, for example, that units of the same string passing through both Zone 1 and Zone 5 have a surface area ratio of 6.09:2.01, or approximately 3:1. That is, the series units in Zone 5 shall have three times the surface area of those in Zone 1. The maximum string voltage buildup would be 400 V d-c (see calculation Appendix I). With midpoint grounding, this would be reduced to 200 V. The maximum power output of this string would be 540 kW.

4. Electrical Performance of Interconnected Converter(s)

Electrical performance (power output at string terminals) has been calculated for each of the zones above (reference Appendix I). The calculations are based on 78 percent of the total surface area being available to thermionics on the air-cooled panels and 95 percent available area on the waterwall panels. In addition, 60 mV of drop are allowed

per series connection between converters. This translates into a power loss of approximately 5 percent. In Zone 1, for the preheat air-cooled panels, the power output per boiler module is 2.1 MWe. For Zone 1, waterwall, the power output is 21.9 MWe per module, and, for Zones 2, 3, 4, and 5, the outputs are 4.4, 4.32, 3.92, and 3.57 MWe, respectively, for a total of 40.21 MWe per module. For eight modules, 322 MWe represents the total power available at the string terminals. Losses in the bus work that interconnects this output with the d-c to a-c motor generator conversion sets and losses in the conversion equipment should not together exceed a total of 8 percent, thus confirming 300 MWe available at the generator output terminals.

5. Electrical System Fault Protection

The electrical interconnection of converters shall form a large number of "strings," each string producing approximately 1,200 amps at 400 V d-c, balanced to ground.

The current from a single string could be destructive if it flowed through paths not designed to carry electric current. And the combined currents from a number of strings could cause catastrophic damage if permitted to flow through such paths. Several failure modes and proposed protective features are discussed in the following paragraphs.

It is proposed that the electrical mid-point of each series string be grounded through a 40 ohm, 1,000 W resistor. This scheme would limit the voltage across any single insulation structure to +200 V, limit the current flow into the first ground fault to 5 amp, and provide a convenient way of monitoring each string to provide indications of either open circuit or short circuit. The monitor would sense either voltage across or current through the grounding resistor, both of which normally would be zero.

The most likely path for fault current is through failed insulation between a collector and the grounded tube on which it is mounted. The fault current for the first failure would depend on how far the fault is from the mid-point of the string, but in any case would not exceed 5 amp. A second ground fault in the same string would produce initial fault current in converters between fault points somewhat larger than full load current. However, it is expected that the initial current would decay rapidly to a lower value as the emitter temperatures are forced downward by the reduced load resistance presented to the faulted portion of the strings.

However, without some method of instantly isolating the string, very large currents might flow from the bus to the fault. Such currents would be capable of vaporizing large masses of metal in the path. To guard against destructive electrical faults, it is proposed that each string be protected from reverse current by connecting each end of the string to its respective bus through a silicon diode. Diodes with ratings adequate for this service are available. These diodes, in

the aggregate, will introduce a loss of about 0.35 percent of available d-c energy.

Another failure mode which must be considered is a collector to emitter fault in one or more converters. Under these conditions, the current delivered to the bus by the faulted string will decrease, the emitter temperature will rise, and a voltage will appear across the mid-point grounding resistor. Therefore, appropriate protective measures can be initiated automatically, as described below.

If a series connector fails (open circuit), an arc will be drawn at the point of failure which could persist until a considerable amount of local damage has been done. And if the arc is extinguished, the current in the string will be interrupted, resulting in a large rise in all emitter temperatures within the string. This type of failure can be detected by monitoring voltage drop across the diode at either end of the strings, but there may be no feasible protective measure short of fuel shutoff in the affected module.

With the diodes in the circuit, it becomes feasible to deliberately short-circuit individual strings, for control or protective reasons. It is therefore proposed that a single pole electrically-operated circuit breaker be connected across each string. During start-up, all breakers would be closed, permitting more rapid build-up of steam pressure. After one or more acyclic motors had been cranked to a speed which would permit introduction of liquid metal for the connection with the rotating armature, these breakers would be opened, one at a time, to accelerate the motor generator sets to synchronizing speed and to load them after they are synchronized to the power system.

During shutdown, the breakers would be reclosed manually to remove current from the motor circuit so that the liquid metal could be withdrawn before shutdown.

Under fault conditions, appropriate individual breakers would be closed rapidly and automatically by protective relays to effectively isolate the faulted string. This action would protect other thermionic converters in that string, to the extent possible, from further damage, and would protect the boiler tubes on which they are mounted.

IV. ECONOMIC FEASIBILITY EVALUATION OF THERMIONIC TOPPED PLANT

A. Approach

1. The estimated cost of an 800 MWe conventional oil-fired plant was developed and is shown in Table IV-1. These estimated costs are arranged by the FPC block method of cost accounting and include interest (at 10 percent) and escalation (at 6.5 percent) during construction. The detailed breakdown of items included in each numbered block is given in Reference 6.
2. The estimated cost of an 800 MWe thermionic topped plant was developed by estimating the cost of the advanced thermionic boiler (estimated cost of thermionic converters was furnished by Thermo Electron Corporation) and other equipment associated with the use of thermionics (to generate 300 MWe) and the cost of balance of plant equipment appropriately sized for output of 500 MWe. These estimated costs are also shown in Table IV-1.
3. The capitalized fuel costs for liquified coal were calculated for the 30-year period starting in 1977. These calculations are in Appendix H.

B. Results

Preliminary cost estimates indicate that a TEC topped liquified coal-fired plant (800 MWe total) using an advanced thermionic boiler can be constructed for \$618 per KWe (1977 dollars), compared to a cost of \$503 per KWe for a conventional 800 MWE liquified coal-fired plant.

Using a solvent refined liquified coal levelized cost of \$4.92 per million Btu (see Appendix H) and a fixed charge rate of 18 percent, the capitalized savings in fuel costs (for 30 years) is \$217 million and the total capital savings of \$156 per KWe could be achieved by using TEC topping.

TABLE IV-1

THERMIONIC TOPPING OF POWER PLANTS - ORDER OF MAGNITUDE ESTIMATE

Page 1 of 3

Block		800 MWe PLANT WITH ADVANCE THERMIONIC BOILER BURNING LIQUIFIED COAL				800 MWe CONVENTIONAL PLANT BURNING LIQUIFIED COAL			
		M	L	T	MH	M	L	T	MH
		(000)	(000)	(000)	(000)	(000)	(000)	(000)	(000)
1	Land and Land Rights								
			By Client					By Client	
2	Yard Work	\$925	\$6,425	\$7,350	263	\$1,220	\$8,550	\$9,770	485
3	Main Power House	6,700	10,000	16,700	579	9,115	13,430	22,545	778
4	Administration Building	516	716	1,232	45	717	1,002	1,719	66
5	Miscellaneous Buildings								
6A	Boiler Equipment by Vendor	16,630	8,815	25,445	383	31,500	16,700	48,200	704
6B	TECO Panels, consisting of thermionic converters and boiler tubes	89,045	6,900	95,945	300		N/A		
6C	M.G. Set, Switchgears, Bus-bars and Accessories	25,354	2,188	27,542	110		N/A		
7A	Balance of Boiler Plant	10,200	8,750	18,950	496	15,100	12,930	28,030	734
7B	Ash Handling	610	500	1,110	34	610	500	1,110	34
8A	Coal Handling		N/A					N/A	
8B	F.O. Equipment and Structures	3,300	3,960	7,260	227	3,300	3,960	7,260	227
8C	Start-up Oil (Light oil tank and equipment)	250	380	630	20	250	380	630	20
9	Stack (concrete w/liner) - 600 ft	895	1,830	2,725	88	1,276	2,925	4,201	132
10	Precipitator	2,200	1,700	3,900	62	2,750	2,100	4,850	77
11	Scrubber - SO ₂		N/A					N/A	
12	Turbine Generator by Vendor	24,500	2,200	26,700	82	33,570	3,555	37,125	120
13	Balance of Turbine Generator Plant	2,750	1,570	4,320	203	4,260	3,110	7,370	120

TABLE IV -1 (Cont)

THERMIONIC TOPPING OF POWER PLANTS - ORDER OF MAGNITUDE ESTIMATE

Page 2 of 3

		800 MWe PLANT WITH ADVANCE THERMIONIC BOILER BURNING LIQUIFIED COAL				800 MWe CONVENTIONAL PLANT BURNING LIQUIFIED COAL			
<u>Block</u>		<u>M</u> <u>(000)</u>	<u>L</u> <u>(000)</u>	<u>T</u> <u>(000)</u>	<u>MH</u> <u>(000)</u>	<u>M</u> <u>(000)</u>	<u>L</u> <u>(000)</u>	<u>T</u> <u>(000)</u>	<u>MH</u> <u>(000)</u>
14	Circ. Water System								
A	Screenwell Structure								
B	Condenser System								
C	Intake and Discharge System	\$8,965	\$5,200	\$14,165	307	\$12,950	\$5,810	\$18,760	345
D									
E	N.D. Cooling Tower								
15	Water Treatment	680	140	820	10	1,010	197	1,207	14
16	Waste Treatment	100	110	210	5	420	155	575	7
17	Accessory Electrical Equipment	5,930	4,720	10,650	257	8,130	6,885	15,015	381
18	Miscellaneous Power Plant Equipment	1,785	920	2,705	79	2,490	1,280	3,770	110
19	Main Transformers, Including Foundations and Accessories	2,466	359	2,825	21	1,538	446	1,984	26
20	Transmission Lines		N/I				N/I		
	Total Direct Accounts	\$203,801	\$67,383	\$271,184	3,571	\$130,206	\$83,915	\$214,121	4,380

TABLE IV-1 (Cont)

THERMIONIC TOPPING OF POWER PLANTS - ORDER OF MAGNITUDE ESTIMATE

Page 3 of 3

Acct. No.		800 MWe PLANT WITH ADVANCE THERMIONIC BOILER BURNING LIQUIFIED COAL				800 MWe CONVENTIONAL PLANT BURNING LIQUIFIED COAL			
		M (000)	L (000)	T (000)	MH (000)	M (000)	L (000)	T (000)	MH (000)
8030)									
8040)	Distributable and Indirect Cost	\$25,386	\$39,700	\$65,086	572	\$20,093	\$31,296	\$51,389	700
8100)									
	Total, Construction and Indirect Cost	229,187	107,083	336,270	4,143	150,299	115,211	265,510	5,080
8300	Allowance for Indeterminates	<u>22,913</u>	<u>10,717</u>	<u>33,630</u>	<u>415</u>	<u>15,101</u>	<u>11,589</u>	<u>26,690</u>	<u>508</u>
	Total, Estimate - Present Day (12/1/77)	252,100	117,800	369,900	4,558	165,400	126,800	292,200	5,588
	Escalation	<u>41,092</u>	<u>22,608</u>	<u>63,700</u>		<u>29,700</u>	<u>26,800</u>	<u>56,500</u>	
	Total, Including Escalation	293,192	140,408	433,600	4,558	195,100	153,600	348,700	5,588
	Interest During Construction			<u>60,700</u>				<u>53,500</u>	
	Total, Including Interest During Construction			494,300	4,558			402,200	5,588
	Cost/KWe			618				503	
	Δ Cost/KWe (618-503)			115					
	Capitalized Fuel Cost Savings/KWe			271					
	Net Capital Savings/KWe (271-115)			156					

V. PLANNING FOR THERMIONIC HARDWARE TESTING

A. Approach for Hardware Test

A hardware test panel containing several thermionic devices should be designed to permit mounting in the high temperature zone of an operating boiler. The test panel should be fabricated the same size as an observation or access door, so the door can be removed and the test panel put in its place. After completion of the testing the test panel could be removed and the door replaced. If the boiler selected for test is a balanced draft unit, these units could be installed, inspected, or removed even with the boiler in operation.

The installed test panel would expose the thermionic devices to representative combustion gas temperatures and heat fluxes and would permit test and demonstration of the thermionics in the application environment.

The module should have at least the following connections:

1. Cooling water, inlet and outlet
2. Electrical connections
3. Hot and cold face thermocouple array

At least the following variables should be measured and recorded:

1. Cooling water flow
2. Cooling water inlet and outlet temperature
3. Overall voltage and current
4. Voltage and current at interconnections, as required
5. Hot and cold face temperatures

B. Suggested Plants For Testing

As confirmed by letters to Thermo Electron Corporation, New England Gas and Electric (NEGEA) and New England Power Company (NEPCo.) have expressed interest in discussing the application of thermionics to one of their operating units. NEGEA has only oil-fired units, but NEPCo. has both coal and oil.

In addition, Boston Edison also has expressed interest in the past.

VI. REFERENCES

1. Stone & Webster Engineering Corporation Report Review of Thermionics for Power Generation, December 29, 1976.
2. Stone & Webster Engineering Corporation Report Study of Thermionics for Power Generation, October 10, 1975.
3. Thermo Electron Corporation Report Thermionic and Steam Bottoming of MHD, Report No. TE 4270-147-77, June 1977.
4. Thermo Electron Corporation Phase One Report Thermionic Conversion Topping System Study, Report No. TE 4207-29-76, December 15, 1975.
5. Northeast Coal Utilization Program, Phase I Final Report, Stone & Webster Engineering Corporation, March 1977.
6. FPC Classification of Accounts by Blocks, March 20, 1972. (Adapted from FPC Uniform System of Accounts Prescribed for Public Utilities and Licensees, March 1, 1965.)

CALCULATION SUMMARY

▲ 5010.47

CALCULATION NO.

APPENDIX A

DATE

10/17/77

PAGE

1 of 4

CLIENT

THERMO ELECTRON CORP.

J. O. NUMBER

12628.02

SUBJECT

EMITTER TEMP FOR VARIOUS FLAME TEMPERATURES.

PROBLEM:

DETERMINE THE INPUT HEAT FLUX AND EMITTER TEMP. FOR VARIOUS FLAME TEMPERATURES.

AUDIT

REMARKS

CHECKER'S REMARKS:

APPROACH/ASSUMPTIONS:

CROSS PLOT INPUT HEAT FLUX AVAILABLE WITH HEAT FLUX REQUIRED AT VARIOUS EMITTER TEMPERATURES. THE RESULTANT EMITTER TEMPERATURES ASSUME A COLLECTOR TEMP. OF 700°K.

AUDIT

REMARKS

CHECKER'S REMARKS:

SOURCES—DATA/EQUATIONS:

SEE REFERENCES AND ASSUMPTIONS ON PAGE 2.

AUDIT

REMARKS

CHECKER'S REMARKS:

CONCLUSIONS:

EMITTER TEMP. CAN BE FOUND FROM FIGURE 1 ON PAGE 4.

AUDIT

REMARKS

CHECKER'S REMARKS:

AUDIT SUMMARY

CHECKS

CORRECTIVE ACTION TAKEN BY

CALCULATED BY

Checked by

DATE

10/17/77

DATE

11/14/77

DATE

11/14/77

INFRACTIONS

DATE

AUDITED BY

Client THERMO ELECTRON CORP

Location

Est. No.

J.O. No. 12628.02

Subject EMITTER TEMP FOR VARIOUS FLAME TEMPS.

Date 10/17/77

By BJD

Checked 11/14/77

By RJS

Based on

Revised

By

REFERENCES:

1. STUDY OF THERMIONICS FOR POWER GENERATION
S&W J.O. 12628 OCT 10, 1975
2. REVIEW OF THERMIONICS FOR POWER GENERATION
S&W J.O. 12628 DEC 29, 1976
3. THERMIONIC AND STEAM BOTTOMING OF
MAGNETOHYDRODYNAMICS, THERMO ELECTRON
REPORT NO. TE4220-147-77 JUNE 1977

ASSUMPTIONS:

BASED ON REF. 1. THE HEAT FLUX FROM AN OIL FLAME AT 2800°F TO A BOILER WATERWALL AT 750°F IS 70,000 BTU/hr-ft²

THIS HEAT FLUX IS PROPORTIONAL TO THE DIFFERENCE OF THE FOURTH POWERS OF THE FLAME AND WALL ABSOLUTE TEMPERATURES.

$$(3,260)^4 - (1,210)^4 = 111 \propto 70,000 \text{ BTU/hr-ft}^2$$

CALCULATIONS:

IN ORDER TO DETERMINE THE HEAT FLUX TO THE EMITTER FOR A PARTICULAR FLAME TEMPERATURE THE DIFFERENCE IN THE FOURTH POWER OF THE FLAME AND EMITTER ABSOLUTE TEMP ARE SCALED TO THE DIFFERENCE ABOVE WHICH PRODUCED 70,000 BTU/hr-ft² HEAT FLUX.

THUS FOR A FLAME TEMP OF 3200°F AND AN ASSUMED EMITTER TEMP OF 2060°F (1400°K) THE HEAT FLUX TO THE EMITTER WOULD BE,

$$\frac{(3,660)^4 - (2,520)^4}{111} 70,000 \text{ BTU/hr-ft}^2 = 87,730 \frac{\text{BTU}}{\text{hr-ft}^2}$$

OR 27.6 W/cm²

STONE & WEBSTER ENGINEERING CORPORATION
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Preliminary
Item A

1 Client THERMO ELECTRON CORP Location Est. No. J.O. No. 12628.02
2 Subject EMITTER TEMP FOR VARIOUS FLAME TEMPS. Date 10/17/77 By BJD
3 Checked 11/14/77 By R90.
4 Based on Revised By

5
6
7 THE SAME PROCEDURE WAS FOLLOWED FOR THE
8 SEVERAL GAS TEMPERATURES OF INTEREST AT
9 VARIOUS ASSUMED EMITTER TEMPERATURES. THIS
10 DATA IS PRESENTED AS THE FAMILY OF CURVES
11 REPRESENTING "HEAT INPUT AVAILABLE" IN
12 FIGURE 1.
13

14 DATA FROM REFERENCE 3, FIGURE 4 WAS
15 USED TO CONSTRUCT A CURVE OF "HEAT INPUT
16 REQUIRED" (800°F COLLECTOR TEMP) FOR VARIOUS
17 EMITTER TEMPERATURES. THIS CURVE IS ALSO
18 PLOTTED IN FIGURE 1.
19

20 THE TEMPERATURE AT WHICH THE EMITTER WILL
21 OPERATE FOR A PARTICULAR FLAME TEMP. IS
22 DETERMINED BY THE INTERSECTION OF THE
23 "HEAT INPUT AVAILABLE" & "HEAT INPUT REQUIRED"
24 CURVES
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CALCULATION SHEET

A Page No. 4-4

Preliminary

Item

A

Client THERMO ELECTRON CORP

Location

Est. No.

J.O. No. 12628.02

Subject EMITTER TEMP FOR VARIOUS FLAME TEMP.

Date 10/17/77

By BJD

Checked 11/4/77

By R.D.

Based on

* Revised 11/14/77

By R.D.

FIGURE 1

* IF DECREASING COLLECTOR TEMPERATURES ACCOMPANYING DECREASING HEAT FLUXES ARE CONSIDERED, THE SOLID CURVE WOULD SHIFT TO THE DASHED CURVE. THE CALC'S IN FOLLOWING APPENDICES USED VALUES FROM SOLID CURVE.

CALCULATION SUMMARY

▲ 5010.47

CALCULATION NO.

APPENDIX B

DATE

10/17/77

PAGE

1 of 3

CLIENT

THERMO ELECTRON CORP

J. O. NUMBER

12628.02

SUBJECT

THEORETICAL AIR-FUEL RATIO FOR LIQUIFIED COAL

PROBLEM:

DETERMINE THE THEORETICAL AIR TO FUEL RATIO
FOR SOLVENT REFINED LIQUIFIED COAL.

AUDIT
REMARKS

CHECKER'S REMARKS:

APPROACH/ASSUMPTIONS:

TRADITIONAL COMBUSTION CALCULATION BASED ON
CONSERVATION OF MASS.

AUDIT
REMARKS

CHECKER'S REMARKS:

SOURCES—DATA/EQUATIONS:

1. THE SCR-II PROCESS - 3RD ANNUAL INTERNATIONAL CONF.
ON COAL GASIFICATION & LIQUIFACTION. B.K. SCHMIDT, &
D.M. JACKSON, GULF MINERAL RESOURCES CO. DENVER COLO.
2. STEAM ITS GENERATION AND USE COPYRIGHT 1972 B+W
PG 6-13 FIG 3 PSYCHROMETRIC CHART

AUDIT
REMARKS

CHECKER'S REMARKS:

CONCLUSIONS:

THEORETICAL AIR 12.59 lbm air/lbm fuel (DRY)

AUDIT
REMARKS

CHECKER'S REMARKS:

AUDIT SUMMARY

CHECKS

CORRECTIVE ACTION TAKEN
BY

CALCULATED BY

CHECKED BY

DATE

DATE

INFRACTIONS

DATE

AUDITED BY

DATE

Brian J. Voherty
Richard J. Dwyer

10/17/77

11/14/77

1 Client THERMO ELECTRON CORP Location _____ Est. No. _____ J.O. No. 12628.02
2 Subject _____ Date 10/17/77 By BTD
3 _____ Checked 11/14/77 By RP
4 Based on _____ Revised _____ By _____

8 TYPICAL PROPERTIES OF SCR II FUEL OIL (REF 1)

10 ELEMENT ANALYSIS % BY WT.

CARBON	87.2
HYDROGEN	7.9
SULFUR	0.3
NITROGEN	0.9
OXYGEN	3.9
	<u>100</u>

19 HEATING VALUE BTU/lb = 17,284 (REF 1)

23 CALCULATION OF AIR-FUEL RATIO

25 $\frac{\text{MOLES C}}{100 \text{ lbm fuel}} = \frac{87.2}{12} = 7.27$

29 $\frac{\text{MOLES H}_2}{100 \text{ lbm fuel}} = \frac{7.9}{2} = 3.95$

33 $\frac{\text{MOLES S}}{100 \text{ lbm fuel}} = \frac{0.3}{32} = 0.009$

37 $\frac{\text{MOLES N}_2}{100 \text{ lbm fuel}} = \frac{0.9}{28} = 0.03$

41 $\frac{\text{MOLES O}_2}{100 \text{ lbm fuel}} = \frac{3.9}{32} = 0.12$

STONE & WEBSTER ENGINEERING CORPORATION
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B

1 Client THERMO ELECTRON CORP Location _____ Est. No. _____ J.O. No. 12628.02
 2 Subject _____ Date 10/17/77 By BTD
 3 _____ Checked 11/14/77 By RJS
 4 Based on _____ Revised _____ By _____

$$\begin{aligned} &9.254 \text{ moles O}_2 / 100 \text{ lbm fuel} \\ &- 0.12 \text{ moles O}_2 \text{ in fuel} / 100 \text{ lbm fuel} \end{aligned}$$

$$9.134 \text{ moles O}_2 \text{ from air} / 100 \text{ lbm fuel}$$

$$\text{THEORETICAL AIR} = \frac{\begin{matrix} \text{moles} \\ \text{O}_2 \end{matrix} \begin{matrix} \text{moles} \\ \text{N}_2 \end{matrix} \begin{matrix} \text{lbm/mole air} \end{matrix} [9.134 + 9.134(3.76)] 28.96}{100} = 12.59 \frac{\text{lbm air}}{\text{lbm fuel}}$$

OR 7.28 lbm air / 10⁴ BTU fired

IF AIR AT 60°F & 60% RELATIVE HUMIDITY THEN

0.006 lbm H₂O / lbm AIR* WILL BE SUPPLIED OR

$$\underline{\underline{12.66 \text{ lbm AIR} / \text{lbm FUEL}}}$$

* REF 2

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APPENDIX C

DATE

10/21/77

PAGE

1 of 4

CLIENT

THERMO ELECTRON CORP

J. O. NUMBER

12628.02

SUBJECT

THERMIONIC CONVERTER COLLECTOR TEMP

PROBLEM:

CALCULATE T_c FOR TYPICAL THERMIONIC DEVICE COUPLED DIRECTLY TO THE BOILER TUBE. RESULTS TO BE USED TO DETERMINE THERMIONIC CONVERTER PERFORMANCE IN TOPPING CYCLE CALCS.

AUDIT
REMARKS

CHECKER'S REMARKS:

APPROACH/ASSUMPTIONS:

ASSUME ONE DIMENSIONAL HEAT TRANSFER FOR GEOMETRY SHOWN ON PAGE 2.

NO ALLOWANCE HAS BEEN MADE FOR THERMAL CONDUCTIVITY PROPERTY OF THE INTERFACE BETWEEN THE ELECTRICAL INSULATOR AND COLLECTOR / BOILER TUBE.

AUDIT
REMARKS

CHECKER'S REMARKS:

SOURCES—DATA/EQUATIONS:

JENS-LOTTE NUCLEATE BOILING HEAT TRANSFER CORRELATION
 $(T_w - T_{SAT}) = 60 e^{-P/900} \left(\frac{Q}{106} \right)^{0.25}$

ONE DIMENSIONAL HEAT TRANSFER

$$Q = KA \Delta T$$

AUDIT
REMARKS

CHECKER'S REMARKS:

CONCLUSIONS:

COLLECTOR TEMPERATURES SHOULD RANGE BETWEEN 650° - 700°K FOR HEAT FLUXES OF INTEREST.

AUDIT
REMARKS

CHECKER'S REMARKS:

AUDIT SUMMARY		CALCULATED BY		DATE	
CHECKS	CORRECTIVE ACTION TAKEN BY	Brian J. DeSanty		10/21/77	
INFRACTIONS	DATE			11/14/77	
		CHECKED BY		DATE	
		Richard J. Dube		11/14/77	
		AUDITED BY		DATE	

Client THERMO ELECTRON CORP Location _____ Est. No. _____ J.O. No. 12628.02
Subject _____ Date 10/21/77 By BJD
Checked 11/14/77 By RJD
Based on _____ Revised _____ By _____

TYPICAL CROSS SECTION
OF BOILER TUBE & THERMIONIC CONVERTER

BOILER TUBE
CARBON STEEL $\frac{1}{4}$ " THICK
 $k = 23 \text{ BTU ft/hr ft}^2 \text{ } ^\circ\text{F}$

THERMIONIC COLLECTOR
STAINLESS STEEL $\frac{1}{16}$ " THICK
 $k = 12 \text{ BTU ft/hr ft}^2 \text{ } ^\circ\text{F}$

SAT. LIQUID
 $T = 673.9$
 $P = 2600 \text{ PSIA}$

ELECTRICAL INSULATION $\frac{1}{32}$ " THICK
 $k = 16 \text{ BTU ft/hr ft}^2 \text{ } ^\circ\text{F}$

I. HEAT FLUX

COLLECTOR TEMP WILL BE CALCULATED FOR
TWO EMITTER HEAT FLUXES (22 W/cm^2 & 11 W/cm^2)

AT 22 W/cm^2 $\eta_T = 36\%$

THUS COLLECTOR HEAT FLUX =

$$22 \text{ W/cm}^2 (1 - 0.36) = 14.08 \text{ W/cm}^2 = \underline{\underline{44,634 \frac{\text{BTU}}{\text{ft}^2 \text{ Hr}}}}$$

AT 11 W/cm^2 $\eta_T = 32\%$

THUS COLLECTOR HEAT FLUX =

$$11 \text{ W/cm}^2 (1 - 0.32) = 7.48 \text{ W/cm}^2 = \underline{\underline{23,712 \frac{\text{BTU}}{\text{ft}^2 \text{ Hr}}}}$$

II. ΔT SAT LIQUID TO BOILER TUBE

USING JENS-LOTTE'S HEAT TRANSFER CORRELATION
FOR NUCLEATE BOILING HEAT TRANSFER

$$\Delta T = (T_w - T_{\text{SAT}}) = 60 e^{-P/400} \left(\frac{q}{10^6} \right)^{1/4}$$

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 2 Subject _____ Date 10/21/77 By RTD
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WHERE $P = \text{PRESSURE PSIA}$

$\dot{Q} = \text{HEAT FLUX BTU/HR FT}^2$

FOR $Q = 44634 \text{ BTU/hr ft}^2$

$$\Delta T = 60 e^{-2600/900} \left(\frac{44634}{100} \right)^{1/4} = \underline{\underline{1.53^\circ F}}$$

$$\Rightarrow T_w = 673.9 + 1.53 = 675.4^\circ F$$

FOR $Q = 23712 \text{ BTU/hr ft}^2$

$$\Delta T = 60 e^{-2600/900} \left(\frac{23712}{100} \right)^{1/4} = \underline{\underline{1.31^\circ F}}$$

$$\Rightarrow T_w = 673.9 + 1.31 = 675.2^\circ F$$

III. ΔT FOR COLLECTOR/INSUL/BOILER TUBE

$$Q = kA \Delta T$$

WHERE $Q = \text{HEAT FLUX BTU/hr}$

$A = \text{AREA FT}^2$

$k = \text{BTU/hr ft}^2 \text{ OF}$

FOR $1/4"$ BOILER TUBE WALL $k =$

$$23 \frac{\text{BTU ft}}{\text{hr ft}^2 \text{ OF}} \times 12 \text{ IN/ft} \div \frac{1}{4} \text{ IN} = \underline{\underline{1104 \frac{\text{BTU}}{\text{hr ft}^2 \text{ OF}}}}$$

FOR $1/32"$ INSULATION $k =$

$$16 \frac{\text{BTU ft}}{\text{hr ft}^2 \text{ OF}} \times 12 \text{ IN/ft} \div \frac{1}{32} \text{ IN} = \underline{\underline{6144 \frac{\text{BTU}}{\text{hr ft}^2 \text{ OF}}}}$$

FOR $1/16"$ COLLECTOR $k =$

$$12 \frac{\text{BTU ft}}{\text{hr ft}^2 \text{ OF}} \times 12 \text{ IN/ft} \div \frac{1}{16} \text{ IN} = \underline{\underline{2304 \frac{\text{BTU}}{\text{hr ft}^2 \text{ OF}}}}$$

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$$\frac{1}{\bar{k}} = \frac{1}{\frac{1}{1104} + \frac{1}{6144} + \frac{1}{2304}} = \frac{1}{1.5 \times 10^{-3}}$$

$$\bar{k} = 665.5 \frac{\text{BTU}}{\text{hr ft}^2 \text{ } ^\circ\text{F}}$$

$$\Delta T \text{ FOR } 44634 \text{ BTU/hr ft}^2 \text{ } ^\circ\text{F} = \frac{44634}{665.5} = \underline{\underline{67^\circ\text{F}}}$$

$$\Delta T \text{ FOR } 23712 \text{ BTU/hr ft}^2 \text{ } ^\circ\text{F} = \frac{23712}{665.5} = \underline{\underline{35.6^\circ\text{F}}}$$

IV COLLECTOR TEMP.

FOR EMITTER HEAT FLUX 22 W/cm^2

$$T_c = T_w + \Delta T = 675.4^\circ\text{F} + 67^\circ\text{F} = 742.4^\circ\text{F} \\ = \underline{\underline{667.6^\circ\text{K}}}$$

FOR EMITTER HEAT FLUX 11 W/cm^2

$$T_c = T_w + \Delta T = 675.2^\circ\text{F} + 35.6^\circ\text{F} = 710.8^\circ\text{F} \\ = \underline{\underline{650^\circ\text{K}}}$$

CALCULATION SUMMARY

▲ 5010.47

CALCULATION NO.

APPENDIX D

DATE

10/19/77

PAGE

1 of 7

CLIENT

THERMO ELECTRON CORP

J. O. NUMBER

12628.02

SUBJECT

CONVENTIONAL BOILER PERFORMANCE (NO THERMIONICS)

PROBLEM:

DETERMINE THE NET CYCLE EFFICIENCY FOR A CURRENT TECHNOLOGY BOILER USING LIQUIFIED COAL AS A FUEL AND OPERATING ON A 2400 PSL/1000°F/1000°F STEAM CYCLE. TO BE USED FOR COMPARISON WITH THE THERMIONIC TOPPED CYCLES.

AUDIT

REMARKS

CHECKER'S REMARKS:

APPROACH/ASSUMPTIONS:

CALCULATION IS BASED ON 1 lbm FUEL
SEE ASSUMPTIONS PG 4.

AUDIT

REMARKS

CHECKER'S REMARKS:

SOURCES—DATA/EQUATIONS:

SEE REFERENCES PAGE 2

AUDIT

REMARKS

CHECKER'S REMARKS:

CONCLUSIONS:

NET CYCLE EFFICIENCY 37.4%

AUDIT

REMARKS

CHECKER'S REMARKS:

AUDIT SUMMARY

CHECKS

CORRECTIVE ACTION TAKEN
BY

CALCULATED BY

Brian J. Doherty

DATE

10/19/77

CHECKED BY

Richard J. Doherty

DATE

11/14/77

INFRACTIONS

DATE

AUDITED BY

DATE

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D

1 Client THERMO ELECTRON CORP Location Est. No. J.O. No. 12628,02
2 Subject Date 10/19/77 By BJD
3 Checked 11/14/77 By RGS
4 Based on Revised By

REFERENCES:

1. 2400/1000/1000 STEAM CYCLE CALC.
APPENDIX D TO STONE & WEBSTER REPORT
"REVIEW OF THERMIONICS FOR POWER GENERATION"
DEC. 29, 1976 JP. No 12628
2. APPENDIX B OF THIS REPORT
CALCULATION OF THEORETICAL AIR
3. APPENDIX F OF THIS REPORT
CALCULATION OF ADVANCED THERMIONIC
BOILER PERFORMANCE
4. GAS TABLES, KEENAN & KAYES 1948

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1 Client THERMO ELECTRON CORP Location Est. No. J.O. No. 12628.02
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CYCLE ARRANGEMENT

Client THERMO ELECTRON CORP Location

Est. No.

J.O. No. 12628.02

Subject

Date 10/19/77

By BJD

Checked 11/14/77

By RJS

Based on

Revised

By

FURNACE CALCULATION:

ASSUMPTIONS:

1. CALCULATION BASED ON 1 lbm/hr OF FUEL AND 105% THEORETICAL AIR
2. MOISTURE LOSSES EXCEPT FOR LATENT HEAT OF H_2O IN FUEL ARE NEGLECTED
3. ENTHALPY OF FUEL AT AMBIENT TEMP IS NEGLECTED
4. ENTHALPY OF COMBUSTION GAS IS ASSUMED EQUAL TO ENTHALPY OF AIR. ALL AIR ENTHALPIES ARE FROM REF 4, TABLE 1
5. AMBIENT AIR AT 60°F & 60% RELATIVE HUMIDITY
6. FUEL USED IS LIQUIFIED COAL SCRT II
HHV = 17284 BTU/lbm

I. MOISTURE LOSS

FROM REF. 3 MOISTURE LOSS FOR THE LIQUIFIED COAL FUEL IS 739.4 BTU/lbm fuel.

II. ENTHALPY OF COMBUSTION GAS

FROM REF. 2 12.66 lbm OF WET AIR ARE REQUIRED FOR EACH lbm OF FUEL. 105% TH = 13.29 lbm

$$h_{\text{AIR AT 600°F}} \quad 13.29 \text{ lbm} \times 255.9 \frac{\text{BTU}}{\text{lbm}} = 3401 \text{ BTU}$$

$$h_{\text{COMBUSTION}} \quad 1 \text{ lbm} \times (17284 - 739.4) \frac{\text{BTU}}{\text{lbm}} = 16544.6 \text{ BTU}$$

$$19945.6 \text{ BTU}$$

$$\frac{19945.6 \text{ BTU}}{1 + 13.29 \text{ lbm comb gas / lbm fuel}} = \underline{\underline{1395.7 \text{ BTU/lbm}}}$$

Client THERMO ELECTRON CORP Location Est. No. J.O. No. 12628.02
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 Based on Revised By

III. ENTHALPY OF GAS LEAVING SUPERHEATER

THIS ENTHALPY IS SET BY LIMITING FLUE GAS TEMP TO 300°F AND BY ENERGY REQUIRED TO PRE HEAT COMBUSTION AIR TO 600°F

ASSUME AIR IS COMPRESSED TO 1.1 ATM, FOR 60°F AIR $h = 124.2 \text{ BTU/lbm}$ $P_r = 1.2122$

AT 1.1 ATM $P_r = 1.2122 \times 1.1 = 1.3334$
 $\Rightarrow h = 127.7 \text{ BTU/lbm}$

$$\Delta h = 127.7 - 124.2 = 3.5 \text{ BTU/lbm}$$

ASSUME FAN 90% EFFICIENT $\Delta h_{\text{ACT}} = \frac{3.5}{.9} = 3.88 \frac{\text{BTU}}{\text{lbm}}$

$$h_{\text{OUT OF FAN}} = 124.2 + 3.88 = 128.08 \frac{\text{BTU}}{\text{lbm}} \Rightarrow T = 76^\circ\text{F}$$

ENTHALPY WHICH MUST BE ADDED TO AIR TO RAISE ITS TEMP FROM 76°F TO 600°F

$$h_{\text{AIR } 600^\circ\text{F}} = 255.9 \text{ BTU/lbm}$$

$$h_{\text{AIR } 76^\circ\text{F}} = 128.08 \text{ BTU/lbm}$$

$$127.8 \text{ BTU/lbm}$$

SINCE 14.29 lbm OF COMBUSTION GAS ARE FORMED FROM 13.29 lbm AIR THE Δh OF THE COMBUSTION GAS IS

$$127.8 \text{ BTU/lbm} \times \frac{13.29}{14.29} = 118.6 \text{ BTU/lbm}$$

THUS ENTHALPY OF GAS LEAVING SUPERHEATER MUST BE

$$h_{\text{GAS } 300^\circ\text{F}} = 182.0 \text{ BTU/lbm}$$

$$\Delta h_{\text{AIR PREHEAT}} = 118.6 \text{ BTU/lbm}$$

$$300.8 \text{ BTU/lbm} \Rightarrow T = 777^\circ\text{F}$$

NOTE: FOR 1 lbm FUEL $3.5 \text{ BTU/lbm} \times 13.29 \text{ lbm AIR} = 1.4 \times 10^{-2} \text{ kW}$
 $3413 \text{ BTU/kW} \times 0.95 (\text{motor eff})$

MUST BE SUPPLIED TO THE FORCED DRAFT FAN.

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IV ENERGY AVAILABLE FOR STEAM CYCLE

THE ENERGY AVAILABLE FOR THE STEAM CYCLE IS
EQUAL TO THE DIFFERENCE BETWEEN THE INITIAL
COMBUSTION GAS ENTHALPY AND THE ENTHALPY OF
THE GAS EXITING THE SUPERHEATER.

$$\begin{aligned} h_{\text{comb gas}} &= 1395.7 \text{ BTU/lbm comb. gas} \\ h_{\text{gas exit of SH}} &= 300.8 \text{ BTU/lbm comb. gas} \\ &= \underline{1094.9 \text{ BTU/lbm comb. gas}} \end{aligned}$$

V ENERGY GENERATION

TURBINE HEAT RATE IS 7829.2 BTU/KWH REF 1
THUS FOR $1 \frac{\text{lbm}}{\text{hr}}$ FUEL

$$\frac{1094.9 \text{ BTU/lbm} \times 14.29 \text{ lbm/hr}}{7829.2 \text{ BTU/KWH} \times 1 \text{ lbm/hr}} = \underline{1.998 \text{ KW/lbm/hr}}$$

VI STATION HEAT RATE

AUXILIARY LOADS ARE ASSUMED TO BE 4.5% OF
TURBINE OUTPUT (EXCLUSIVE OF FORCED DRAFT
FANS)

$$\begin{aligned} 1.998 \text{ KW} \times .955 &= 1.908 \text{ KW} \\ - \text{FORCE DRAFT FAN} &= .014 \text{ KW} \\ \text{NET PER lbm FUEL} &= \underline{1.894 \text{ KW}} \end{aligned}$$

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$$\text{HEAT RATE} = \frac{\text{HEATING VALUE OF FUEL}}{\text{NET KW / lbm FUEL}} = \frac{17284 \frac{\text{BTU}}{\text{lbm}}}{1,894 \text{ KW/lbm}} =$$

$$\underline{\underline{9123.2 \text{ BTU/KWHR}}}$$

$$\text{EFFICIENCY} = \frac{3413}{9123.2} = \underline{\underline{37.4\%}}$$

VII 800 MWe (NET) STATION

FUEL REQUIRED

$$\frac{800 \text{ MWe}}{1,894 \text{ KW/lbm fuel}} = \underline{\underline{422386 \text{ lbm fuel/hr}}}$$

AIR REQUIRED

$$13.29 \text{ lbm AIR / lbm fuel} \times 422386 \frac{\text{lbm fuel}}{\text{hr}} =$$

$$\underline{\underline{5.613 \times 10^6 \frac{\text{lbm AIR}}{\text{Hr}}}}$$

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APPENDIX E

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CLIENT

THERMO ELECTRON CORP

J. O. NUMBER

12628.02

SUBJECT

CONVENTIONAL BOILER PERFORMANCE (WITH THERMIONICS)

PROBLEM:

DETERMINE THE NET CYCLE EFFICIENCY FOR A CURRENT TECHNOLOGY BOILER USING LIQUIFIED COAL AS A FUEL AND OPERATING ON A 2400 PSI/1000°F/1000°F STEAM CYCLE, AND EMPLOYING A THERMIONIC TOPPING CYCLE.

AUDIT
REMARKS

CHECKER'S REMARKS:

APPROACH/ASSUMPTIONS:

CALCULATION BASED ON ECONOMIZER OUTLET TEMP GIVEN IN REF. 2. THIS TEMP IS HIGHER THAN THAT USED IN CONVENTIONAL BOILERS BY ~50°F. USE OF THE LOWER ECONOMIZER OUTLET TEMP WOULD REQUIRE THE ADDITION OF MORE EVAPORATOR AREA (IE MORE PLATTENS) THIS WOULD SIGNIFICANTLY INCREASE PLANT COST BUT WOULD NOT SIGNIFICANTLY INCREASE EFFICIENCY.

AUDIT
REMARKS

CHECKER'S REMARKS:

SOURCES—DATA/EQUATIONS:

SEE REF. PG. 2

AUDIT
REMARKS

CHECKER'S REMARKS:

CONCLUSIONS:

NET CYCLE EFFICIENCY 41.24%

AUDIT
REMARKS

CHECKER'S REMARKS:

AUDIT SUMMARY		CALCULATED BY		DATE	
CHECKS	CORRECTIVE ACTION TAKEN BY	Brian J. Roberts		10/19/77	
INFRACTIONS	DATE			11/14/77	
		CHECKED BY		DATE	
		Richard J. Dube		11/14/77	
		AUDITED BY		DATE	

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REFERENCE:

1. APPENDIX D OF THIS REPORT
CALCULATION OF CONVENTIONAL BOILER PERFORMANCE.
2. 2400/1000/1000 STEAM CYCLE CALC. APPENDIX D
TO STONE & WEBSTER REPORT "REVIEW OF
THERMIONICS FOR POWER GENERATION" DEC 29, 1976
J.O. No. 12628
3. APPENDIX A OF THIS REPORT
CALCULATION OF EMITTER TEMP FOR VARIOUS
FLAME TEMPERATURES
4. THERMIONIC EFFICIENCY CHARTS - APPENDIX K TO
STONE & WEBSTER REPORT CITED IN 2. ABOVE

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APPROACH:

THE ANALYSIS OF THE ADDITION OF THERMIONICS TO A CONVENTIONAL BOILER DOES NOT LEND ITSELF TO THE SAME TECHNIQUE AS WAS USED FOR THE ADVANCE DESIGN BOILER SINCE WE MUST NOT ONLY DETERMINE THE AREA SPLIT REQUIRED BETWEEN EVAPORATOR AND SUPERHEATER/REHEATER BUT ALSO THE SPLIT WITHIN THE EVAPORATOR FIREBOX AND PLATTEN REGION.

A BOILER (SIZE TYPICAL OF MPOOMWE UNIT) WAS SELECTED AND THE WATERWALL AREA USABLE FOR THERMIONICS WAS DETERMINED AS FOLLOWS:

BOILER (TYPICAL)

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1. AREA IN FIREBOX USABLE FOR THERMIONICS
(1 WALL LOST DUE TO BURNERS)

$$82' \times 57' + 2 \times 44' \times 57' = 9690 \text{ ft}^2$$

2. WATERWALL AREA IN PLATTEN REGION USABLE
FOR THERMIONICS

$$2 \times 44' \times 53' + 53' \times 82' - 2 \times 15' \times 82' + 44' \times 82' = 10158 \text{ ft}^2$$

3. AREA IN FIREBOX WHICH WILL NOT HAVE THERMIONICS
BUT WHICH WILL ABSORB HEAT. (BURNER WALL)

$$57' \times 82' = 4674 \text{ ft}^2$$

4. WATERWALL AREA IN PLATTEN REGION THAT WILL NOT
HAVE THERMIONICS BUT WILL ABSORB HEAT.

$$2 \times 15' \times 82' + 0.5 \times 53' \times 82' = 4633 \text{ ft}^2$$

← SINCE THIS AREA IS BEHIND
SUPERHEATER IT IS ASSUMED
TO BE $\frac{1}{2}$ AS EFFECTIVE FOR
HEAT TRANSFER

BASED ON CALC REF 1 7829.2 BTU/hr MUST
BE DELIVERED TO THE STEAM CYCLE IN ORDER
TO GENERATE 1 KW OF ELECTRICITY (NET)
FOR 800000 KW 7829.2 BTU/hr $\times$ 800000 KW
= 6.263×10^9 BTU/hr MUST BE DELIVERED.
THE RATIO OF HEAT DELIVERED TO THE BOILER
(EVAPORATOR) TO THAT DELIVERED TO THE
SUPERHEATER/REHEATER IS 1.0940 (REF 2)
THUS

$$\Delta H_{\text{BOILER}} + \Delta H_{\text{SH}} = 6.263 \times 10^9$$

$$\Delta H_{\text{BOILER}} + \frac{1}{1.094} \Delta H_{\text{BOILER}} = 6.263 \times 10^9$$

$$\Delta H_{\text{BOILER}} = 3.272 \times 10^9 \text{ BTU/hr}$$

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Q/A PROFILE ADJUSTED TO AVE Q/A OF 70,800 BTU/HR FT²

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6 ASSUME THAT 57% OF ΔH_{boiler} IS ADDED IN
7 THE EVAPORATOR AND 43% IS ADDED IN ECONOMIZER.

9 THUS HEAT WHICH MUST BE ABSORBED IN THE BOILER
10 IS:

$$11 \quad 3.272 \times 10^9 \text{ BTU/hr} \times 0.57 = \underline{\underline{1.865 \times 10^9 \text{ BTU/hr}}}$$

14 THE HEAT ABSORBED IN THE FIXED AREA (WATERWALLS)
15 OF THE BOILER WILL NOW BE DETERMINED AND
16 THEN THE PLATTEN AREA REQUIREMENT WILL
17 BE DETERMINED.

20 I. THERMIONIC PERFORMANCE IN FIREBOX

22 $Q/A = 75500 \text{ BTU/hr}$ DETERMINE FLAME TEMP
24 PER REF 3

$$26 \quad \frac{T^4 (1.210)^4}{111} = \frac{75500}{70000}$$

$$30 \quad T^4 = 119.72 + (1.210)^4$$

$$32 \quad T = 3.322 \Rightarrow \text{TEMP } 3322^\circ \text{R}$$

$$33 \quad 2862^\circ \text{F}$$

35 WHICH GIVES EMITTER TEMP 1500°K
36 FROM REF 3.

$$38 \quad T_e = 1500^\circ \text{K}$$

$$39 \quad T_c = 670^\circ \text{K}$$

40 EMITTER HEAT FLUX 13.5 W/cm^2 (REF 3)

41 η_T 33% (SEE GRAPH ON FOLLOWING PAGE)

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II. HEAT TRANSFER TO FIRE BOX REGION WATERWALL

$$9690 \text{ ft}^2 \times 42795 \text{ BTU/hr ft}^2 \times (1 - .33) +$$

$$\uparrow 13.5 \text{ W/cm}^2$$

$$4674 \text{ ft}^2 \times 75500 \text{ BTU/hr ft}^2 = \underline{\underline{6.307 \times 10^8 \text{ BTU/hr}}}$$

III. THERMIONIC PERFORMANCE IN PLATTEN REGION

$$Q/A = 61200 \text{ BTU/hr ft}^2$$

DETERMINE FLAME TEMP PER REF 3

$$\frac{T^4 - (1210)^4}{111} = \frac{61200}{70000}$$

$$T^4 = 97.04 + 1210^4$$

$$T = 3155 \Rightarrow \text{TEMP } 3155^\circ \text{R}$$

$$2695^\circ \text{F}$$

WHICH GIVES EMITTER TEMP OF 1460°K
FROM REF. 3.

$$T_E = 1460^\circ \text{K}$$

$$T_C = 660^\circ \text{K}$$

EMITTER HEAT FLUX 10.5 W/cm^2

$$\eta_T = 31.5\% \text{ (SEE GRAPHS ON FOLLOWING PAGES)}$$

IV. HEAT TRANSFER TO PLATTEN REGION OF WATERWALL

$$10158 \text{ ft}^2 \times 33285 \text{ BTU/hr ft}^2 (1 - .315) +$$

$$\uparrow 10.5 \text{ W/cm}^2$$

$$4633 \text{ ft}^2 \times 61200 \text{ BTU/hr ft}^2 = \underline{\underline{5.151 \times 10^8 \text{ BTU/hr}}}$$

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EMITTER HEAT FLUX W/CM^2

BASED ON DATA IN REF 4

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$T_E = 1400^\circ K$

BASED ON DATA IN REF 4

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V. HEAT TRANSFER TO PLATTENS

$$\begin{array}{rcl} \text{TOTAL TO BOILER} & 18.65 \times 10^8 & \text{BTU/hr} \\ \text{TO FIRE BOX WATERWALL} & - 6.30 \times 10^8 & \text{"} \\ \text{TO PLATTEN REGION WATERWALL} & - 5.15 \times 10^8 & \text{"} \\ \hline & 7.2 \times 10^8 & \text{BTU/hr} \end{array}$$

HEAT TRANSFER PER FT² OF PLATTEN WITH THERMIONICS

$$33285 \text{ BTU/hr ft}^2 (1 - .315) = 22800 \text{ BTU/hr ft}^2$$

AREA REQUIRED

$$\frac{7.2 \times 10^8 \text{ BTU/hr}}{22800 \text{ BTU/hr ft}^2} = 31579 \text{ ft}^2$$

ASSUME EACH PLATTEN HAS ~ 1500 FT² (PER OSWEGO) THEN 21 PLATTENS WOULD BE REQUIRED.

VI THERMIONIC D.C. POWER GENERATION

FIREBOX REGION WATERWALL

$$9690 \text{ FT}^2 \times (30.48)^2 \frac{\text{cm}^2}{\text{FT}^2} \times 13.5 \frac{\text{W}}{\text{cm}^2} \times .33 = 40,105,267 \text{ WATTS}$$

PLATTEN REGION

$$(10158 \text{ FT}^2 + 31579 \text{ FT}^2) \times (30.48)^2 \frac{\text{cm}^2}{\text{FT}^2} \times 10.5 \frac{\text{W}}{\text{cm}^2} \times .315 = 128,248,120 \text{ WATTS}$$

$$\text{TOTAL} \quad \underline{\underline{168,353,387 \text{ WATTS}}}$$

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VII FUEL REQUIREMENTS

STEAM CYCLE
THERMIONICS

$$6.263 \times 10^9 \text{ BTU/hr}$$

$$5.746 \times 10^8 \text{ BTU/hr}$$

TOTAL

$$6.837 \times 10^9 \text{ BTU/hr}$$

FROM REF 1 1bm OF COMBUSTION GAS WILL
GIVE UP 1094.9 BTU AND EACH 1bm FUEL
FORMS 14.29 1bm COMBUSTION GAS.

$$\frac{6.837 \times 10^9 \text{ BTU/hr}}{1094.9 \text{ BTU} \times \frac{14.29}{1\text{bm}}} = \underline{\underline{436,977 \text{ 1bm fuel/hr}}}$$

VIII STATION HEAT RATE

800MW TURBINE GROSS BY INITIAL PROBLEM
DEFINITION

AUX LOAD 4.5% OF TURBINE GROSS (EXCLUSIVE
OF FORCED DRAFT FAN)

FORCED DRAFT FAN 0.014 KW PER 1bm FUEL
(PER REF 1)

THERMIONICS 168,353,387 WATTS DC
DC TO AC CONVERSION 92% EFFICIENT

NET GENERATION =

$$800,000 \text{ KW} \times 0.955 - 436,977 \frac{\text{1bm fuel}}{\text{hr}} \times 0.014 \text{ KW} + 168,353 \text{ KW} \times 0.92$$

$$= \underline{\underline{912,767 \text{ KW NET}}}$$

$$\text{STATION HEAT RATE} = \frac{\text{HEAT VALUE OF FUEL} \times \text{1bm FUEL/HR}}{\text{NET KW GENERATED}} =$$

$$\frac{17,284 \frac{\text{BTU}}{1\text{bm}} \times 436,977 \frac{\text{1bm}}{\text{hr}}}{912,767 \text{ KW}} = \underline{\underline{8,274.5 \text{ BTU/KW hr}}}$$

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$$\text{EFFICIENCY} = \frac{3413}{8274.5} = \underline{\underline{41.24\%}}$$

II INCREASED SURFACE AREA REQUIRED.

HEAT TRANSFER REQUIRED $1.865 \times 10^9 \text{ BTU/hr}$
AVE Q/A NO THERMIONICS $70000 \text{ BTU/hr ft}^2$ (pg 5-12)

AREA WITH NO THERMIONICS

$$\frac{1.865 \times 10^9}{70000} = 26643 \text{ ft}^2$$

AREA WITH THERMIONICS

$$26643 \text{ ft}^2 + 31579 \text{ ft}^2 = 58222 \text{ ft}^2$$

c 21 PLATTENS

OR 218% OF AREA WITH NO THERMIONICS

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APPENDIX F

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CLIENT

THERMO ELECTRON CORP

J. O. NUMBER

12628.02

SUBJECT

ADVANCED THERMIONIC BOILER PERFORMANCE

PROBLEM:

DETERMINE THE NET CYCLE EFFICIENCY FOR AN
ADVANCED DESIGN THERMIONIC BOILER, ALSO
ESTABLISH FRACTION OF POWER GENERATED BY
TURBINE & THERMIONICS

AUDIT
REMARKS

CHECKER'S REMARKS:

APPROACH/ASSUMPTIONS:

1. SEE ASSUMPTIONS PG 4.
2. ALL THERMIONIC EFFICIENCIES USED ARE BASED ON
GRAPHICAL INTERPOLATION OF DATA IN REF 4

AUDIT
REMARKS

CHECKER'S REMARKS:

SOURCES—DATA/EQUATIONS:

SEE PG 2

AUDIT
REMARKS

CHECKER'S REMARKS:

CONCLUSIONS:

STATION HEAT RATE 7234.8 BTU/KW HR $\eta_T = 47.17\%$
FOR 1 lbm fuel TURBINE GENERATES 1.565 KW GROSS
THERMIONICS " 0.924 KW

AUDIT
REMARKS

CHECKER'S REMARKS:

AUDIT SUMMARY

CHECKS

CORRECTIVE ACTION TAKEN
BY

CALCULATED BY

Brian J. Doherty

DATE

10/18/77

CHECKED BY

Richard J. Doherty

DATE

11/14/77

INFRACTIONS

DATE

AUDITED BY

DATE

Client THERMO ELECTRON CALC

Location

Est. No.

J.O. No. 12628.02

Subject

Date 10/18/77

By BTD

Checked 11/14/77

By R/D

Based on

Revised

By

REFERENCES:

1. GAS TABLES, KEENAN & KAYES 1948
2. APPENDIX A OF THIS REPORT
CALCULATION OF EMITTER TEMP. FOR VARIOUS
FLAME TEMPS.
3. APPENDIX C OF THIS REPORT
CALCULATION OF COLLECTOR TEMP.
4. THERMIONIC EFFICIENCY CHARTS APPENDIX K
TO STONE & WEBSTER "REVIEW OF
THERMIONICS FOR POWER GENERATION"
DEC. 29, 1976, J.O. No 12628
5. APPENDIX B OF THIS REPORT
CALCULATION OF THEORETICAL AIR
6. THE SCR II PROCESS - 3RD ANNUAL INTERNATIONAL
CONF. ON COAL GASIFICATION & LIQUEFACTION,
B. K. SCHMIDT & D. M. JACKSON GULF MINERAL
RESOURCES CO. DENVER COLO.
7. 2400/1000/1000 STEAM CYCLE CALC.
APPENDIX D TO STONE & WEBSTER
REPORT CITED IN #4 ABOVE.

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FURNACE CALCULATION:

ASSUMPTIONS:

1. CALCULATION BASED ON 1 lbm/hr OF FUEL AND 105% THEORETICAL AIR.
2. MOISTURE LOSSES EXCEPT FOR LATENT HEAT OF H₂O IN FUEL ARE NEGLECTED
3. ENTHALPY OF FUEL AT AMBIENT TEMP IS NEGLECTED
4. ENTHALPY OF COMBUSTION GAS IS ASSUMED EQUAL TO ENTHALPY OF AIR. ALL AIR ENTHALPIES ARE FROM REF. 1 TABLE 1
5. AMBIENT AIR AT 60°F & 60% RELATIVE HUMIDITY.

I. MOISTURE LOSS

FROM REFS THERE ARE 3.95 MOLES OF H₂O FORMED PER 100 lbm OF FUEL.

THE LATENT HEAT OF THIS WATER IS

$$3.95 \frac{\text{MOLES H}_2\text{O}}{100 \text{ lbm fuel}} \times 18 \frac{\text{lbm}}{\text{MOLE}} \times 1040 \frac{\text{BTU}}{\text{lbm}} = 73944 \frac{\text{BTU}}{100 \text{ lbm fuel}}$$

$$\text{OR } 739.4 \frac{\text{BTU}}{\text{lbm fuel}}$$

II. ENTHALPY OF COMBUSTION GAS

FROM REFS 12.66 lbm OF WET AIR ARE REQUIRED FOR EACH lbm OF FUEL. 105% TA = 13.29 lbm

(REF 1)	h _{air} AT 800°F	13.29 lbm X 306.6 $\frac{\text{BTU}}{\text{lbm}}$	4075.6 BTU
	h _{combustion}	1 lbm X (17284 - 739.4) $\frac{\text{BTU}}{\text{lbm}}$	16544.6 BTU
(REF 6)	HHV LIQUIFIED COAL	17284 $\frac{\text{BTU}}{\text{lbm}}$	20620.2 BTU

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$$\frac{20,620.2 \text{ BTU}}{(1 + 13.29) \text{ lbm}} = \underline{\underline{1442.9 \text{ BTU/lbm}}}$$

COMB. GAS PER lbm FUEL

III. THERMIONIC AIR PREHEATERS

HEAT IS REJECTED VIA HEAT PIPES FROM THERMIONIC CONVERTERS LOCATED IN THE 3200°F ISOTHERMAL FURNACE SECTION. THIS REJECTED HEAT, HEATS COMBUSTION AIR FROM 600°F TO 800°F

$$h_{\text{AIR AT 800°F}} = 306.6 \text{ BTU/lbm}$$

$$h_{\text{AIR AT 600°F}} = 255.9 \text{ BTU/lbm}$$

$$\Delta h = 50.7 \text{ BTU/lbm}$$

FOR 1 lbm FUEL 13.29 lbm AIR ARE REQUIRED

$$\Delta H = 50.7 \text{ BTU/lbm} \times 13.29 \text{ lbm} = \underline{\underline{673.8 \text{ BTU}}}$$

THE THERMIONIC CONVERTERS IN THIS REGION OPERATE AT

$$T_e = 1600^\circ\text{K}$$

$$T_c = 700^\circ\text{K}$$

$$\eta = 34.5\%$$

$$\text{HEAT FLUX TO EMITTER } 22 \text{ W/cm}^2$$

ENERGY LOST BY COMBUSTION GASES TO THERMIONIC AIR PREHEATERS

$$\frac{673.8 \text{ BTU}}{(1 - 0.345)} = 1028.7 \text{ BTU}$$

$$\text{OR } \frac{1028.7 \text{ BTU}}{14.29 \text{ lbm}} = \underline{\underline{72 \text{ BTU/lbm COMB GAS}}}$$

OFF THIS 24.8 BTU/lbm IS CONVERTED TO ELECTRICITY BY THE THERMIONIC DEVICE.

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$$\text{OR } \frac{24.8 \text{ BTU/hr}}{3413 \text{ BTU/KW}} = 7.27 \times 10^{-3} \text{ KW/lbm OF COMB GAS}$$

IV. WATERWALL THERMIONICS IN THE 3200°F ISOTHERMAL REGION.

ENERGY AVAILABLE

$$\begin{array}{ccccccc} 1442.9 \frac{\text{BTU}}{\text{lbm}} & - & 72 \frac{\text{BTU}}{\text{lbm}} & - & 986 \frac{\text{BTU}}{\text{lbm}} & = & 384.9 \frac{\text{BTU}}{\text{lbm}} \\ h_{\text{comb}} & & \text{TO AIR PREHEAT} & & h_{\text{GAS } 3200^{\circ}\text{F}} & & \text{COMB. GAS} \end{array}$$

THERMIONIC PERFORMANCE

$$T_e = 1600^{\circ}\text{K}$$

$$T_c = 670^{\circ}\text{K}$$

$$\eta_T = 36\%$$

$$\text{HEAT FLUX TO EMITTER } 22 \text{ W/cm}^2$$

$$\text{THUS } 384.9 \frac{\text{BTU}}{\text{lbm}} \times .36 = 138.56 \frac{\text{BTU}}{\text{lbm}} \text{ IS CONVERTED TO ELECTRICITY } (4.06 \times 10^{-2} \text{ KW/lbm COMB GAS})$$

$$\text{AND } 384.9 \frac{\text{BTU}}{\text{lbm}} \times (1 - .36) = 246.3 \frac{\text{BTU}}{\text{lbm COMB GAS}} \text{ IS DELIVERED TO THE STEAM CYCLE}$$

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Based on Revised By

II THERMIONIC PERFORMANCE IN REGION OF FURNACE WHERE GAS COOLS FROM 3200°F TO 3000°F (TAVE 3100°F)

ENERGY TRANSFER TO EMITTER / WATERWALL

$$h_{GAS\ 3200} = 986\ BTU/lbm\ C.G.$$

$$h_{GAS\ 3000} = 926.5\ BTU/lbm\ C.G.$$

$$\Delta h = 59.5\ BTU/lbm\ C.G.$$

$$T_E = 1575^\circ K\ (REF\ 2)$$

$$T_C = 670^\circ K$$

$$\eta_T = 35.3\% \quad \text{HEAT FLUX TO EMITTER } 19.5\ W/cm^2$$

$$\begin{aligned} \text{THERMIONIC GENERATION } 59.5\ BTU/hr \times .353 &= 21\ BTU/hr \\ &= 6.15 \times 10^{-3}\ KW/lbm\ C.G. \end{aligned}$$

$$\text{HEAT TO STEAM CYCLE } 59.5\ BTU/hr \times (1 - .353) = 38.5\ BTU/hr$$

III THERMIONIC PERFORMANCE IN REGION OF FURNACE WHERE GAS COOLS FROM 3000°F TO 2800°F (TAVE = 2900°F)

ENERGY TRANSFER TO EMITTER / WATERWALL

$$h_{GAS\ 3000} = 926.5\ BTU/lbm\ C.G.$$

$$h_{GAS\ 2800} = 867\ BTU/lbm\ C.G.$$

$$\Delta h = 59.5\ BTU/lbm\ C.G.$$

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1 Client THERMO ELECTRON CORP Location _____ Est. No. _____ J.O. No. 12628.02
2 Subject _____ Date 10/18/77 By BJD
3 _____ Checked 11/14/77 By KJD
4 Based on _____ Revised _____ By _____

$T_E = 1525^\circ K$ (REF 2)

$T_C = 670^\circ K$

$\eta_T = 33\%$ HEAT FLUX TO EMITTER $15 W/cm^2$

THERMIONIC GENERATION $59.5 \text{ BTU/hr} \times .33 = 19.6 \text{ BTU/hr} =$
 $5.74 \times 10^{-3} \text{ KW/lbm C.G.}$

HEAT TO STEAM CYCLE $59.5 \text{ BTU/hr} \times (1-.33) = \underline{39.8 \text{ BTU/hr}}$

VII THERMIONIC PERFORMANCE IN REGION OF FURNACE
WHERE GAS COOLS FROM $2800^\circ F$ TO $2600^\circ F$
(TAKE = $2700^\circ F$)

ENERGY TRANSFER TO EMITTER/WATERWALL

$h_{\text{GAS } 2800} = 867 \text{ BTU/lbm C.G.}$

$h_{\text{GAS } 2600} = 808 \text{ BTU/lbm C.G.}$

$\Delta h = 59 \text{ BTU/lbm C.G.}$

$T_E = 1450^\circ K$ (REF 2)

$T_C = 660^\circ K$

$\eta_T = 32\%$ HEAT FLUX TO EMITTER $11.8 W/cm^2$

THERMIONIC GENERATION $59 \text{ BTU/hr} \times .32 = 18.9 \text{ BTU/hr}$
 $5.53 \times 10^{-3} \text{ KW/lbm COMB GAS}$

HEAT TO STEAM CYCLE $59 \text{ BTU/hr} \times (1-.32) = \underline{40.1 \text{ BTU/hr}}$

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VIII. THERMIONIC PERFORMANCE IN REGION OF FURNACE
WHERE GAS COOLS FROM 2600°F TO 2400°F (TAIR 2500°F)

ENERGY TRANSFER TO EMITTER/WATERWALL

$$h_{\text{GAS } 2600} = 808 \text{ BTU/lbmCG}$$

$$h_{\text{GAS } 2400} = 749.8 \text{ BTU/lbmCG}$$

$$\Delta h = 58.2 \text{ BTU/lbmCG}$$

$$T_e = 1400^\circ\text{K}$$

$$T_c = 660^\circ\text{K}$$

$$\eta_f = 29.5\% \quad \text{HEAT FLUX TO EMITTER } 7.5 \text{ W/cm}^2$$

THERMIONIC GENERATION $58.2 \text{ BTU/hr} \times .295 = 17.2 \text{ BTU/hr}$

$$5.04 \times 10^{-3} \text{ KW/lbm CGAS}$$

HEAT TO STEAM CYCLE $58.2 \text{ BTU/hr} \times (1 - .295) = 41 \text{ BTU/hr}$

IX. ENTHALPY OF GAS LEAVING SUPERHEATER

THIS ENTHALPY IS SET BY LIMITING FLUE
GAS TEMP TO 300°F AND BY PREHEATING
AIR WITH COMB. GAS TO 600°F

ASSUME AIR IS COMPRESSED TO 1.2 ATM
FOR 60°F AIR $h = 124.2 \frac{\text{BTU}}{\text{lbm}}$ $P_r = 1.2122$

at 1.2 ATM $P_r = 1.2122 \times 1.2 = 1.4547 \Rightarrow h = 130.86 \frac{\text{BTU}}{\text{lbm}}$

$$\Delta h = 6.66 \text{ BTU/lbm} \quad \text{HOWEVER FAN IS NOT 100\% EFF}$$

ASSUME 90% EFF $\Delta h_{\text{ACTUAL}} = \frac{6.66}{0.9} = 7.40 \text{ BTU/lbm}$

$$h_{\text{OUT OF FAN}} = 124.2 + 7.40 = 131.6 \frac{\text{BTU}}{\text{lbm}} \Rightarrow T = 91^\circ\text{F}$$

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 Based on _____ Revised _____ By _____

ENTHALPY WHICH MUST BE ADDED TO AIR RAISING ITS
TEMP FROM 91°F TO 600°F

$$h_{\text{AIR } 600^{\circ}\text{F}} \quad 255.9 \text{ BTU/lbm}$$

$$h_{\text{AIR } 91^{\circ}\text{F}} \quad 131.6 \text{ BTU/lbm}$$

$$\Delta h = 124.3 \text{ BTU/lbm AIR}$$

SINCE 14.29 lbm COMBUSTION GAS ARE FORMED FROM
13.29 lbm AIR THE Δh OF THE COMBUSTION GAS
IS

$$124.3 \text{ BTU/lbm} \times \frac{13.29}{14.29} = 115.6 \text{ BTU/lbm COMB GAS}$$

THUS ENTHALPY OF GAS LEAVING SUPERHEATER
MUST BE

$$h_{\text{GAS } 300} \quad 182.0 \text{ BTU/lbm}$$

$$+ \Delta h_{\text{AIR PREHEAT}} \quad 115.6 \text{ BTU/lbm}$$

$$\underline{\underline{297.6 \text{ BTU/lbm}}}$$

NOTE: FOR 1 lbm FUEL $\frac{7.40 \text{ BTU/lbm} \times 13.29 \text{ lbm AIR}}{3413 \text{ BTU/KW} \times 0.95} = 3.03 \times 10^{-2} \text{ KW}$
 ↑ MOTOR EFFICIENCY

ENERGY MUST BE SUPPLIED TO THE FORCE DRAFT FAN.

X ENERGY ADDITION IN BOILER VS. ENERGY ADDITION
IN SUPERHEATER REHEATER.

INITIAL EVALUATION INDICATES THAT EVEN
THOUGHT ADDITION OF MORE THERMIONIC SURFACE
IN THE BOILER (EVAPORATOR) WOULD INCREASE
THE CYCLE EFFICIENCY, IT WOULD NOT BE
ECONOMICALLY ATTRACTIVE, THUS WE MUST
DETERMINE HOW MUCH ADDITIONAL WATERWALL
MUST BE ADDED IN ORDER TO OBTAIN THE
PROPER SPLIT OF ENERGY ABSORPTION BETWEEN

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1 Client THERMO ELECTRON CORP Location _____ Est. No. _____ J.O. No. 12628.02
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 4 Based on _____ Revised _____ By _____

THE BOILER AND SUPERHEATER/REHEATER,

FROM CALCULATIONS PERFORMED IN REF. 7 THE
 SPLIT SHOULD BE $\frac{\Delta H_{\text{BOILER}}}{\Delta H_{\text{SHRH}}} = 1.0940$ FOR A

2400 PSIG/1000°F/1000°F CYCLE WITH TURBINE HEAT
 RATE = 7829.2 BTU/KWH

FROM THE CALC. IN SECTION IV THROUGH VIII
 IT CAN BE SEEN THAT 405.7 BTU HAVE
 BEEN ADDED TO THE BOILER BY THE TIME THE
 COMB. GAS HAS COOLED TO 2400°F ($h = 749.8 \text{ BTU/lbm}$)

$$\frac{\Delta H_{\text{BOILER}}}{\Delta H_{\text{SHRH}}} = \frac{405.7 + X}{(749.8 - X) - 297.6} = 1.0940$$

$\uparrow$ h OF GAS LEAVING
 SHRH

$$X = 42.5 \text{ BTU/lbm}$$

ENTHALPY OF COMB GAS LEAVING BOILER →
 ENTERING SUPERHEATER/REHEATER IS $707.3 \frac{\text{BTU}}{\text{lbm}}$

$$\Rightarrow T = 2345^\circ\text{F}$$

AND 409.7 BTU/lbm COMB GAS ARE ADDED TO
 THE STEAM CYCLE IN THE SH/RH.

XI ENERGY GENERATION

THE THERMIONIC CONVERTER ELECTRICAL GENERATION
 IS TABULATED BELOW ALONG WITH THE
 THERMAL ENERGY DELIVERED TO THE
 STEAM CYCLE. THE TABLE IS BASED ON
 UTILIZATION OF 1 lbm COMBUSTION GAS.

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Client THERMO ELECTRON CORP Location

Est. No.

J.O. No. 1269.02

Subject

Date 10/18/77

By BJD

Checked 11/14/77

By RGS

Based on

Revised

By

FURNACE
SECTIONTHERMIONIC
GENERATION
KW DCHEAT TO
STEAM CYCLE
BTU/HRTHERMIONIC
AIR PRE HEAT 7.27×10^{-3}

0

3200°F
ISOTHERMAL 40.6×10^{-3}

246.3

3200°F -
3000°F 6.15×10^{-3}

38.5

3000°F -
2800°F 5.75×10^{-3}

39.5

2800°F -
2600°F 5.53×10^{-3}

40.1

2600°F -
2400°F 5.04×10^{-3}

41

2400°F -
2345°F

0

42.5

SUPERHEAT
+ REHEAT

0

409.7

TOTAL/
1lbm comb GAS 70.34×10^{-3} KW857.6 $\frac{\text{BTU}}{\text{hr}}$

CONVERTING TO 1 lbm FUEL REFERENCE, 14.29 lbm comb GAS
ARE EQUIVALENT TO 1 lbm FUEL; THUS

THERMIONIC GENERATION 1 lbm FUEL

$$70.34 \times 10^{-3} \text{ KW/lbm comb GAS} \times 14.29 \text{ lbm} = 1.005 \text{ KW DC}$$

DC-AC CONVERSION EFFICIENCY IS 92%

$$\text{THERMIONIC AC GENERATION IS } 1.005 \text{ KW} \times 92 = 924 \text{ KW AC}$$

1 Client THERMO ELECTRON CORP Location _____ Est. No. _____ J.O. No. 12628.02
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THE TURBINE HEAT RATE IS 7829.2 BTU/KW HR
 THUS FOR 1 lbm FUEL/hr

$$\frac{857.6 \frac{\text{BTU}}{\text{lbm}} \times 14.29 \text{ lbm/hr}}{7829.2 \text{ BTU/KW HR}} = 1.565 \text{ KW/lbm/hr}$$

FROM TURBINE

XII STATION HEAT RATE

AUXILIARY LOADS ARE ASSUMED TO BE 4.5%
 OF TURBINE OUTPUT (EXCLUSIVE OF FORCED
 DRAFT FANS)

$$\begin{aligned} 1.565 \text{ KW} \times .955 &= 1.495 \text{ KW} \\ + \text{ THERMIONIC GENERATION} &.924 \text{ KW} \\ - \text{ FORCED DRAFT FAN} &.030 \text{ KW} \\ \hline \text{NET PER 1 lbm FUEL} &2.389 \text{ KW} \end{aligned}$$

$$\text{HEAT RATE} = \frac{\text{HEATING VALUE OF FUEL}}{\text{NET KW / 1 lbm FUEL}} =$$

$$\frac{17284 \text{ BTU/lbm} \times 1 \text{ lbm/hr}}{2.389 \text{ KW}} = 7234.8 \frac{\text{BTU}}{\text{KW HR}}$$

$$\text{EFFICIENCY} = \frac{3413}{7234.8} = 47.17\%$$

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2 Subject _____ Date 10/18/77 By BJD
3 _____ Checked 11/4/77 By BJD
4 Based on _____ Revised _____ By _____

XIII 800 MWe (NET) STATION

$$\frac{800 \text{ MWe}}{2.389 \text{ KW/lbm fuel}} = 334868 \text{ lbm fuel/hr}$$

$$\Rightarrow 5.787 \times 10^9 \text{ BTU/hr FIRED}$$

TURBINE OUTPUT

$$334868 \text{ lbm fuel} \times 1.565 \text{ KW/lbm fuel} = \underline{\underline{524 \text{ MWe GROSS}}}$$

THERMIONICS

$$334868 \text{ lbm fuel} \times 0.924 \text{ KW/lbm fuel} = \underline{\underline{309 \text{ MWe}}}$$

AIR REQUIRED

$$\frac{13.29 \text{ lbm AIR}}{\text{lbm fuel}} \times 334868 \frac{\text{lbm fuel}}{\text{hr}} = \underline{\underline{4.450 \times 10^6 \frac{\text{lbm AIR}}{\text{hr}}}}$$

CALCULATION SUMMARY

▲ 5010.47

CALCULATION NO.

APPENDIX G

DATE

10/25/77

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CLIENT

THERMO ELECTRON CORP

J. O. NUMBER

12628.02

SUBJECT

ADVANCED BOILER SIZING

PROBLEM:

DETERMINE THE SIZE OF THE THERMIONIC WATERWALL
AND AIR PREHEAT PANELS FOR USE IN BOILER
COST ESTIMATES

AUDIT
REMARKS

CHECKER'S REMARKS:

APPROACH/ASSUMPTIONS:

1. BASED ON BOILER PERFORMANCE CALC OF APPENDIX F
2. SEE ASSUMPTIONS PG 2

AUDIT
REMARKS

CHECKER'S REMARKS:

SOURCES—DATA/EQUATIONS:

REF 1. APPENDIX F

AUDIT
REMARKS

CHECKER'S REMARKS:

CONCLUSIONS:

SEE PG 8

AUDIT
REMARKS

CHECKER'S REMARKS:

AUDIT SUMMARY

CHECKS

CORRECTIVE ACTION TAKEN
BY

CALCULATED BY

Brian J. Liberty

DATE

10/25/77

CHECKED BY

Richard J. Dube

DATE

11/14/77

INFRACTIONS

DATE

AUDITED BY

DATE

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1 Client THERMO ELECTRON CORP Location Est. No. J.O. No. 12628.02
2 Subject Date 10/25/77 By BJD
3 Checked 11/14/77 By RJD
4 Based on Revised By

ASSUMPTIONS:

1. BOILER MODULES WILL BE AT LEAST 8 FT. LONG IN ORDER TO PREVENT FLAME IMPINGEMENT ON THE WALLS. (4 FT CLEARANCE REQUIRED ON EACH SIDE OF 50×10^6 BTU/Hr BURNER)
2. BOILER WILL BE APPROX 30 FT WIDE BECAUSE FLAME FROM 50×10^6 BTU/Hr BURNER IS ~ 15 FT LONG AND BOILER IS FRONT & BACK FIRED.
3. WATERWALLS WILL BE FABRICATED IN PANELS $20' \times 8'$ TO FACILITATE HANDLING AND SHIPPING. EACH PANEL WILL HAVE AN EVEN NUMBER OF TUBES TO FACILITATE ELECTRICAL CONNECTIONS BETWEEN THERMIONIC DEVICES.
4. 95% OF WATERWALL AREA IS AVAILABLE FOR HEAT TRANSFER AND 5% WILL BE DEVOTED TO ELECTRICAL CONNECTORS.
5. BOILER TUBES WILL BE RECTANGULAR AS ILLUSTRATED

1 Client THERMO ELECTRON CORP Location _____ Est. No. _____ J.O. No. 1262802
2 Subject _____ Date 10/25/77 By BJD
3 _____ Checked 11/14/77 By R98
4 Based on _____ Revised _____ By _____

I. DETERMINE THE SURFACE AREA AVAILABLE FOR HEAT TRANSFER PER FOOT OF WATERWALL HEIGHT

THE ILLUSTRATED ARRANGEMENT WAS SELECTED FOR THE WATER WALL PANELS BASED ON THE CONSTRAINTS DESCRIBED IN "ASSUMPTIONS"

THE WATER WALL WILL BE 4 PANELS WIDE TO GIVE THE ~30' BOILER DIMENSION DESIRED.

HEAT TRANSFER AREA PER FOOT (HEIGHT) OF WATERWALL

$$= 95\% (7.86' \times 4) = \underline{\underline{29.87 ft^2}}$$

$$\text{OR } \underline{\underline{27748 cm^2}}$$

Client THERMO ELECTRON CORP Location Est. No. J.O. No. 12628.02
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II. DETERMINE THE SURFACE AREA AVAILABLE FOR HEAT TRANSFER PER FOOT OF HEIGHT OF FURNACE WALL COVERED BY THERMIONIC AIR PREHEATERS.

THE ILLUSTRATED ARRANGEMENT WAS SELECTED FOR THE THERMIONIC AIR PREHEATERS WHICH ARE ASSUMED TO BE LOCATED ON THE FRONT AND BACK BOILER WALLS ALONG WITH THE BURNERS

ASSUME 78% OF AREA IS USABLE FOR HEAT TRANSFER.

HEAT TRANSFER AREA PER FOOT (HEIGHT) OF AIR PREHEATER

$$= 78\% (8' \times 1') = \underline{\underline{6.24 ft^2}}$$

$$\text{or } \underline{\underline{5792 cm^2}}$$

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III HEIGHT OF THE 3200°F ISOTHERMAL REGION OF BOILER (BASED ON DATA IN REF 1)

1. TOTAL HEAT TRANSFER = $456.9 \frac{\text{BTU}}{\text{lbm comb gas}} \times 334868 \frac{\text{lbm fuel}}{\text{hr}}$
 $\times 14.29 \frac{\text{lbm comb gas}}{\text{lbm fuel}} = 2.186 \times 10^9 \text{ BTU/hr}$

2. HEAT FLUX = $22.4 \text{ W/cm}^2 \times 3170 \frac{\text{BTU/hr ft}^2}{\text{W/cm}^2} = 69740 \frac{\text{BTU}}{\text{hr ft}^2}$

3. HEAT TRANSFER AREA PER FOOT OF BOILER HEIGHT

$2 \times A_{\text{WATERWALL}} + 2 \times A_{\text{BURNER WALL}} =$
 $2 \times 29.87 \text{ ft}^2 + 2 \times 6.24 \text{ ft}^2 = 72.22 \text{ ft}^2/\text{ft}$

4. BOILER WATERWALL TOTAL COMBINED HEIGHT

$\frac{2.186 \times 10^9 \text{ BTU/hr}}{69740 \frac{\text{BTU}}{\text{hr ft}^2} \times 72.22 \text{ ft}^2/\text{ft}} = 434 \text{ ft}$

USING 8 BOILER MODULES THE HEIGHT OF THE 3200°F ISOTHERMAL REGION IS

$\frac{434}{8} = 54.2 \text{ ft}$

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IV HEIGHT OF 3200°F - 3000°F REGION OF BOILER

1. TOTAL HEAT TRANSFER

$$59.5 \frac{\text{BTU}}{\text{lbm Com Gas}} \times 334868 \frac{\text{lbm fuel}}{\text{hr}} \times 14.29 \frac{\text{lbm Com Gas}}{\text{lbm fuel}} =$$

$$\underline{2.847 \times 10^8 \frac{\text{BTU}}{\text{hr}}}$$

2. HEAT FLUX

$$19.5 \frac{\text{W}}{\text{cm}^2} \times 3170 \frac{\text{BTU/hr-ft}^2}{\text{W/cm}^2} = \underline{61815 \frac{\text{BTU}}{\text{hr-ft}^2}}$$

3. HEAT TRANSFER AREA PER FOOT OF BOILER HEIGHT

$$2 \times A_{\text{WATERWALL}} = 2 \times 29.87 \text{ ft}^2 = \underline{59.74 \text{ ft}^2/\text{ft}}$$

4. BOILER HEIGHT

$$\frac{2.847 \times 10^8 \frac{\text{BTU}}{\text{hr}}}{61815 \frac{\text{BTU}}{\text{hr-ft}^2} \times 59.74 \text{ ft}^2/\text{ft}} = \underline{77.09 \text{ ft}}$$

WITH 8 MODULES $H = \underline{\underline{9.6 \text{ ft}}}$

V HEIGHT OF 3000°F - 2800°F REGION OF BOILER

1. TOTAL HEAT TRANSFER

$$59.5 \frac{\text{BTU}}{\text{lbm Com Gas}} \times 334868 \frac{\text{lbm fuel}}{\text{hr}} \times 14.29 \frac{\text{lbm Com Gas}}{\text{lbm fuel}} =$$

$$\underline{2.847 \times 10^8 \text{ BTU/hr}}$$

2. HEAT FLUX

$$15 \frac{\text{W}}{\text{cm}^2} \times 3170 \frac{\text{BTU/hr-ft}^2}{\text{W/cm}^2} = \underline{47550 \frac{\text{BTU}}{\text{hr-ft}^2}}$$

Client THERMO ELECTRON CORP Location Est. No. J.O. No. 12628.02
Subject Date 10/25/77 By BJD
Checked 11/14/77 By RQE
Based on Revised By

3. HEAT TRANSFER AREA 59.74 ft²/ft

4. BOILER HEIGHT

$$\frac{2.847 \times 10^8 \text{ BTU/hr}}{47550 \frac{\text{BTU}}{\text{hr ft}^2} \times 59.74 \text{ ft}^2/\text{ft}} = 100.2 \text{ ft}$$

WITH 8 MODULES H = 12.5 ft

VI HEIGHT OF 2800°F - 2600°F REGION OF BOILER

1. TOTAL HEAT TRANSFER

$$59 \frac{\text{BTU}}{\text{lbm COMB GAS}} \times 334868 \frac{\text{lbm fuel}}{\text{hr}} \times 14.29 \frac{\text{lbm COMB GAS}}{\text{lbm fuel}} =$$

$$2.823 \times 10^8 \text{ BTU/hr}$$

$$2. \text{ HEAT FLUX} = 11.8 \text{ W/cm}^2 \times 3170 = 37406 \frac{\text{BTU}}{\text{hr ft}^2}$$

3. HEAT TRANSFER AREA 59.74 ft²/ft

4. HEIGHT OF BOILER

$$\frac{2.823 \times 10^8 \text{ BTU/hr}}{37406 \frac{\text{BTU}}{\text{hr ft}^2} \times 59.74 \text{ ft}^2/\text{ft}} = 126.3 \text{ ft}$$

WITH 8 MODULES H = 15.8 ft

Client THERMO ELECTRON CORP Location _____ Est. No. _____ J.O. No. 12628.02
Subject _____ Date 10/25/77 By BJD
Checked 11/14/77 By RJD
Based on _____ Revised _____ By _____

VII. HEIGHT OF 2600°F - 2400°F REGION OF BOILER

1. TOTAL HEAT TRANSFER

$$58.2 \frac{\text{BTU}}{\text{lbm Com Gas}} \times 334868 \frac{\text{lbm fuel}}{\text{hr}} \times 14.29 \frac{\text{lbm Com Gas}}{\text{lbm fuel}} = 2.785 \times 10^8 \text{ BTU/hr}$$

2. HEAT FLUX $7.5 \text{ W/cm}^2 \times 3170 = 23775 \frac{\text{BTU}}{\text{hr ft}^2}$

3. HEAT TRANSFER AREA $59.74 \text{ ft}^2/\text{ft}$

4. HEIGHT OF BOILER

$$\frac{2.785 \times 10^8 \text{ BTU/hr}}{23775 \frac{\text{BTU}}{\text{hr ft}^2} \cdot 59.74 \text{ ft}^2/\text{ft}} = 196 \text{ ft}$$

WITH 8 MODULES $H = \underline{\underline{24.5 \text{ ft}}}$

VIII. SIZE OF THERMIONIC WATERWALL

HEIGHT = $54.2 + 9.6 + 12.5 + 15.8 + 24.5 = \underline{\underline{116.6 \text{ FT}}}$

TOP ROW 217'
VICE 20'

1	2	3	4
5	6	7	8
9	10	11	12
13	14	15	16
17	18	19	20
21	22	23	24

116.6'

31.4'

CALCULATION SUMMARY

▲ 5010.47

CALCULATION NO.

APPENDIX H

DATE

10/17/77

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CLIENT

THERMO ELECTRON CORP.

J. O. NUMBER

12628.02

SUBJECT

LIQUIFIED COAL LEVELIZED FUEL COST AND CAPITALIZED FUEL COST SAVINGS

PROBLEM:

CALCULATE LEVELIZED FUEL COSTS FOR LIQUIFIED COAL FOR 30 YEAR PERIOD STARTING IN 1977, CALCULATE CAPITALIZED FUEL COST SAVINGS FOR BOONWE THERMIONIC TOPPED PLANT AS COMPARED WITH BOONWE CONVENTIONAL PLANT (BOTH USING LIQUIFIED SOLVENT REFINED COAL)

AUDIT

REMARKS

CHECKER'S REMARKS:

APPROACH/ASSUMPTIONS:

INTEREST RATE 10%
INFLATION RATE 6.5%
CAPITAL CHARGE RATE 14%
LEVELIZATION PERIOD 30 YRS

AUDIT

REMARKS

CHECKER'S REMARKS:

SOURCES—DATA/EQUATIONS:

AUDIT

REMARKS

CHECKER'S REMARKS:

CONCLUSIONS:

LEVELIZED FUEL COST OF LIQUIFIED COAL = \$ 4.92/MM BTU
CAPITALIZED FUEL COST SAVINGS = \$ 260 x 10⁶ (FOR FIXED CHARGE RATE = 15%)
= \$ 217 x 10⁶ (" " " " = 18%)

AUDIT

REMARKS

CHECKER'S REMARKS:

AUDIT SUMMARY

CHECKS

CORRECTIVE ACTION TAKEN BY

CALCULATED BY

Richard J. Dwyer

DATE

10/17/77

CHECKED BY

Brian J. Wokaty

DATE

11/15/77

INFRACTIONS

DATE

AUDITED BY

DATE

CALCULATION SHEET

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Item

Client THERMO ELECTRON CORP. Location _____ Est. No. _____ J.O. No. 12628.02
 Subject LIQUIFIED COAL LEVELIZED FUEL Date 10-17-77 By DUBE
COST Checked 11/15/77 By BJD
 Based on _____ Revised _____ By _____

DATE	YEAR	CAPITAL CHARGE @ 14%	OP. COST @ 6.5% INFLATION	DISCOUNT FACTOR @ 10%	\$/MM BTU H-COAL FUEL COST AT YEAR	\$/MM BTU H-COAL DISCOUNTED TO 1977
1977	1	0.99	2.09	0.909	3.08	2.80
	2		2.23	0.826	3.22	2.66
	3		2.37	0.751	3.36	2.52
	4		2.52	0.683	3.51	2.40
	5		2.69	0.621	3.68	2.29
	6		2.86	0.565	3.85	2.18
	7		3.05	0.513	4.04	2.07
	8		3.25	0.467	4.24	1.98
	9		3.46	0.424	4.45	1.89
	10		3.68	0.386	4.67	1.80
	11		3.92	0.351	4.91	1.72
	12		4.18	0.319	5.17	1.65
	13		4.45	0.290	5.44	1.58
	14		4.74	0.263	5.73	1.51
	15		5.05	0.239	6.04	1.44
	16		5.38	0.218	6.37	1.39
	17		5.72	0.198	6.71	1.33
	18		6.10	0.180	7.09	1.28
	19		6.49	0.164	7.48	1.23
	20		6.91	0.149	7.90	1.18
	21		7.36	0.135	8.35	1.13
	22		7.84	0.123	8.83	1.09
	23		8.35	0.112	9.34	1.05
	24		8.90	0.102	9.89	1.01
	25		9.47	0.092	10.46	0.96
	26		10.09	0.084	11.08	0.93
	27		10.75	0.076	11.74	0.89
	28		11.44	0.069	12.43	0.86
	29		12.19	0.063	13.18	0.83
2006	30		12.98	0.057	13.97	0.79

$$\Sigma = 46.44$$

$$C_{77} = 0.10608$$

$$46.44 \times 0.10608 = \$4.92 = \text{LEVELIZED FUEL COST}$$

Client THERMO ELECTRON CORP. Location _____ Est. No. _____ J.O. No. 12628.02
 Subject CAPITALIZED FUEL COST, Date 10-17-77 By DURE
LIQUIFIED COAL Checked 11/15/77 By BJD
 Based on _____ Revised _____ By _____

1. CONVENTIONAL BOILER VS. ADVANCED THERMIONICS BLR: *

SEE ASSUMED HEAT RATE CURVES AND
 LEVELIZED FUEL COST CALC'S.

**

$$C.F.C. = \Delta H.R. \times L.F.C. \times 8760 \times OPER. FACTOR \times$$

$$FULL LOAD, KW / F.C.R.$$

$\Delta H.R.$ = DELTA UNIT NET HEAT RATE

$L.F.C.$ = LEVELIZED FUEL COST, \$/10⁶ Btu

OPER. FACTOR = 60% (ASSUMED)

$F.C.R.$ = FIXED CHARGE RATE —
 (RANGE = 15 - 18%)

$$C.F.C. (1) = \frac{(9123 - 7235)(4.92)(8760)(0.60)(800,000)}{0.15 (10^6)}$$

$$= \$260 \times 10^6 \text{ (IN FAVOR OF THERMIONICS BLR.)}$$

$$C.F.C. (2) = \frac{(9123 - 7235)(4.92)(8760)(0.60)(800,000)}{0.18 (10^6)}$$

$$= \$217 \times 10^6 \text{ (IN FAVOR OF THERMIONICS BLR.)}$$

* BOTH BOILERS FIRED WITH LIQUIFIED COAL.

** C.F.C. = CAPITALIZED FUEL COST

CALCULATION SUMMARY

▲ 5010.47

CALCULATION NO.

APPENDIX I

DATE REV. 11/28/77 PAGE

10/28/77 1 of 11

CLIENT

THERMO ELECTRON CORPORATION

J. O. NUMBER

12628.02

SUBJECT

ELECTRICAL POWER OUTPUT FROM THERMIONIC DEVICES

PROBLEM:

DETERMINE THE TOTAL ELECTRIC POWER GENERATED BY THERMIONIC DEVICES, MOUNTED AND INTERCONNECTED WITHIN: I) A CURRENT TECHNOLOGY BOILER
II) AN ADVANCED TECHNOLOGY BOILER

AUDIT

REMARKS

CHECKER'S REMARKS:

APPROACH/ASSUMPTIONS:

1. DEVICE CURRENT WILL FALL IN THE RANGE 300-600 AMPS
2. FOR LARGE VOLTAGE BUILD UP, STRINGS WILL BE ASSUMED TO BE GROUNDED AT MIDPOINT
3. LOSSES IN INTERCONNECTORS ASSUMED 60 MILLIVOLTS
4. LOSSES IN EXTERNAL BUSWORK $\leq$ 4 PERCENT

AUDIT

REMARKS

CHECKER'S REMARKS:

SOURCES—DATA/EQUATIONS:

1. SEE APPENDIX A THROUGH F, THIS REPORT
2. REVIEW OF THERMIONICS FOR POWER GENERATION, DEC 29, 1976 REPORT

AUDIT

REMARKS

CHECKER'S REMARKS:

CONCLUSIONS:

CASE I) P_{OUTPUT} AT GENERATOR TERMINALS: 131 MWE
CASE II) P_{OUTPUT} AT GENERATOR TERMINALS: 300 MWE

AUDIT

REMARKS

CHECKER'S REMARKS:

AUDIT SUMMARY		CALCULATED BY		DATE	
CHECKS	CORRECTIVE ACTION TAKEN BY			REV 11/28/77	10/28/77
INFRACTIONS	DATE			DATE	12/1/77
		AUDITED BY		DATE	

Client THERMO ELECTRON CORP Location _____ Est. No. _____ J.O. No. 12628.02
 Subject _____ Date 10-28-77 By WWEINBERGER
 Checked RAV 11/28/77 By _____
 Based on _____ Revised _____ By _____

I) CURRENT TECHNOLOGY BOILERS - ELECTRICAL POWER OUTPUT FROM THERMIONIC DEVICES

A.) FIRE BOX REGION (FLAME TEMP = 2862°F, REF E/6)

REFERENCE

1. DEVICE PARAMETERS:

$$T_E = 1500^\circ \text{K}$$

APPENDIX A/4

$$T_C = 670^\circ \text{K}$$

C/4

$$P_{IN} = 13.5 \text{ W/cm}^2$$

A/4

$$P_{OUT} = \eta \times P_{IN}$$

$$P_{OUT} = 4.4 \text{ W/cm}^2$$

$$V_{OUT} = 1.1 \text{ V dc}$$

TECO REPORT 12/76

$$I_{OUT} = 4.05 \text{ amp/cm}^2$$

$$\eta = 0.33$$

E/8

2. ZONE HEIGHT: ~ 1750 CM

3. DEVICE GEOMETRY: ~ 7.5 x 20 CM, AREA = 150 CM²

4. DEVICE CURRENT: $I_D = (4.0 \text{ AMP/CM}^2)(150 \text{ CM}^2)$

$$I_D = 600 \text{ AMP}$$

5. STRING CURRENT (TWO DEVICES IN PARALLEL): $I_S = 2 \times 600$

$$I_S = 1200 \text{ AMP}$$

6. STRING LENGTH: 1750 CM x 4

$$= 7000 \text{ CM}$$

7. NO. OF UNITS IN A STRING, ASSUMING 5% OF DEVICE LENGTH FOR INTERCONNECTION:

$$= \frac{7000 \text{ CM}}{20 \text{ CM}(1.05)}$$

$$= 334 \text{ UNITS IN SERIES}$$

8. STRING OUTPUT VOLTAGE, ASSUMING 60 MV LOSS IN EACH SERIES CONNECTOR:

$$V_S = (334)(1.1 - .060) \text{ V dc}$$

$$V_S = 348 \text{ V dc}$$

9. POWER FROM EACH STRING, AT OUTPUT TERMINALS:

$$P_S = (348 \text{ V dc})(1200 \text{ AMP}) \text{ WATTS}$$

$$= 418 \text{ KW}$$

Client THERMO ELECTRON CORP Location _____ Est. No. _____ J.O. No. 12628.02
 Subject _____ Date 10/28/77 By JWW
 Checked REV 11/28/77 By _____
 Based on _____ Revised _____ By _____

A.) FIRE BOX REGION (CONT.)

9. POWER FROM STRING: $P_S = 418 \text{ KW}$

10. NO. OF TUBES WITH MOUNTED THERMIONICS: $\frac{170'}{0.25'} = 680 \text{ TUBES}$
 USEABLE PERIM. = $82' + 2 \times 44'$ (REF E/3)

11. NO. OF STRINGS IN ZONE (FOUR TUBE SURFACES PER STRING): $\frac{680}{8} = 85 \text{ STRINGS}$

12. TOTAL ELECTRICAL GENERATION, FIREBOX ZONE: $P_T = (85 \text{ STRINGS})(418 \text{ KW})$
 $P_T = 35.5 \text{ MW}$

B.) PLATTEN REGION (FLAME TEMP 2695 °F; REF E/7)

1. DEVICE PARAMETERS:

$T_E = 1460^\circ \text{K}$

REFERENCE

$T_C = 660^\circ \text{K}$

E/7

$P_{IN} = 10.5 \text{ W/cm}^2$

E/7

$P_{OUT} = \eta \times P_{IN}$

$P_{OUT} = 3.3 \text{ W/cm}^2$

A/4

$V_{OUT} = 1.1 \text{ V dc}$

TECO REPORT 12/76

$J = 3.0 \text{ amp/cm}^2$

$\eta = 31.8\%$

E/4

FOR WATERWALLS

2. WATERWALL HEIGHT: $\sim 1600 \text{ CM}$

E/3

3. DEVICE GEOMETRY: $7.5 \times 13.3 \text{ CM}$, AREA = 100 CM^2

4. DEVICE CURRENT: $I_D = (3 \text{ amp/cm}^2)(100 \text{ CM}^2)$

$I_D = 300 \text{ amp}$

5. STRING CURRENT: (4 DEVICES IN PARALLEL ALONG EACH PAIR OF TUBES)

$I_S = 4 \times I_D$

$I_S = 1200 \text{ amp}$

6. STRING LENGTH: (STRING TRAVERSES WATERWALL SIX TIMES)

$6 \times 1600 \text{ CM} = 9600 \text{ CM}$

1 Client HERMO ELECTRON CORP Location _____ Est. No. _____ J.O. No. 12628.02
 2 Subject _____ Date 10/28/77 By WW
 3 _____ Checked REV 11/28/77 By _____
 4 Based on _____ Revised _____ By _____

3.) PLATTEN REGION (CONT.)

REFERENCE

7. NO. OF UNITS IN A STRING: $\frac{9600 \text{ CM}}{(13.33 \times 1.05) \text{ CM}} = 686 \text{ UNITS}$

8. STRING OUTPUT VOLTAGE: $V_s = \frac{686}{2} (1.1 - 1.060)$
 (60 MVOLT LOSS IN EACH SERIES CONNECTION)
 $V_s = 357 \text{ V dc}$

9. POWER/STRING: $P_s = (357 \text{ V dc}) (1200 \text{ amp})$
 $P_s = 428 \text{ KW}$

10. NO. OF TUBES WITH THERMIONICS: $(680) \left(1 - \frac{15 \times 82}{53 \times 82 + 2 \times 44 \times 53} \right)$ E/4
 $(680) (.8635)$
 588

11. NO OF STRINGS: $= \frac{588}{12}$
 (12 TUBE SURFACES)
 PER STRING $= 49 \text{ STRINGS}$

12. TOTAL ELECTRICAL GENERATION FROM THE WATERWALL, PLATTEN REGION:
 $= P_T = (49 \text{ STRINGS}) (428 \text{ KW})$
 $P_T = 21 \text{ MWe}$

FOR PLATTENS

13. PLATTEN LENGTH: $\sim 1550 \text{ CM}$ E/10
 (ASSUMING 60 TUBES/PLATTEN)

14. LENGTH OF STRING FOR THERMIONICS ON PLATTENS (ASSUMING SIX TRAVERSES): 9300 CM

15. NO. OF DEVICES: $\frac{9300 \text{ CM}}{13.33 \times 1.05 \text{ CM}} = 664$
 (4 DEVICES IN PARALLEL)
 EACH $7.5 \times 13.33 \text{ CM}$

16. VOLTAGE OUTPUT, PLATTEN STRINGS: $V_s = \frac{664}{2} (1.04 \text{ Vdc})$

Client THERMO ELECTRON CORP Location _____ Est. No. _____ J.O. No. 12628.02
Subject _____ Date 10/18/77 By JWW
Checked REV 11/28/77 By _____
Based on _____ Revised _____ By _____

B.) PLATTEN REGION (CONT.)

16. VOLTAGE OUTPUT: $V_s = 346 \text{ Vdc}$

17. STRING CURRENT: $I_s = (7.5 \text{ cm} \times 13.33 \text{ cm}) (3 \text{ amps/cm}^2) (4)$
(GEOMETRY $7.5 \times 13.33 \text{ cm}$) $I_s = 1200 \text{ amp}$

18. POWER STRING: $P_s = 415 \text{ KW}$

19. BASED ON APPENDIX E,
THERE ARE APPROX
60 TUBES / PLATTEN.
NO. OF STRINGS PER
PLATTEN:

$$\frac{60}{12} \times (2 \text{ SURFACES/TUBE}) = 10 \text{ STRINGS}$$

20. POWER OUTPUT, PER
PLATTEN:

$$P_p = (10) (415 \text{ KW})$$

$$P_p = 4.15 \text{ MWe}$$

21. TOTAL ELECTRICAL
GENERATION FROM
PLATTENS:

$$P_T = (21 \text{ PLATTENS}) (4.15 \text{ MWe})$$

$$P_T = 87 \text{ MWe}$$

22. TOTAL ELECTRICAL
GENERATION (DIRECT)

FROM PLATTEN REGION: $21 \text{ MWe} + 87 \text{ MWe}$

PLATTENS	108 MWe
WATERWALLS	35.5
	<u>143.5 MWe</u>

SUMMARY

1. OUTPUT AT TERMINALS OF
THERMIONICS, FOR CONVENTIONAL
BOILER, INCLUDING CONNECTOR
LOSSES AND DEVICE SPACING:

$\sim 143.5 \text{ MWe}$

Client HERMO ELECTRON

Location

Est. No.

J.O. No. 12628.02

Subject

Date 10/18/77

By JWW

Checked REV 11/28/77 By

Based on

Revised

By

II) ADVANCED TECHNOLOGY BOILERS— ELECTRICAL POWER OUTPUT FROM THERMIONIC DEVICES

A.) ZONE 1, PREHEAT AND WATERWALL

1. DEVICE PARAMETERS:

$$T_E = 1600^\circ K$$

REFERENCE S

$$T_C = 670^\circ K$$

APPEND. F/6

$$P_{IN} = 22 \text{ W/CM}^2$$

F/6

$$P_{OUT} = \eta \times P_{IN}$$

$$P_{OUT} = 7.59 \text{ W/CM}^2 / 7.93$$

$$V_{OUT} = 1.3 \text{ V d-c (TECO REPORT)}$$

$$I = 5.83 \text{ amp/CM}^2 / 6.09$$

$$\eta = 36 \%$$

F/6

2. PREHEAT REGION SURFACE AREA, PER MODULE:

$$A = 8'25'$$

$$A = 200 \text{ SQ FT}$$

SEVENTY EIGHT PERCENT OF AREA ASSUMED AVAILABLE FOR THERMIONICS:

$$A = 156 \text{ SQ FT OR}$$

$$A = 145,000 \text{ CM}^2$$

3. POWER OUTPUT, THERMIONICS; INCLUDING CONNECTOR LOSSES

$$P_{OUT} = (145,000)(7.59 \text{ W/CM}^2)$$

$$P_{OUT} = 1100 \text{ KW}$$

$$\text{CONN LOSS} = \left(\frac{60 \text{ MV}}{1.3 \text{ V}} \right) (1100 \text{ KW})$$

$$\text{CONN LOSS} = 51 \text{ KW}$$

$$P_{OUT, NET} = 1050 \text{ KW}$$

POUTPUT/MODULE:

$$2 \times 1050 \text{ KW}$$

$$P_M = 2.1 \text{ MW}$$

3. WATERWALL REGION, SURFACE AREA, PER MODULE:

$$A = 1512,861 \text{ CM}^2$$

LESS: 58 CONNECTORS

$$A = 1437218 \text{ CM}^2$$

POWER OUTPUT

$$P_{OUT} = (1437218)(7.93 \text{ W/CM}^2)$$

Client THERMO ELECTRON CORP Location Est. No. J.O. No. 12628.02
Subject Date 10/28/77 By JWW
Checked REV 11/28/77 By
Based on Revised By

A) ZONE 1, PREHEAT AND WATERWALL (CONT.)

3. WATERWALL REGION (CONT.)

$$P_{OUT} = 11.4 \text{ MW}$$

$$\text{LESS } \left(\frac{.06}{1.3} \right) 11.4 \text{ MW}$$

$$P_{OUT, NET} = 11.4 - .53 \text{ MW}$$

$$P_{OUT, NET} = 10.9 \text{ MW}$$

TWO WALLS PER MODULE,
HENCE P_{OUT} / MODULE:

$$P_{OUT} / \text{MODULE} = 21.8 \text{ MW}$$

4. SUM OF ELECTRICAL POWER
GENERATION, PER MODULE
ZONE 1

$$P_1 = 24 \text{ MWe}$$

B) ZONE 2

1. DEVICE PARAMETERS:

$$P_{OUT} = \eta \times P_{IN}$$

REFERENCES
 $T_E = 1575^\circ \text{K}$ APPENDIX F/7
 $T_C = 670^\circ \text{K}$ F/7
 $P_{IN} = 19.5 \text{ W/cm}^2$ F/7
 $P_{OUT} = 6.88 \text{ W/cm}^2$
 $V_{OUT} = 1.3 \text{ Vdc}$ (TECO REPORT)
 $J = 5.29 \text{ amp/cm}^2$
 $\eta = 35.3\%$ F/7

2. AREA PER MODULE: $(115.8") \times (96" \times 2 + 377" \times 2)$

$$A = 109,546.8 \text{ sq in}$$

$$A = 706,752 \text{ sq cm}$$

$$A_{\text{LESS } 5\%} = 671,414 \text{ cm}^2$$

3. POWER OUT:

$$P_{OUT} = (6.88)(671,414) = 4.62 \text{ MW}$$

4. LESS CONNECTOR LOSSES: $\left(\frac{.06}{1.3} \right) (4.62)$

$$= .21$$

$$P_{OUT, NET2} = 4.4 \text{ MW}$$

Client THERMO ELECTRON CORP Location _____ Est. No. _____ J.O. No. 12628.02
Subject _____ Date 10/28/77 By WW
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Based on _____ Revised _____ By _____

C) ZONE 3

1. DEVICE PARAMETERS:

$$P_{OUT} = \eta \times P_{IN}$$

$$T_E = 1525^\circ K$$

$$T_C = 670^\circ K$$

$$P_{IN} = 15 W/cm^2$$

$$P_{OUT} = 4.95 W/cm^2$$

$$V_{OUT} = 1.2 V dc$$

$$J = 4.12 amp/cm^2$$

$$\eta = 33\%$$

REFERENCES

F/8

F/8

F/8

TECO REPORT

F/8

2. AREA:

$$A = (150")(946")$$

$$A = 918,412 cm^2$$

$$LESS 5\% A = 872,491 cm^2$$

$$3. POWER OUTPUT: P_{OUT} = (872,491 cm^2)(4.95)(1 - \frac{.06}{1.2})$$

$$P_{OUT3} = 4.32 Mwe$$

D) ZONE 4

1. DEVICE PARAMETERS:

$$P_{OUT} = \eta \times P_{IN}$$

$$T_E = 1450^\circ K$$

$$T_C = 660^\circ K$$

$$P_{IN} = 11.8 W/cm^2$$

$$P_{OUT} = 3.77 W/cm^2$$

$$V_{OUT} = 1.1 V dc$$

$$J = 3.43 amp/cm^2$$

$$\eta = 29.5\%$$

F/8

F/8

F/8

TECO REPORT

F/8

2. AREA:

$$A = (189.6")(946")$$

$$A = 1,157,169 cm^2$$

$$LESS 5\%; A = 1,099,311 cm^2$$

$$3. POWER OUTPUT: P_{OUT} = (1,099,311 cm^2)(3.77)(1 - \frac{.06}{1.1})$$

$$P_{OUT4} = 3.92 Mwe$$

E) ZONE 5

1. DEVICE PARAMETERS:

$$P_{OUT} = \eta \times P_{IN}$$

$$T_E = 1400^\circ K$$

$$T_C = 660^\circ K$$

$$P_{IN} = 7.5 W/cm^2$$

$$P_{OUT} = 2.21 W/cm^2$$

$$V_{OUT} = 1.1 V dc$$

$$J = 2.01 amp/cm^2$$

$$\eta = 29.5\%$$

F/9

F/9

F/9

TECO REPORT 12/76

F/9

1 Client THERMO ELECTRON CORP Location Est. No. J.O. No. 2628.02
2 Subject Date 10/28/77 By JWW
3 Checked REV 11/28/77 By
4 Based on Revised By

E) ZONE 5 (CONT.)

2. AREA, PER MODULE: $A = (294.6'')(946'')$
 $A = 1,798,006 \text{ CM}^2$

LESS 52: $A = 1,708,106 \text{ CM}^2$

3. POWER OUTPUT

$P_{\text{OUT}} = (1,708,106)(2.21 \text{ W/CM}^2)(1 - \frac{.06}{1.1})$

$P_{\text{OUT}} = 3.57 \text{ MWe}$

F) TOTAL POWER/MODULE: $P_{T/M} = P_1 + P_2 + P_3 + P_4 + P_5$

$P_{T/M} = 40.21 \text{ MWe}$

TOTAL POWER = (8 MODULE)(40.21)

$P_T = 322 \text{ MWe}$

G) A STRING CONFIGURATION, WATERWALL ZONES

1. DEVICE GEOMETRY, ZONE 1, SELECTED TO BE

$7.5 \text{ CM} \times 14.7 \text{ CM}$

THUS $I_{\text{STRING}} = 2 \times (7.5 \times 14.7)(6.09 \text{ amp/CM}^2)$

$I_s = 1350 \text{ AMP}$

2. DEVICE GEOMETRY, ZONE 2, FOR THIS STRING

$7.5 \text{ CM} \times \frac{(6.09)(10 \text{ CM})}{5.29}$

$7.5 \text{ CM} \times 17.0 \text{ CM}$

THUS MATCHING I_s OF 1350 AMP

3. DEVICE GEOMETRY, ZONE 3, FOR THIS STRING

CALCULATION SHEET

Client HERMO ELECTRON CORP

Location

Est. No.

J.O. No. 12628.02

Subject

Date 10/28/77By JWWChecked REV 11/28/77 By

Based on

Revised

By

II) ADVANCED BOILER TECHNOLOGY (CONT.)

G) STRING CONFIGURATION (CONT.)

3) DEVICE GEOMETRY, ZONE 3:

$$7.5 \text{ CM} \times \left(\frac{6.09}{4.12} \right) (10) \text{ CM}$$

$$7.5 \text{ CM} \times 21.8 \text{ CM}$$

4) DEVICE GEOMETRY, ZONE 4:

$$7.5 \text{ CM} \times \left(\frac{6.09}{3.43} \right) (10 \text{ CM})$$

$$7.5 \text{ CM} \times 26.2 \text{ CM}$$

5) DEVICE GEOMETRY, ZONE 5:

$$7.5 \text{ CM} \times \left(\frac{6.09}{2.01} \right) (10 \text{ CM})$$

$$7.5 \text{ CM} \times 44.8 \text{ CM} \quad (\text{DEVICES CAN ACTUALLY BE ONE HALF THIS LENGTH AND CONNECTED IN PARALLEL})$$

H) STRING VOLTAGE

WE ASSUME A STRING TRAVERSES THE WATERWALL TWO TIMES

$$1) \text{ NO. OF UNITS IN ZONE 1: } \frac{1582 \text{ CM} \times 2}{14.7 \text{ CM} (1.05)}$$

WHERE 1582 CM IS HEIGHT OF ZONE 1

1.05 ACCOUNTS FOR FIVE PERCENT CONNECTOR LENGTH

NO. UNITS: 204 UNITS

$$V_{S, \text{ZONE 1}} = (204) (1.3 - .06)$$

$$= 253 \text{ VOLTS}$$

$$2) \text{ NO. OF UNITS IN ZONE 2: } \frac{294 \text{ CM} \times 2}{(17.0) (1.05)}$$

= 33 UNITS

$$V_{S, \text{ZONE 2}} = 33 (1.3 - .06)$$

$$V_{S, 2} = 40 \text{ VOLTS}$$

1 Client THERMO ELECTRON CORP Location _____ Est. No. _____ J.O. No. 12628.02
 2 Subject _____ Date 10/28/77 By JWW
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H) STRING VOLTAGE (CONT.)

3) NO. OF UNITS ZONE 3 : $\frac{382 \text{ CM} \times 2}{(21.8)(1.05)}$

= 33 UNITS

$V_{S, \text{ZONE 3}} = (33)(1.2-.06)$

$V_{S, 3} = 38 \text{ VOLTS}$

4) NO. OF UNITS ZONE 4 : $\frac{482 \text{ CM} \times 2}{(26.2)(1.05)}$

= 35 UNITS

$V_{S, 4} = 35(1.1-.06)$

= 36 VOLTS

5) NO. OF UNITS, ZONE 5 : $\frac{748 \text{ CM} \times 2}{(44.8 \text{ CM})(1.05)}$

= 32 UNITS

$V_{S, 5} = 32(1.1-.06)$

$V_{S, 5} = 33 \text{ VOLTS}$

6) $\therefore V_S = V_{S1} + V_{S2} + V_{S3} + V_{S4} + V_{S5}$

$V_S = 400 \text{ VOLTS}$

WITH MIDPOINT GROUNDING, MAXIMUM VOLTAGE
 WOULD BE 200 VOLTS

I) STRING POWER

$P_S = (400 \text{ V dc})(1350 \text{ AMP})$

$P_S = 540 \text{ KW}$