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THERMOELECTRIC GENERATORS POWERED BY THERMAL WASTE FROM ELECTRIC POWER PLANTS

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THERMOELECTRIC GENERATORS POWERED BY THERMAL WASTE FROM ELECTRIC POWER PLANTS

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ABSTRACT

The feasibility of recovering electricity from the waste heat of electric power plants was assessed. Sources considered were: stack flue gas, gas-turbine exhaust, and condensing steam. Typical 1600 MW fossil-fuel steam plants and gas-turbine plants were used as examples. Flat plate heat exchangers were designed with thermoelectric couples arranged in series within the plates. Heat flux, conversion efficiencies, and flow friction losses were calculated. Except for the condenser application, the friction losses are several times the thermoelectric power generated. Under favorable conditions, 3 to 9 MW is obtainable from the thermoelectric condensers. The high material cost, however, precludes all such applications today.

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SECTION I

INTRODUCTION

Thermoelectric generators are known to be inefficient. Their use in direct energy conversion lead to several times greater fuel consumption than conventional methods. Thermoelectric generators powered by the waste heat would not suffer, at least, from this disadvantage. The high material and development costs have rendered such generators commercially unattractive, if not outright prohibitive today.

If the quest for a higher efficiency persists despite an economic penalty, thermoelectric devices could be used to reduce the waste heat from conventional power plants by converting a minute portion of it to electricity. This, admittedly, is not the most efficient or least expensive way. Against this background, we proceed to entertain the possibility of applying thermoelectric materials to waste heat utilization, leaving its economic feasibility aside for the moment. When we return to the discussion of the latter problem, we are able to present only a rough estimate of the cost involved.

In a study by Embry and Tudor (1), thermoelectric generators were powered by the exhaust heat of an auto engine. The generators were shown to supply the entire electrical requirements of an automobile, thereby slightly increasing the overall engine efficiency. Embry and Tudor presented several references to earlier attempts concerning similar application of thermoelectric devices.

The present study applies thermoelectric devices to the utilization of waste heat from thermal power plants. In particular, the low pressure steam from a steam power cycle could be condensed in a special "thermoelectric condenser" that has thermocouple circuits embedded within its walls. While conducting the latent heat from the condensing steam to the cooling water, the couples convert a portion of this heat to electricity.

A second source of waste heat in a conventional fossil-fueled electric power plant is the hot stack gases released into the atmosphere. A third source is the hot exhaust gases from a gas turbine electric power system. The hot gases from these sources, too, could be passed through special (thermoelectric) heat exchangers for generating electricity while conducting heat through the couples to the ambient air on the cold side.

The condenser application receives the primary emphasis. The gas-turbine exhaust and the stack gas applications will be discussed briefly, showing in each case some of the results without presenting the details. While reasonable designs will be sought in the course of this study to enable meaningful estimates, no attempt will be made to find the most optimum situations.

SECTION II

THERMOELECTRIC HEAT EXCHANGER ANALYSIS

It is convienent for the purpose of this discussion to specify a feasible heat exchanger geometry. Consider as a possible arrangement the plate-fin surface geometry shown in Figure 1-a. The heat exchanger is of the cross-flow type. The plates (or modules) separating the hot fluid from the cold fluid contain the thermoelectric elements. Such plates lend themselves easily to the current manufacturing techniques for thermoelectric modules. The module surfaces are protected on each side by a sheet of stainless steel plate, 0.005-inch thick. The elements within the plates are connected in series as shown in Figure 1-b. The plates themselves are arranged in pairs in such a way that the hot (or cold) junctions of their couples face one another. For condensers, the plates are finned only on the water side. For gas-to-gas exchangers, the plates are finned on both sides. The spacing between the plates on the cold side is held equal to that on the hot side for all cases considered here. The heat exchangers are made in the shape of cubes - 12-feet on each side. The plate module thickness is varied between 0.004-inch to 0.2-inch. The packing density of the couples is not specified, but the calculations assume very dense packing.

The convective heat transfer coefficient is calculated from:

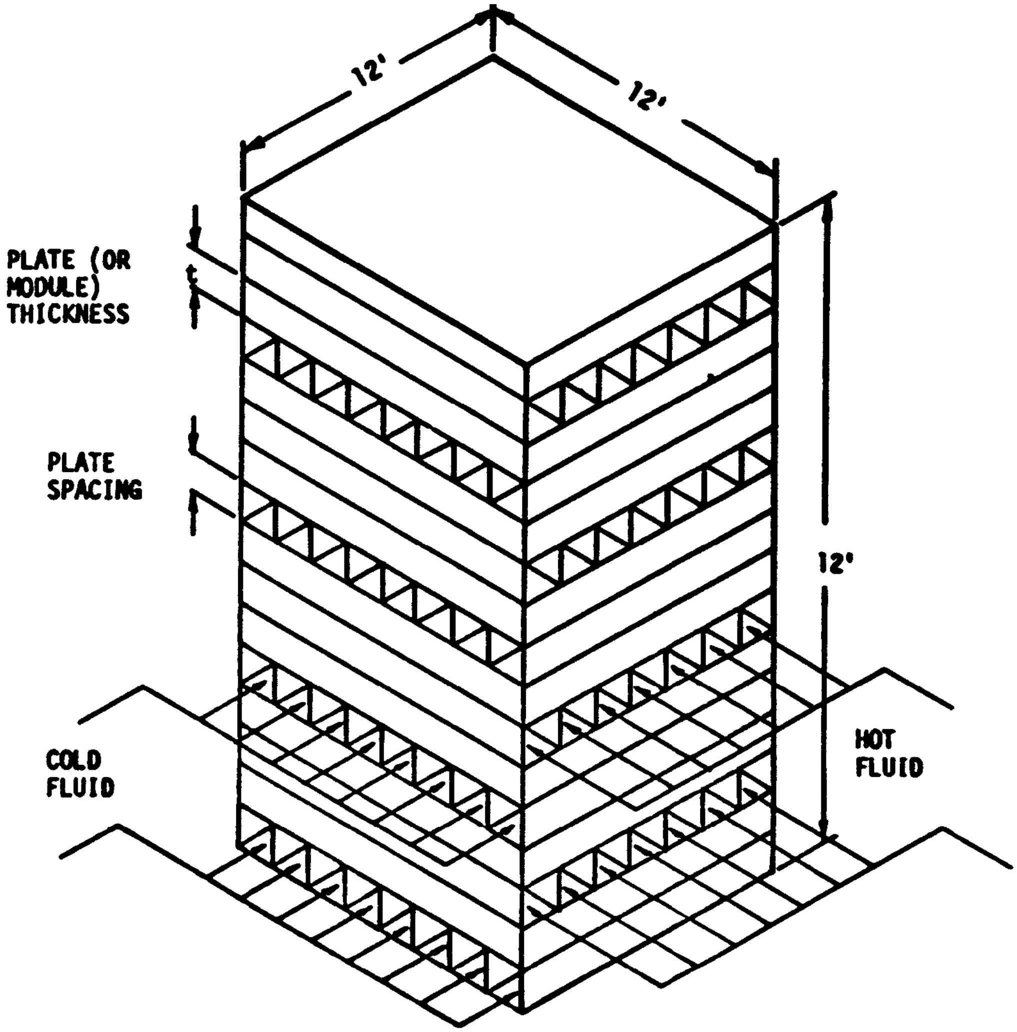
$$h = (c_p\mu/D\ N_{Pr}^{-2/3})\ (N_{St}N_{Pr}^{2/3})\ N_R \qquad (1)$$

where N_R is the Reynolds Number, N_{St} is the Stanton Number, N_{Pr} the Prandtl Number, D the hydraulic diameter, μ the viscosity coefficient, and c_p the specific heat. The friction power expended per unit surface area is evaluated from:

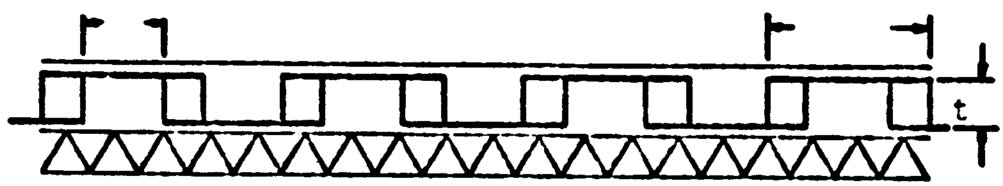
$$E = (2\rho^2 g)^{-1}(\mu/D)^3\ f\ N_R^{3} \qquad (2)$$

where ρ is the density, and f the friction factor. The heat transfer and flow friction performance data are found in Reference 2.

To enable calculation of the thermoelectric conversion efficiencies, surface temperatures on the hot and cold sides of the modules will be evaluated by allowing a linearly proportionate temperature drop for all the fluid and material thermal resistances across the plate. The effect of the fluid temperature changes along the heat exchanger path is accounted for by taking a simple arithmetic average of the inlet and outlet fluid temperatures. The maximum thermal conversion efficiency is evaluated from:



a) HEAT EXCHANGER UNIT SHOWING PLATE ARRANGEMENT AND DIMENSIONS.



b) CROSS-SECTION OF A MODULE SHOWING THE THERMOELECTRIC ELEMENTS, STAINLESS STEEL PROTECTING SHEET AND FINS.

FIGURE 1. A CROSSFLOW THERMOELECTRIC HEAT EXCHANGER UNIT.

$$\eta = \frac{T_1 - T_o}{T_1} \quad \frac{m_o - 1}{m_o + T_o/T_1} \quad \text{......................(3)}$$

where

$$m_o = (1 + Z\ (T_o + T_1)/2)^{1/2} \text{....................(4)}$$

T_1 is the hot junction and T_o the cold junction temperatures both measured in the absolute scale, and Z is the figure of merit for the thermoelectric material.

The thermoelectric material properties were evaluated at the mean temperatures of the hot and cold fluids. The alloys considered in this study were Bi_2Te_3-Bi_2Se_3 (n-type) and Bi_2Te_3-Sb_2Te_3 (p-type).

SECTION III

THERMOELECTRICITY FROM CONDENSING STEAM

In this application, the condenser cooling water serves as the heat sink and the condensing steam as the heat source. The "thermoelectric condenser" has the general features just described and it serves the usual function of condensing steam and maintaining the appropriate turbine back pressure in a steam power cycle. In addition, it converts a fraction of the latent heat of the steam to electricity. The magnitude of electric power so generated is governed by the temperature of the cooling water, the turbine back pressure, and the steam rate. In order to fix our ideas, we choose the three power plants listed in Table 1. The steam flow delivered to the condensers is assumed equal to 7×10^6 lb/hr. The temperature of the cooling water available to the first power plant is 40°F and the corresponding back pressure is 0.5-inches of mercury absolute. The second and third power plants have access to cooling waters at temperatures of 65°F and 80°F, respectively. The corresponding turbine back pressures are 1.5" Hg and 4" Hg, absolute. The second power plant is an example of a typical modern generating system. The other two are examples of somewhat extreme pressure and temperature conditions. The latter could conceivably represent two off-design operating conditions of the second power plant. The terminal temperature difference between the condensate and the cooling water exit temperature is 5°F or more. The approximate relationship between the cooling water temperature is 5°F or more. The approximate relationships between the cooling water temperature and the most economical condenser pressure are assumed in accordance with the data presented in References 3 and 4. The total heat carried away by the cooling water decreases slightly with condensing pressure because of the corresponding decrease in the latent heat of vaporization. The efficiency of the power plant increases at lower back pressures with the corresponding increase in the power plant output. For a station efficiency of 40 percent, the electric output from power plant II is roughly 1640 MW.

The third power plant represents the most favorable conditions for thermoelectric power generation for it has the highest condenser temperature. Detailed thermoelectric condenser design calculations are carried out for this case alone. The other two cases are treated in a general way only. The design procedure followed is briefly outlined below.

The cooling water rate was estimated for a specific surface geometry and an initial plate thickness. The flow velocity and the Reynolds Number were calculated to be used in Equations (1) and (2). With the aid of experimental performance data from Reference 2 and these equations, the convective heat transfer coefficient and the friction power on the water side were evaluated. The convective heat transfer coefficient was then used in evaluating the fin efficiency, the overall unit heat transfer conductance, the number of heat transfer unit (NTU), and the heat exchanger effectiveness. From the last item, the total heat transfer for a single condenser unit was calculated. More units were added to achieve a total heat transfer equal to 2063 MW - a quantity equal to the latent heat released by the incoming steam in the third power plant. At this point, if the outlet water temperature was equal to 98°F or slightly below, the heat transfer calculation was considered complete and Equation (3) was used to evaluate the thermoelectric conversion efficiency and the power generated. This procedure was repeated for several plate thicknesses and heat exchanger surface geometries. In all calculations, an attempt was made to find the minimum number of condenser units that would condense the incoming steam without raising the outlet water temperature above 98°F. For the convenience in calculation, the convective heat transfer coefficient on the steam side was held at an arbitrary, but a reasonable, magnitude of 3000 Btu/hr ft^2 °F.

TABLE 1

CONDENSER DESIGN DATA FOR THREE COOLING WATER TEMPERATURES

	Plant I	Plant II	Plant III
Cooling water in t_1, °F	40	65	80
Steam pressure, "Hg abs	.5	1.5	4.
Steam temperature t_s, °F	58.9	91.7	125.4
Cooling water out t_2, °F	53	85	98
Steam rate 10^6 lb/hr	7	7	7
Heat to cooling water, MW	2176	2137	2063

Many important parameters were calculated for each design. Among these, three quantities were of particular significance. They were: (1) the power density, P_d, expressed in net kilowatts generated per cubic foot of thermoelectric material; (2) the power density, P_i, expressed in watts per square foot of generating surface; and (3) the net power, P_{net}, expressed in megawatts.

D-1 The Power Density, P_d:

The total thermoelectric power generated in all units less the total power expended in flow friction was divided by the total volume of the thermoelectric material used in the plate modules to find the (net) power density P_d. The plots of P_d against the plate thickness for the three heat transfer surfaces is shown in Figure 2. For each surface geometry, the power density curve peaks at an optimum plate thickness t_d. The power density maxima for the surfaces examined are seen to occur between 0.003-inch to 0.01-inch. The magnitudes of P_d at these points vary between 10.23 KW/ft^3 to 16.67 KW/ft^3.

A physical explanation for the peaking behavior of these curves may be as follows: when the plate is thick, the thermal resistance is great and the heat flux is low. Hence, the surface area and thus the material volume must be increased to allow the required heat transfer to take place. The conversion efficiency is relatively large, but with a small heat flux, only a small amount of power per unit volume can be generated. As the plate thickness is reduced, the heat flux increases, too. This trend continues up to a certain point, t_d, where the plate is still thick enough to maintain a relatively large differential temperature between the hot and cold sides. At this point, the maximum power density occurs. As the plate thickness is further reduced, it becomes progressively more difficult to maintain an adequately large differential temperature across the plate. From there on, the conversion efficiency is drastically reduced and the power density eventually drops to zero.

For the convenience in presenting the condenser performance data, three separate tables are provided. Table 2-a contains the geometric data including the unit and passage dimensions, the number of plate modules per unit, the total number of units, and the optimum plate thickness, t_d, of the power density curve. Table 2-b contains the heat transfer and flow data including the outlet water temperature, water velocity, the Reynolds Number, the heat transfer coefficients, the heat exchanger effectiveness, the total heat transfer and the flow friction. Particular reference should be made in this table to the small magnitudes of the fin effectiveness (18 to 35 percent) and the heat exchanger effectiveness (about 38 percent). These quantities could stand much improvement. Practical heat exchangers have better than 70 percent fin and heat transfer effectiveness. Also, the ratio of the friction power to the heat transfer for these calculations is at least an order of magnitude greater than the corresponding values found in a shell and tube condenser. Some of this is attributed to the poor thermal conductivity of the thermoelectric material as compared with copper alloys and the consequent effect this has on reducing the heat flux and increasing the surface area. Friction is also increased by the presence of fins in these designs.

Finally, the power generation data are shown listed in Table 2-c. These include surface temperatures, Carnot cycle efficiency, conversion efficiency, the ratio of net power to heat transfer, and the flow friction power. The Carnot cycle efficiency is between 0.5 to 2 percent, the conversion efficiency is about ten times less than the latter, and the ratio of net power to heat transfer is slightly less than the conversion efficiency.

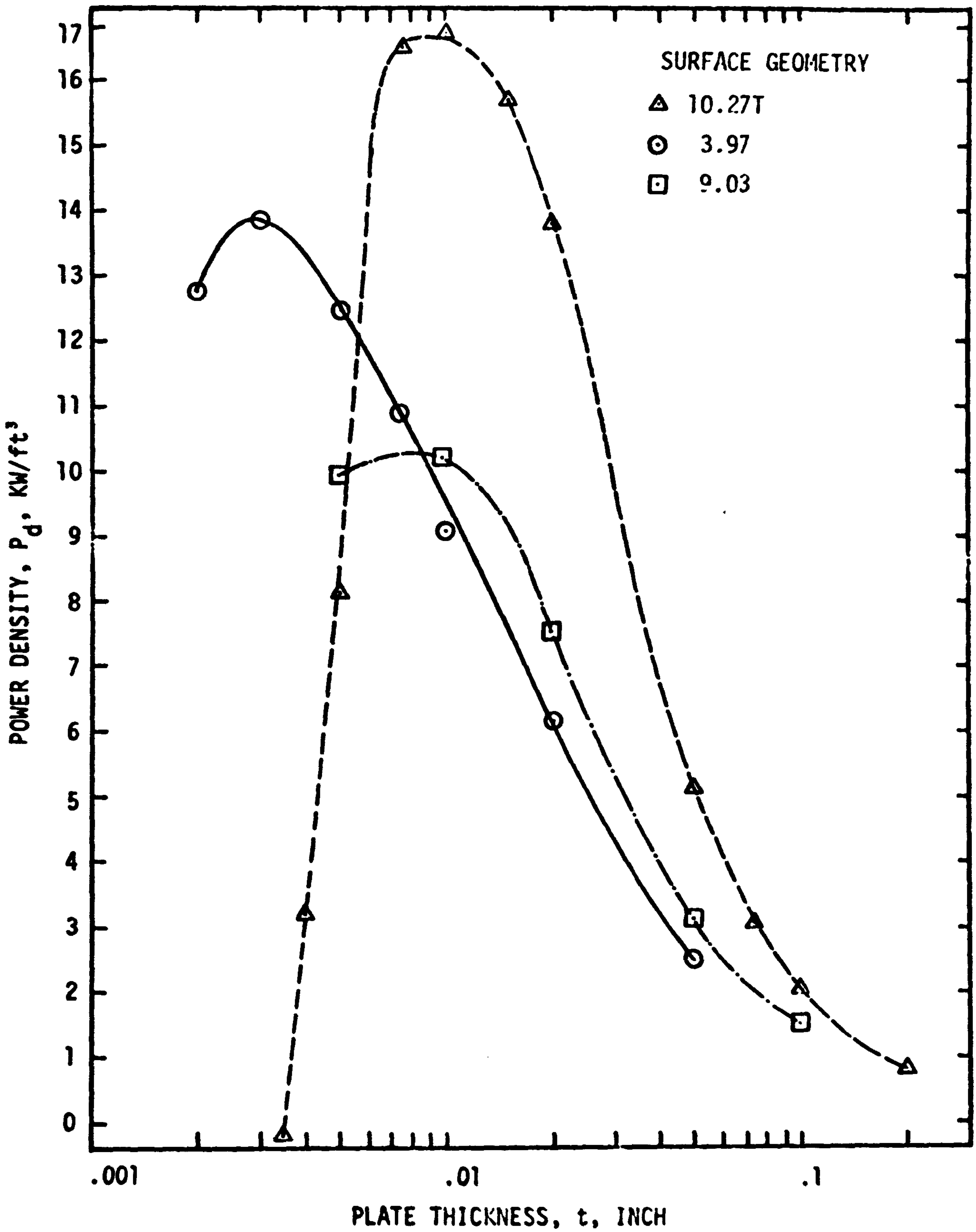


FIGURE 2. POWER DENSITY FOR SEVERAL THERMOELECTRIC CONDENSERS

TABLE 2

PEAK PERFORMANCE DATA FOR THE THREE THERMOELECTRIC CONDENSERS SHOWN IN FIGURE 2

(a) Geometric Data

Unit Designation	10.27T	9.03	3.97
Plate spacing (in.)	.544	.823	.75
Hydraulic diameter (in.)	.151	.1828	.338
Frontal area, ft^2	144	144	144
Unit volume, ft^3	1728	1728	1728
Fin area/Total area	.863	.888	.766
Water side area, Million ft^2	1.450	2.676	1.115
Plate thickness (in.)	.01	.01	.003
Number of plates per unit	256	266	189
Number of units	6	13	11

(b) Heat Exchange & Flow Data

Unit Designation	10.27T	9.03	3.97
Outlet water temperature, °F	97.5	96.6	97.0
Water velocity, ft/sec	4.75	2.24	2.83
Reynolds Number	8080	4620	10800
Water side h, Btu/hr ft^2 °F	1174	562	715
Steam side h, Btu/hr ft^2 °F	3000	3000	3000
Fin effectiveness, %	.216	.18	.350
Total thermal resistance, hr ft^2 °F/Btu	.00741	.0139	.00576
Heat exchanger effectiveness	.385	.365	.374
Flow Friction, Watt/ft^2	1.061	.1307	.211
Total Heat Transfer, MW	2060	2064	2059
Friction/heat Transfer, %	.0746	.0169	.0114

TABLE 2 (CONT.)

PEAK PERFORMANCE DATA FOR THE THREE THERMOELECTRIC CONDENSERS SHOWN IN FIGURE 2

(c) Power Generation Data

Unit Designation	10.27T	9.03	3.97
Hot junction temperature, °K	321.1	322.4	321.4
Cold junction temperature, °K	314.9	318.3	319.7
Carnot cycle efficiency, %	1.924	1.279	.532
Conversion efficiency, %	.2235	.1490	.0618
Net power/heat transfer, %	.1488	.1321	.0504
Friction power, MW	1.538	.350	.236
Net power, MW	3.066	2.726	1.04
Friction power/electric power	.3340	.1137	.1851
Power intensity, Watt/ft^2	13.89	8.526	3.47
Maximum power density, KW/ft^3	16.67	10.23	13.9
Thermoelectric material volume, ft^3	184.0	266.4	74.74

It appears from the comparison of these data that 10.27T has the best performance with the highest electrical output. This configuration has the smallest hydraulic radius and plate spacing. To find a better configuration, other selections should be made. There is no reason why further improvements could not be introduced in these designs.

D-2 Power Intensity, P_i:

The power density curve has a logical design appeal for it maximizes the power output per unit volume. However, the peak of this curve occurs at a very small plate thickness and it may become too costly to slice very thin couples. The designer then may choose a thicker module to avoid the high manufacturing cost. The power intensity curve can be used as a guideline in this situation. The power intensity curve is obtained from dividing the net power output per module by the total module surface area. A design that is based on maximizing power intensity relaxes some of the emphasis that might be unduly placed on the saving of material in return for a possible gain in the total power generated.

The plots of P_i against the material volume is seen in Figure 3. These plots, too, exhibit definite peaks but the peaks occur at optimum plate thicknesses $t_i > t_d$ as indicated by the markers on the curves.

It is interesting to note that t_i and t_d could be used in a design to define two limits for a possible module thickness, t. If t is made more nearly equal to t_d, then the greatest power per unit volume of the thermoelectric material is obtained. Alternately, if t is made more nearly equal to t_i, maximum power per unit module area is obtained.

In an ideal situation, the surface geometry is so selected that for a given application the power density curve and the power intensity curve both peak at the same plate thickness, that is,

$$t = t_d = t_i \text{ (for the most optimum design).}$$

D-3 Net Power, P_{net}:

The plots for the total power generated and the net power for 10.27T alone are shown in Figure 4. The deviation between these plots are due to the friction losses on the water side. It is seen that the power output increases with thickness. From a 1640 MW electric power plant operating under the third conditions in Table 1, more than 12 MW can be generated with a thermoelectric condenser having a plate thickness equal to 0.2-inch. It was shown earlier that a better condenser design has a plate thickness as small as 0.01" and no greater than 0.05". The net power generated for these limits are between 3.066 MW and 8.79 MW and the material volume between 184 ft^3 and 1575 ft^3, respectively.

Additional performance data of interest on 10.27T are presented in Table 3. The data are listed for plate thicknesses ranging from 0.004" to 0.2". The items included in this table are the heat flux per unit area, net power to heat transfer ratio, conversion efficiency, friction power to heat transfer ratio, and the Carnot cycle efficiency. Attention is drawn to the five-fold variations in the heat flux, conversion and Carnot cycle efficiencies for the range of plate thicknesses indicated. The conversion efficiency approaches the net power to heat transfer ratio at large plate thicknesses. The Carnot cycle efficiency is nearly ten times the conversion efficiency and it varies between 1.058 percent at t = 0.004" to 5.015 percent at t = 0.2". The friction to heat transfer ratio varies with the plate thickness, but in a subtle way. It is most affected by the flow Reynolds Number and the associated heat transfer coefficient.

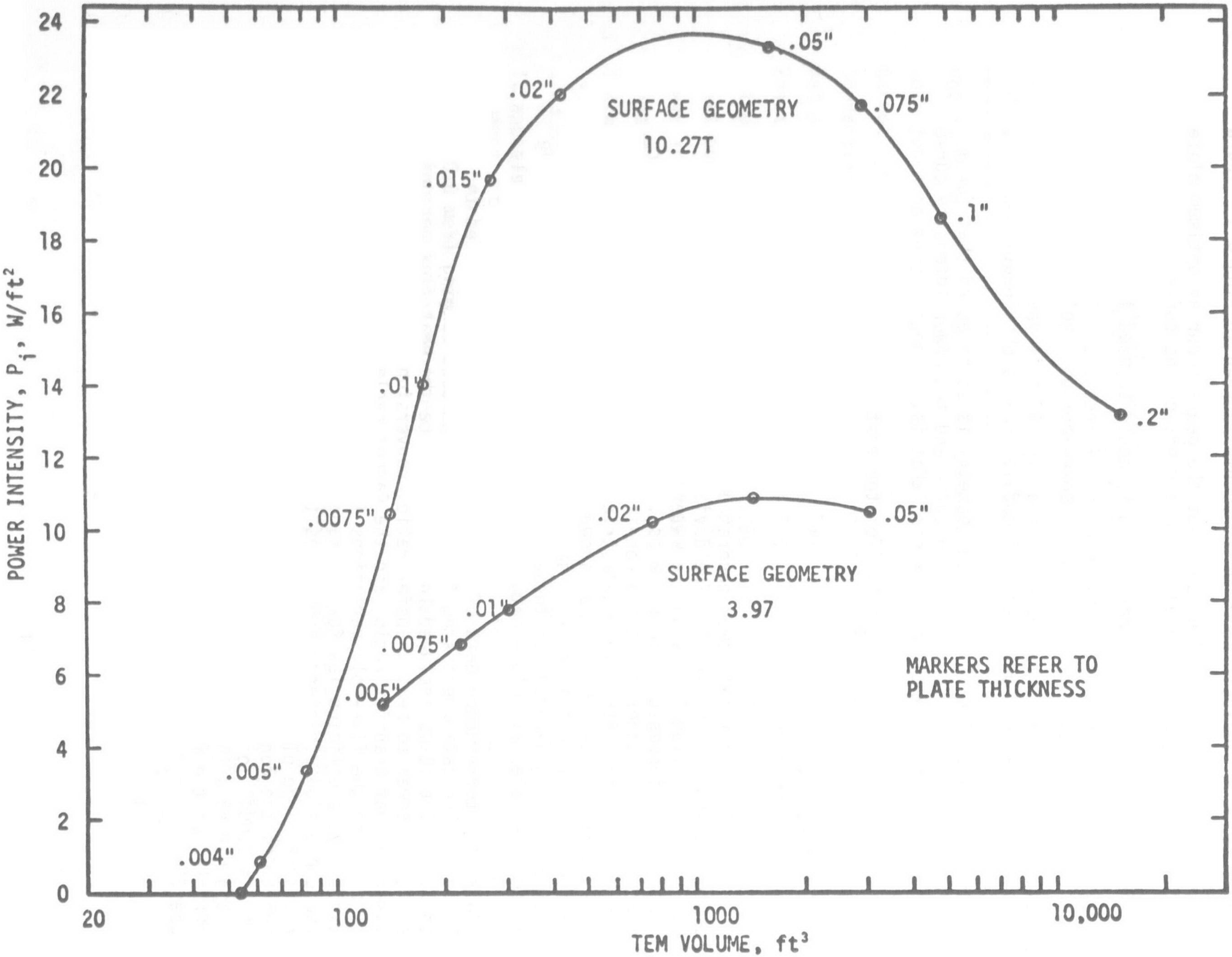


FIGURE 3. THE (NET) POWER INTENSITY FOR GEOMETRIES INDICATED

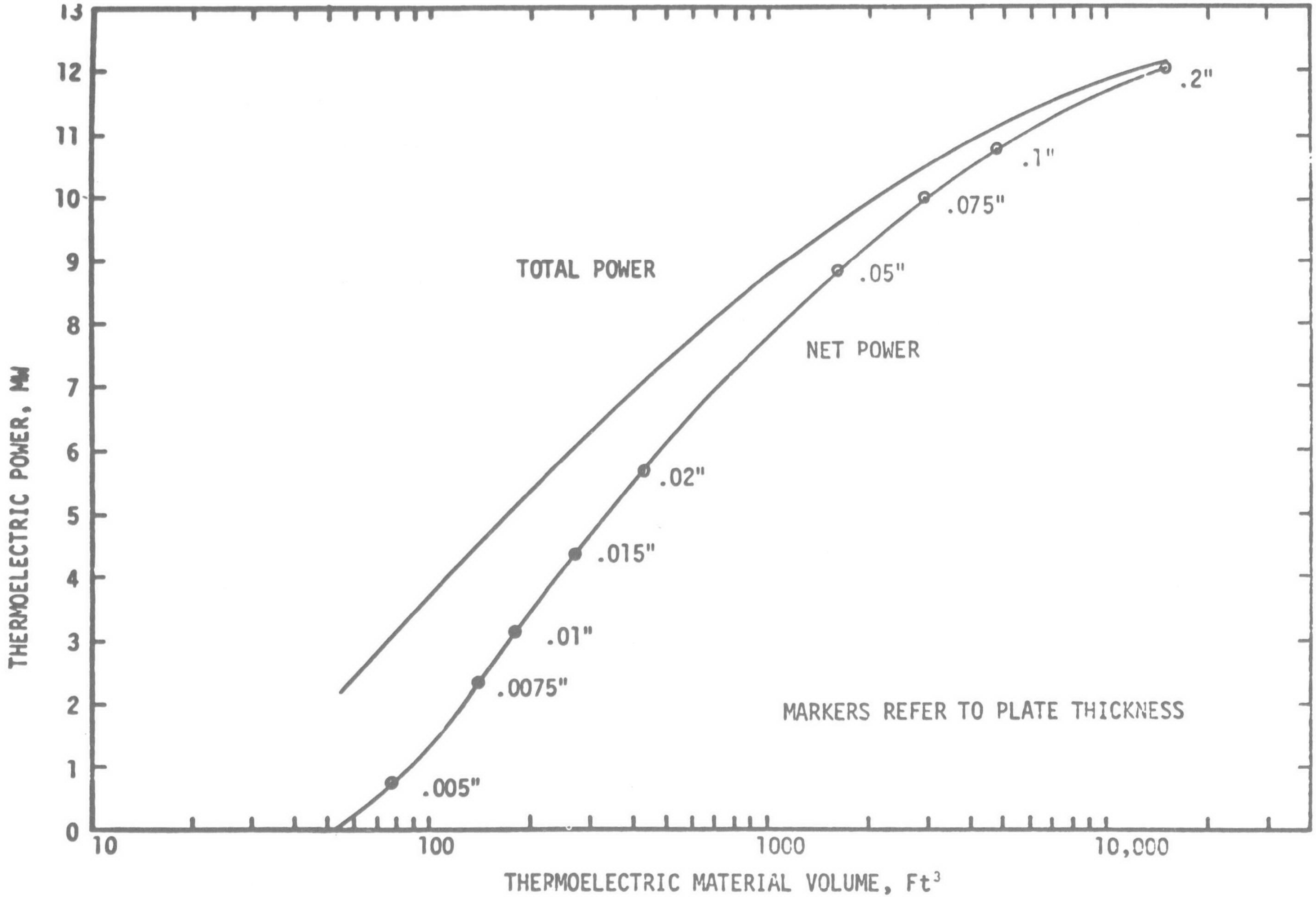


FIGURE 4. POWER GENERATED FROM THERMOELECTRIC CONDENSER 10.27T FROM THE WASTE OF A 1600 MW STEAM POWER PLANT OPERATING AT 4" Hg abs

TABLE 3

HEAT TRANSFER AND GENERATING PERFORMANCE CHARACTERISTICS
OF THE THERMOELECTRIC CONDENSER 10.27T

Plate Thickness in.	Heat Flux Q KW/ft^2	Net Power/Q %	Conversion Efficiency %	Friction/Q	Carnot Efficiency %
.004	11.169	.0097	.1224	.1127	1.058
.005	11.105	.0304	.1484	.1180	1.282
.0075	9.328	.1101	.1754	.0653	1.511
.01	9.333	.1488	.2235	.0746	1.924
.015	9.426	.2085	.3113	.1028	2.678
.02	8.185	.2700	.3400	.0700	2.919
.05	5.461	.4259	.4459	.0333	3.926
.075	4.461	.4875	.5138	.0263	4.388
.1	3.623	.5122	.5285	.0163	4.510
.2	2.271	.5803	.5885	.0082	5.015

D-4 The Effects of Turbine Back Pressure on Power Generated:

The foregoing calculations were all carried out for the third power plant with 4" Hg abs back pressure. A rough comparison of the relative magnitude of the thermoelectric power generated with back pressures 0.5" and 1.5" Hg abs can be established if the flow friction is not considered and if some arbitrary but reasonable values are assumed for the heat transfer coefficients. Accordingly, we let the water side convective heat transfer coefficient equal 750 Btu/hr ft^2 °F and that for the steam side equal 3000 Btu/hr ft^2 °F. The hot and cold side surface temperatures are then calculated and inserted in Equation (3) to find the appropriate conversion efficiencies and the related maximum power obtainable in each case. The results of these calculations are shown plotted in Figure 5. It is found that the plot for the 4" Hg back pressure closely approximates the similar plot shown in Figure 4. It is also seen that the power generated increases with back pressure. The maximum power obtainable with 4" Hg abs back pressure is 13.04 MW. With back pressures 1.5" Hg, and 0.5"Hg the maximum power is 6.28 and 4.81 MW, respectively.

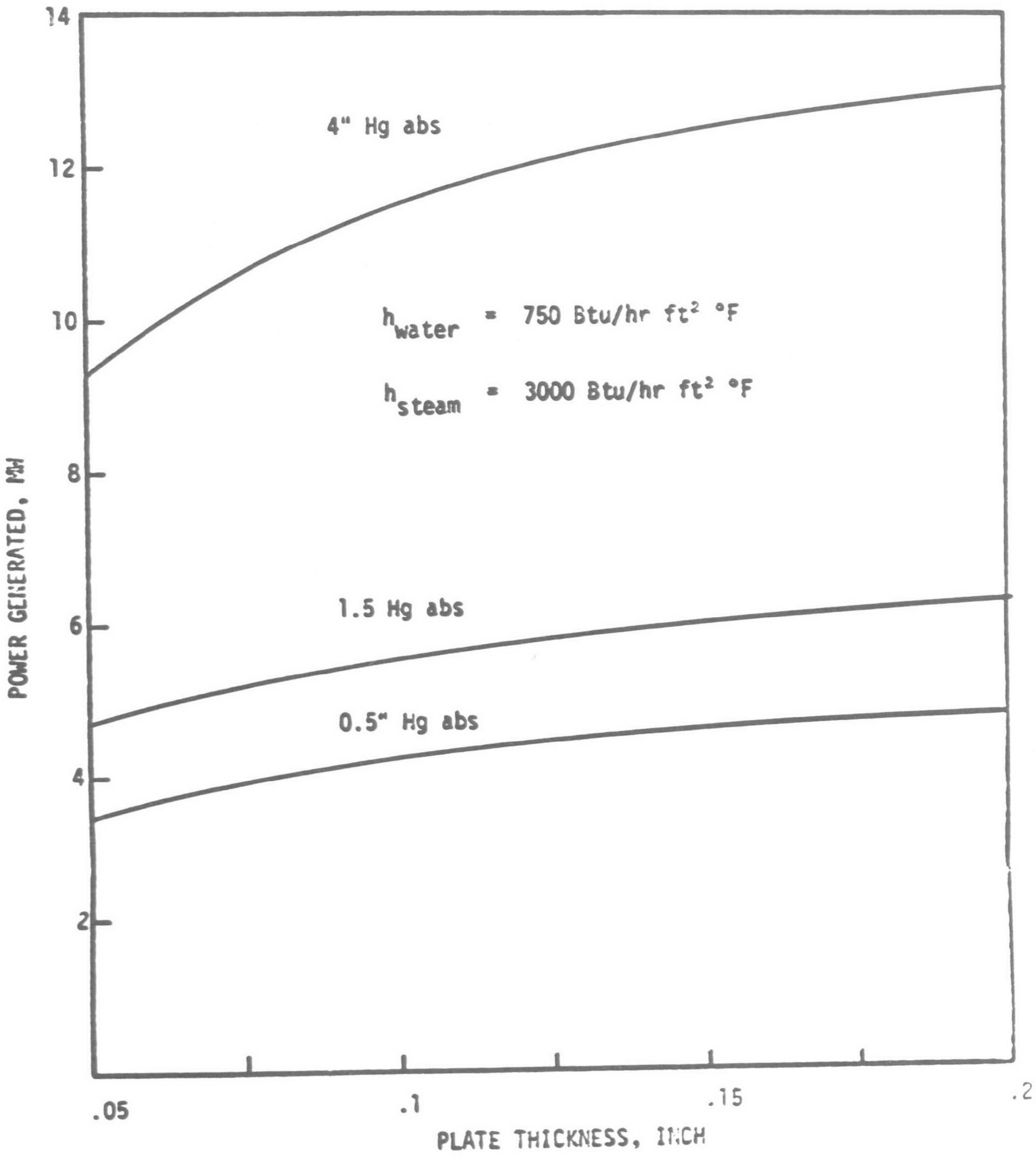


FIGURE 5. MAXIMUM THERMOELECTRIC POWER GENERATED AS A FUNCTION OF TURBINE BACK PRESSURE FROM THE WASTE OF A 1600 MW STEAM POWER PLANT.

D-5 The Effects of a High Figure of Merit:

The figure of merit Z_{pn} used in the previous calculations was equal to 1.85×10^{-3} $°K^{-1}$. It is difficult to assess the effects of a better thermoelectric material on the condenser power generation merely from the knowledge of the figure of merit. The thermal conductivity of the material also must be known in order to carry out the heat transfer calculations. However, if we ignore this fact for the sake of argument, we find that if Z_{pn}-factor is doubled, a 166 percent improvement in the conversion efficiency is obtained. If we triple the factor, the improvement will be 215 percent. Presumably, similar increases could be expected for the power output. This is a rough estimate, but the agrument shows that using better materials that are available with today's technology, one expects at least 20 MW power output. This is equal to 1.2 percent of the plant capacity.

SECTION IV

THERMOELECTRICITY FROM COMBUSTION PRODUCTS

There is a substantial quantity of heat released to the atmosphere as a result of burning fossil fuels for power generation. Usually, air at an ambient temperature enters the process to enable combustion, but when released, the exhaust is inadequately cooled and thus carries with it a portion of the combustion energy. Consider a typical 1000 MW fossil-fueled steam power plant. The stack gas temperature ranges between 250°F and 300°F. It can be shown that the waste heat carried away through the stack is between 100 MW and 150 MW. An open cycle gas turbine power plant of the same capacity releases as much as 780 MW through the exhaust at a temperature of the order of 750°F. It appears, therefore, that the hot exhaust could be passed through thermoelectric heat exchangers for generating electricity as the heat is conducted through the thermocouples to the ambient air on the cold side of the exchanger.

Let us assume an ambient air temperature of 70°F and calculate the maximum conversion efficiency based on the inlet temperature conditions from Equation (3). The efficiencies so obtained are 5.7 percent and 10.8 percent for the stack gas and gas turbine conditions, respectively. Clearly, these relatively high efficiencies are impossible to realize in a thermoelectric heat exchanger. The major difficulty is maintaining a high differential junction temperature across the module. Since the fluid film resistance is much greater than the plate resistance, the principal temperature drop occurs across the film and not the plate. Unless the plate is made extremely thick, it will offer only a minor resistance to the passage of heat, thereby maintaining only a small temperature differential between the hot and cold junctions. This reduces the conversion efficiency drastically. A sample calculation was carried out for plate thicknesses between 0.01-inch and 0.5-inch. At a plate thickness equal to 0.01, the efficiency dropped two orders of magnitude from the theoretical limit. It dropped to within one-half of this limit at a plate thickness equal to 0.5". In these calculations, the convective heat transfer coefficients on both air and gas sides were equal to 45 Btu/hr ft^2 °F. Initial pressures equal to 15 psia were assumed for both fluids. Under these conditions, the friction power was at least an order of magnitude greater than the thermoelectric power generated.

Several calculations were carried out at higher inlet air and gas pressures. Progressively better results were obtained as the initial pressure was increased. The pressure at which the friction power and the generated power became of the same order of magnitude was about 100 psia for the stack gas conditions and somewhat below that for the gas turbine exhaust conditions. The plate thickness in these calculations was held at 0.05-inch. Better results are expected with thicker plates. It is of some interest to note that the friction power to heat transfer ratios in the latter calculations were less than 0.5 percent. This is a reasonable number for conventional regenerators.

SECTION V

COST ANALYSIS

An attempt is made to compare the cost of generating electricity by conventional methods with the cost of generation by thermoelectric condensers. Only a rough estimate of the latter cost is possible, primarily because thermoelectric material supplies in large quantities have not been in demand and thus are not presently available. The cost of manufacturing large quantities must be extrapolated from the present level of availability of the raw materials and the processing technology. The manufacturing cost of large scale modular units to fit thermoelectric condensers must be estimated in a similar way. The cost estimate presented here is furthermore biased by the particular choice of the heat exchanger geometry. It may be possible to select other surface geometries that combine effectiveness for heat transfer with the ease in manufacturing.

The capital cost, the annual fixed charges, and the annual operating costs for conventional steam power generation systems were obtained from Reference 5. The capital cost of $150/KW based on 1970 estimates was used. Other cost items were 13.55 percent fixed charges and 2.97 Mills/KWH operating cost with 2.5 Mills/KWH for fuel. A thirty-year operating life with 6000 hours of operation per year was assumed. An interest rate of 8 percent was charged as a part of annual fixed costs.

The capital cost for thermoelectric power generation is estimated below. There were no fuel charges in this case, but all other cost items were computed at the rates just mentioned for the conventional steam power generation.

The capital cost estimate for the construction of thermoelectric condensers was based on the cost of Bismuth Telluride Alloy. An order of magnitude estimate of the cost of this material as obtained from Reference 6 was $16 per pound, or $6,912 per cubic foot. In the process of cutting very thin slices, some material may be lost, but since very large quantities are involved, such losses could be subjected to recycling. There is some manufacturing cost penalty for slicing very thin pieces that is difficult to assess and is not included in the cost estimate. Plate thicknesses of 0.05" are relatively easy to achieve (6). The cost of producing thinner plates could have penalty factors approaching the cost of the material itself for each 0.01" reduction.

If we arbitrarily allow a uniform cost factor equal to the material cost for manufacturing the modules (i.e., cutting, bonding, etc.) and an additional factor for constructing the condenser, the condenser would cost $20,736 for each cubic foot of thermoelectric material that goes into it.

In the previous analysis, we have shown the thermoelectric power output as a function of the thermoelectric material volume that goes into the condenser. A capital cost estimate based on these data and for three plate module thicknesses of 0.01", 0.02", and 0.05" are thus estimated at $1250/KW, $1560/KW, and $3720/KW, respectively.

Detailed cost estimates for a condenser of a plate module thickness equal to 0.01" are listed in Table 4. In this table, the annual fixed charges do not include the 8 percent interest rate. The present worth cost was obtained from the amortized 30 years annual fixed and operating charges. Note the relatively low annual operating cost of thermoelectric generation from waste heat due to the absence of fuel requirements. Based on its present worth cost, the latter is nearly five times costlier than the steam electric generation despite the low annual operating cost just mentioned.

TABLE 4

COMPARATIVE COST OF THERMOELECTRIC GENERATION (Plate Thickness .01") WITH STEAM-ELECTRIC GENERATION BOTH FOR 3.066 MWe

Cost ($10^6$$)	Fossil $150/KW	Thermoelectric $1250/KW
Annual fixed charges	0.026	0.218
Annual operating charges	0.055	0.009
Present worth estimate	0.912	2.56
Capital cost	0.460	3.83
Total present worth	1.37	6.39

The cost of thermoelectric generation with plate module thicknesses equal to 0.02" and 0.05" were also calculated and the results together with those for plate thicknesses of 0.01" are listed in Table 5. The cost items in this table are plant capital cost and investment, present worth cost and the total present worth, all given for thermoelectric/steam-electric generation. It is shown that this cost ratio increases with plate thickness. In particular, the ratio of the total present worth costs for a plate thickness of 0.05" is as high as 14.

TABLE 5

COMPARATIVE COST OF THERMOELECTRIC
GENERATION/STEAM ELECTRIC GENERATION

Cost	Plate thickness (in.) (Plant capacity, MWe) .01" (3.066 MWe)	.02" (5.607 MWe)	.05" (8.787 MWe)
Capital cost ($/KW)	1250 / 150	1560 / 150	3750 / 150
Present worth cost ($10^6$$)	2.56 / 0.912	5.79 / 1.67	21.3 / 2.61
Capital investment ($10^6$$)	3.83 / 0.460	8.75 / 0.841	32.7 / 1.32
Total present worth ($10^6$$)	6.39 / 1.37	14.5 / 2.51	54.0 / 3.93

SECTION VI

CONCLUSION

The foregoing analysis was a demonstration of a method that can be applied to designing thermoelectric heat exchangers for generating electricity from waste heat. No attempt was made to select the most suitable surface geometry for the applications considered; neither was a serious attempt made to optimize the designs. We have presented order of magnitude estimates of the cost involved. Based on these estimates, at the present time, it is uneconomical to use thermoelectric devices to generate electricity from waste heat.

SECTION VII

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ACCESSION NO.

ABSTRACT: The feasibility of recovering electricity from the waste heat of electric power plants was assessed. Sources considered were: stack flue gas, gas-turbine exhaust, and condensing steam. Typical 1600 MW fossil-fuel steam plants and gas-turbine plants were used as examples. Flat plate heat exchangers were designed with thermoelectric couples arranged in series within the plates. Heat flux, conversion efficiencies, and flow friction losses were calculated. Except for the condenser application, the friction losses are several times the thermoelectric power generated. Under favorable conditions, 3 to 9 MW is obtainable from the thermoelectric condensers. The high material cost, however, precludes all such applications today.

KEY WORDS:

Thermodynamics

Heat Transfer

(Waste) Heated Water

Thermoelectric Condenser

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