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FINAL REPORT FOR  
SOLAR THERMOELECTRIC GENERATORS

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SYNCAL CORPORATION

430 Persian Drive • Sunnyvale, California 94086

FINAL REPORT FOR  
SOLAR THERMOELECTRIC GENERATORS

29 July 1977

Contract Number 954637

Prepared For  
Jet Propulsion Laboratory  
Pasadena, California

Prepared By  
Syncal Corporation  
Sunnyvale, California

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# FINAL REPORT FOR SOLAR THERMOELECTRIC GENERATORS

## I. INTRODUCTION

This report discusses the methods, the findings and the conclusions of a study for the design of a Solar Thermoelectric Generator (STG) intended for use as a power source for a spacecraft orbiting the planet Mercury. The study underlying this report consists of essentially three separate, but related areas. First, it considers several state-of-the-art thermoelectric technologies in the intended application and selects one of them as the one most likely to satisfy the intended goals of the study. Second, the study considers in detail the design of various STG configurations based on the thermoelectric technology selected from among the various technologies; the study derives a recommended STG design. Third, the performance characteristics of the selected STG technology and associated design are studied in detail as a function of the orbital characteristics of the STG in Mercury orbit and throughout the orbit of Mercury around the sun.

In order to make the present study as complete as possible, attention has been paid to three separate types of STG design concepts. The first of these, a flat-plate type STG configuration, is the standard type of STG that is not unique to this study. The flat-plate type STG design configuration has been previously considered in connection with past programs. Such consideration has addressed itself to Earth orbital as well as near sun missions. Two other design concepts considered by the present study are relatively unique, especially as regards the use of STGs in space applications. One of these concepts utilizes a compact type STG that obtains its input heat by means of a single relatively large solar concentrator. The other type concept is a combination of a flat-plate type STG design and the compact type STG design that utilizes a solar concentrator. In this design, use is made of individual solar concentrators for each thermopile comprising the STG. This particular STG design concept combines the advantages of both the flat-plate type and the compact type STG designs. In addition to these

three basic STG design concepts, the present study has also considered in a cursory manner other concepts. One of the more interesting of these is a cylindrically configured STG which obtains its heat from a centrally located heat source in the form of a heat pipe. The heat pipe serves as a heat transfer member between a single solar concentrator and the hot junctions of the thermopiles. The solar concentrator is located outside of the envelope that defines the STG.

Another innovative feature of the present study is the use of multithermo-couple thermopiles. All previous STG studies have considered the use of thermopiles that consist of individual thermocouples. The present study considers the use of thermopiles that consist of a sufficient number of thermocouples to provide the voltage output required of the STG. This enables the parallel connection of all thermopiles within the STG and thereby provides considerable electrical redundancy to the overall system. It must be mentioned that this latter design feature of the STG only applies to the silicon-germanium alloy technology and not to any of the other thermoelectric technologies considered by the present study. As will be seen below, it is the silicon-germanium technology that offers the greatest number of advantages to a STG design for use in connection with a Mercury orbiting spacecraft.

In order that the present report be as complete as possible, it includes the detailed mathematical models used in the basic design calculations for each type of STG considered by the study. The mathematical model used for the flat-plate STG is a modification of a model previously developed\*. The mathematical models used for the other STG design concepts have specifically been developed on the present study.

As a point of reference it should be mentioned that the STG designs considered by the present study are formulated on the assumption that the STG is required to provide 300 watts of electrical power at a load voltage of 32 volts when operating at 0.45 AU from the sun, the greatest distance attained by Mercury while it orbits the sun.

\* V. Raag, Energy Conversion, 8, 169 (1968).

## II. CONCEPTUAL AND MATHEMATICAL STG MODELS

The present study has addressed itself in detail to three basic types of STG design concepts. Another type of STG design concept has been investigated in somewhat less detail. The three design concepts considered in detail include a planar flat-plate STG design, a compact STG design that utilizes a single solar concentrator and a distributed panel-like STG design that utilizes a separate solar concentrator for each thermopile. The STG design concept considered in less detail includes a cylindrical STG which contains a central heat source in the form of a heat pipe. The heat pipe extends beyond the cylindrical envelope and is heated by means of a solar concentrator. Each of these basic design concepts is amenable for use with practically any of the existing thermoelectric technologies. Although differences exist between STGs using different technologies, these differences can be easily accommodated within the mathematical models developed as a part of the present program; such differences are taken into account by assigning different thermal and electrical property values to the thermoelectric material and by using appropriate values for hot and cold side structural members within the STG. In a few cases, it has also been necessary to include or delete certain structural members from the STG design. The mathematical models presented in this section are quite general and, as such, address themselves to the mechanics of the calculational sequence, without necessarily belaboring themselves with all of the structural finepoints of each STG design. Such detail has been included in the actual calculations, whenever appropriate.

### A. Flat-Plate STG Design

The flat-plate STG design concept assumes the STG to consist of a great number of identical cells, with each cell containing a hot and a cold side heat exchanger, a thermopile, thermal insulation and interface components used in attaching the heat exchangers to the thermopile. In this design concept, the hot and the cold side heat exchangers

possess identical areas. The flat-plate STG design concept is schematically illustrated in terms of a unit cell in Figure 1. As can be seen in Figure 1, the thermopile only occupies a relatively small amount of the total volume contained in between the hot and the cold side heat exchangers. Inasmuch as it is obviously desirable to minimize the amount of direct heat transfer between the heat exchangers, the void volume is normally filled with thermal insulation. In the present study it is assumed that this thermal insulation is of a fibrous-type, such as one of the min-K series insulations. The thermopile within each unit STG cell consists of a monolithic structure that contains a sufficient number of individual thermocouples to provide the direct output voltage required of the STG. This permits the parallel connection of all of the thermopiles within the STG and thereby provides an extreme amount of redundancy for the STG. The power produced by each thermopile is normally in the one or two watt range. This type of thermopile construction, although used for most of the parametric design studies of the present program, strictly applies to only the technology that uses silicon-germanium alloy thermoelectric material. If other materials are used, it becomes necessary to treat each thermopile as consisting of a single thermocouple. The reason for this is that thermoelectric technologies other than that based on silicon-germanium alloys do not conveniently lend themselves to thermopile construction of the type here described. This means that the single thermocouple thermopiles of technologies other than those using silicon-germanium alloys not only provide a relatively small power output, but also provide a very low output voltage. This eliminates the possibility of interconnecting all of the thermopiles in a parallel arrangement and thereby permits considerably less STG redundancy. The mathematical model enables both thermopile concepts to be conveniently included by means of the assignment of an appropriate number of thermocouples per thermopile. The hot and the cold side heat exchangers

Cross-Section of Flat-Plate  
STG Cell

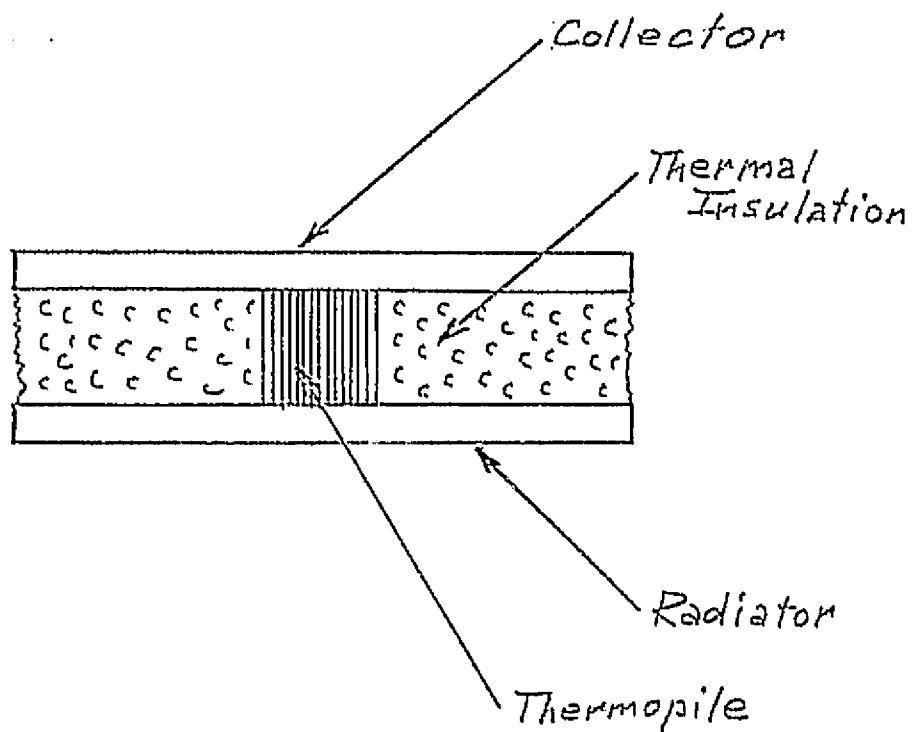


Figure 1

are assumed to possess on their outer surfaces special coatings. The characteristics of the coating on the hot side heat exchanger are such that it possesses a very high value of absorptivity and a very low value of emissivity. The opposite is true of the coating on the cold side heat exchanger. In the former case, a coating with the stated characteristics maximizes the amount of incident heat absorbed by the heat exchanger and minimizes the amount of heat reradiated. The characteristics of the coating on the cold side heat exchanger assure a maximum amount of heat rejection and a minimum of heat absorption. Although in most space applications, the absorptivity characteristics of a cold side heat exchanger are unimportant because usually the heat exchanger faces outer space, in the present case this is not so because the cold side heat exchanger faces the planet Mercury during at least a portion of its orbit around the planet. It is important therefore to minimize the amount of heat absorbed from the planet.

The total heat transported through the flat-plate STG,  $Q_T$ , may be expressed by

$$Q_T = \alpha W A_C - \sigma \epsilon_C A_C [T_H + \gamma \Delta T]^4, \quad (1)$$

where  $\alpha$  is the absorptivity of the hot side heat exchanger,  $W$  is the incident solar flux,  $A_C$  is the total surface area of the hot side heat exchanger,  $\sigma$  is the Stefan-Boltzmann constant,  $\epsilon_C$  is the emissivity of the hot side heat exchanger,  $T_H$  is the hot junction temperature of the thermocouples,  $\gamma$  is the fractional temperature drop between the hot junction temperature and the temperature of the hot side heat exchanger and  $\Delta T$  is the temperature difference between the hot and cold junctions of the thermocouples. Equation (1) represents the heat balance of the STG at the hot side heat exchanger. The first term in the equation represents the amount of heat absorbed by the heat exchanger and the second term represents the amount of heat reradiated by it. The difference between these two terms, of course, represents the amount of heat that must traverse the STG. The heat balance at the cold side heat exchanger requires that the amount of heat transported through the STG be rejected by the cold

side heat exchanger; because some of the heat is converted to electricity, energy conservation principles require that the power produced by the STG be subtracted from the total heat transported through the STG in this heat balance relationship. Because heat is rejected from the cold side heat exchanger strictly by radiation, the expression can be solved for the cold side heat exchanger temperature,  $T_R$ , as follows:

$$T_R = \left[ \frac{Q_T - P}{\sigma \epsilon_R A_R} + T_A^4 \right]^{\frac{1}{4}}, \quad (2)$$

where  $P$  is the power produced by the STG,  $\epsilon_R$  is the emissivity of the cold side heat exchanger,  $A_R$  is the surface area of the cold side heat exchanger and  $T_A$  is the temperature of the ambient into which heat is rejected by the cold side heat exchanger. It should be noted that even though separate expressions are used in Eqs. (1) and (2) for the surface areas of the hot and cold side heat exchangers, in a flat-plate STG, by definition, these two areas are identical. This is easily accomplished in actual use of the equations by making use of identical numerical values for the areas of both heat exchangers. The cold junction temperature of the thermocouples,  $T_C$ , may be calculated from the cold side heat exchanger temperature by adding to it the temperature drop between these two components of the STG:

$$T_C = T_R + \beta \Delta T, \quad (3)$$

where  $\beta$  is the fractional temperature difference between the cold side heat exchanger temperature and the thermocouple cold junctions. Equations (1), (2) and (3) give the STG heat balance equations at the hot and cold side heat exchangers in terms of incident heat, thermocouple hot and cold junction temperatures and the effective ambient in which the STG operates. In order to relate these two heat balance relationships, it is necessary to define an internal configuration for the STG and to write an additional heat balance equation, an equation that pertains to the internal components of the STG. Prior to the expression of that heat balance relationship, the following configurational terms are defined for the internal components

of the STG in terms of a variety of input parameters. The side width of individual thermoelements,  $s$ , is defined in terms of the total cross-sectional area of each thermoelement,  $A$ :

$$s = [A]^{\frac{1}{2}} \quad (4)$$

It is noted that Eq. (4) implicitly assumes that each thermoelement possesses a square cross-section. This assumption is made for convenience. Other cross-sections could be assumed equally well. Inasmuch as the configuration of thermoelement cross-section does not have a great bearing on the results of the analyses obtained from the present mathematical model, it suffices to use the assumption here made. The cross-sectional area of inter-thermoelement insulation,  $A_{IS}$ , may be expressed in terms of the thermoelement side width and the thickness of inter-thermoelement insulation,  $\omega_{IS}$ :

$$A_{IS} = \omega_{IS} [s + \omega_{IS}] \quad (5)$$

The thermopile side width,  $S_{TP}$ , may be expressed in terms of the previously defined variables and the number of thermocouples in the thermopile,  $N$ :

$$S_{TP} = [2N]^{\frac{1}{2}} [s + \omega_{IS}] + \omega_{IS} \quad (6)$$

Total thermopile height,  $H_{TP}$ , is given by

$$H_{TP} = \ell + t_{HS} + t_{HSS} + t_{CS} + t_{CSS} \quad (7)$$

where  $\ell$  is thermoelement length,  $t_{HS}$  is the thermopile hot shoe thickness,  $t_{HSS}$  is the electrical insulation covering the thermopile hot shoes,  $t_{CS}$

is the thermoelement cold shoe thickness and  $t_{CSS}$  is the thickness of the insulation covering thermopile cold shoes. The total cross-sectional area of the thermal insulation separating the individual thermopiles of the STG may be expressed as:

$$A_S = A_R - MS_{TP}^2, \quad (8)$$

where  $M$  is the number of individual thermopiles within the STG. The temperature difference between the thermocouple hot and cold junctions is related by a heat balance equation that pertains to the internal components of the STG. Namely, this relationship pertains to heat transfer within the thermoelements, the interthermoelement insulation and the insulation that separates individual thermopiles. Several of the terms within this relationship are dependent on the temperature difference between the hot and the cold sides of the STG. As a consequence, in order to obtain an expression for the temperature difference across the thermoelements, it suffices to solve the heat balance equation for  $\Delta T$  as follows:

$$\Delta T = \frac{Q_T - \frac{NPS T_C}{nE_L} + \frac{P}{2m}}{\frac{MN}{L} \left[ Ak + A_{IS} k_{IS} \right] + \frac{NPS}{nE_L} + \frac{A_S k_S}{H_{TP}} [1 + \beta + \gamma]}, \quad (9)$$

where  $S$  is the temperature averaged Seebeck coefficient of the combined n- and p-type thermoelements within the STG,  $T_C$  is the thermoelement cold junction temperature,  $m$  is the ratio of the load to internal electrical resistance,  $k$  is the temperature averaged thermal conductivity of the combined n- and p-type thermoelements of each thermocouple,  $k_{IS}$  is the thermal conductivity of the interelement insulation within each thermopile,  $n$  is the electrical circuit redundancy of each

thermopile,  $E_L$  is the load voltage of the STG, and  $k_S$  is the thermal conductivity of the interthermopile insulation. The load voltage of the STG is calculated by means of the following equation:

$$E_L = \frac{mNS\Delta T}{n(1+m)} \quad (10)$$

The electrical power delivered to the load of the STG is calculated as follows:

$$P = \frac{n^2 E_L^2 A M}{m N k \rho R_E} \quad (11)$$

where  $\rho$  is the temperature averaged electrical resistivity of the combined n- and p-type thermoelements within each thermopile and  $R_E$  is an extraneous electrical resistance multiplier that accounts for extraneous electrical resistance within the STG. Finally, the thermocouple hot junction temperature is obtained by adding the temperature difference across the thermoelements to their cold junction temperature as follows:

$$T_H = T_C + \Delta T \quad (12)$$

Equations (1) to (12) complete the performance equations of a Flat-Plate STG. An inspection of these equations shows that STG performance can only be evaluated iteratively because a number of the equations depend on parameters that are not evaluated until a particular equation containing such parameters has already been used. For example, it is noted that Eq. (1) contains thermocouple hot junction temperature and the temperature difference across the thermocouple. And yet, at the beginning of the calculational sequence, it is not known what these temperatures are because they are only calculated subsequently by means of equations (9) and (12).

Therefore, at the start of the calculation, it is necessary to assume arbitrary values for these parameters. Once these parameters have been, in fact, calculated by means of Eqs. (9) and (12), they are then inserted in Eq. (1) at the start of the second approximation calculation. The same observation may be made about other parameters such as power and load voltage produced by the STG. In the beginning of the calculation, all geometric terms pertaining to the STG, including the number of thermocouples in a thermopile and the number of thermopiles within the STG are assigned numerical values. This is also true of the electrical redundancy and the ratio of load to internal electrical resistance. Arbitrary, albeit typical values are assumed for the thermoelectric properties because initially STG temperatures are not known and thermoelectric properties are typically temperature dependent functions. As the calculation proceeds and temperatures are calculated, these temperatures are then used at the start of each iteration to define appropriate values of the thermoelectric properties. For any assumed STG design, it is normally observed that as many as 10 to 20 iterations are required between Eqs. (1) to (12) before convergence obtains within practical criteria such as convergence of any desired parameter within one percent or less.

Inasmuch as STG geometry is an input parameter and inasmuch as Eqs. (1) to (12) enable the calculation of the performance and temperatures of that particular STG configuration, it is necessary to input a wide variety of different STG configurations in order that an optimum STG design may be determined. The parameters typically varied in such an optimization procedure include thermoelement geometry, the number of thermocouples, the number of thermopiles, electrical redundancy and the ratio of load to internal electrical resistance. Moreover, inasmuch as a solar thermoelectric generator receives its heat directly from the sun and inasmuch as input heat therefore does not present an economic factor, it is common

to optimize a STG design for minimum weight at any desired distance from the sun. In order that a weight optimization be possible, it is necessary to translate each assumed STG configuration into a corresponding weight. The optimization is then performed in terms of the so-called specific power which is simply the quotient of the power produced by the STG and its total weight. It is therefore necessary to define weight relationships for the various components of the STG in terms of its assumed geometry. Prior to this being done, however, a few additional relationships will be defined in order that the STG may be scaled for any desired voltage and power output. This is necessary because normally a STG powers a load with very specific voltage and power requirements. An inspection of Eqs. (1) to (12), however, shows that load voltage and power output are not input parameters, but are rather calculated quantities. It is therefore very likely that the values calculated for these quantities for any assumed STG configuration are not those that are required of a given application. The calculated load voltage and power output of the STG are used in conjunction with the required load voltage,  $E_{LD}$ , and required power output,  $P_D$ , along with the assumed number of thermocouples and thermopiles within the STG to determine a necessary number of thermocouples within each thermopile,  $N_D$ , and the necessary number of thermopiles within the STG,  $M_D$ , in order that the STG produce the required voltage and power. The following equations accomplish this:

$$N_D = \frac{E_{LD}}{E_L} N \quad \text{and} \quad (13)$$

$$M_D = \left[ \frac{P_D}{P} \frac{E_L}{E_{LD}} \right] M \quad . \quad (14)$$

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When the number of thermocouples within each thermopile and the number of thermopiles within the STG are modified, it will also be necessary to modify the overall surface area of the STG in order that the basic heat balance of the STG not be upset. This is simply accomplished by increasing the hot and cold side heat exchanger areas by means of the following relationship:

$$A_{CD} = A_{RD} \left[ \frac{M_D N_D}{MN} \right] A_C \quad (15)$$

Equations (13) to (15) upscale the assumed STG configuration to yield any desired voltage and power output. It is now appropriate to determine the weight of the STG and consequently its specific power. As already stated, variation of STG configuration will then enable the optimization of its specific power for any distance from the sun; the distance from the sun is introduced into the calculation, of course, by means of the solar constant,  $W$ . The total weight of the thermopiles,  $W_{TP}$ , is calculated by means of the following relationship:

$$W_{TP} = M_D \left\{ \left[ t_{HS} \delta_{HS} + (t_{HSS} + t_{CSS}) \delta_{SS} + t_{CS} \delta_{CS} \right] \left[ (2N_D)^{\frac{1}{2}} (s + w_{IS}) + w_{IS} \right]^2 + \left[ 2N_D \ell (A \delta_{TE} + A_{IS} \delta_{IS}) \right] \right\} \quad (16)$$

where  $\delta_{HS}$  is the density of the thermocouple hot shoes,  $\delta_{IS}$  is the density of the electrical insulation separating thermocouple hot and cold shoes from the hot and cold side heat exchangers,  $\delta_{CS}$  is the density of the thermocouple cold shoes,  $\delta_{TE}$  is the density of the thermoelectric material and  $\delta_{IS}$  is the density of the interelement insulation. The weight of the thermal insulation separating the individual thermopiles may be expressed as:

$$W_S = \left\{ A_{RD} - M_D \left[ (2N_D)^{\frac{1}{2}} (s + \omega_{IS}) + \omega_{IS} \right]^2 \right\} H_{TP} \delta_S , \quad (17)$$

where  $\delta_S$  is the density of the thermal insulation. The weight of the hot side heat exchanger is given by the following expression:

$$W_C = A_{CD} t_C \delta_C , \quad (18)$$

where  $\delta_C$  is the density of the hot side heat exchanger and  $t_C$  is the thickness of the hot side heat exchanger. The weight of the cold side heat exchanger is calculated by means of the relationship:

$$W_R = A_{RD} t_R \delta_R , \quad (19)$$

where  $\delta_R$  and  $t_R$  are the density and thickness of the cold side heat exchanger. The total weight of the STG is obtained by summing the various weight terms and including an extra term,  $W_E$ , that accounts for any additional weight not included in the other weight terms. The total STG weight,  $W_T$ , may thus be written as:

$$W_T = W_{TE} + W_S + W_C + W_R + W_E . \quad (20)$$

Finally, the specific power of the STG,  $P_S$ , is defined as:

$$P_S = \frac{P_D}{W_T} . \quad (21)$$

As already stated, it is the specific power that normally is the parameter that is optimized in a STG design.

## B. Compact STG Design With Single Concentrator

The compact STG design concept assumes the STG to consist of a solar concentrator, a compact thermoelectric generator that is surrounded by thermal insulation and a radiator for heat rejection. The thermoelectric generator consists of individual thermopiles arrayed in such a way that all of the thermopiles are adjacent to each other, with no separation between them. A single solar concentrator heats the hot sides of all of the thermopiles concurrently. The thermopiles that make up the thermoelectric generator are attached to the center of the radiator which normally extends considerably beyond the outer limits of the generator. The thermal insulation that surrounds the thermoelectric generator minimizes direct heat transfer between the solar concentrator and the radiator and also minimizes the amount of direct heat input into the back side of the radiator from the sun. As with the flat-plate type STG, the front side of the radiator of the compact type STG must possess a coating that enhances heat rejection by means of radiation and also minimizes the amount of heat that enters from a high radiation heat sink, such as the sun side of the planet Mercury. The solar concentrator must possess high reflectivity. The hot face of the thermoelectric generator may or may not possess a high absorptivity coating. Inasmuch as the surface area of the thermoelectric generator is generally considerably smaller than the total aspect of the solar concentrator, the reradiation of heat from the hot face of the generator is not especially important in this design concept. For this reason, considerable latitude exists in the emissivity/absorptivity characteristics of the hot face of the thermoelectric generator. It is important, however, to minimize the reflectivity of that surface in order to eliminate the reflection of the concentrated solar flux from the generator. The type

of solar concentrator used with the compact STG is subject to considerable latitude. The present study has considered three basic types of concentrator. The first is a trough-like concentrator that possesses tapered and slanting sides and has a square or rectangular cross-section. The second type of concentrator considered is a truncated cone. The third type of concentrator considered on the present study is a parabolic concentrator. Although all of these different types of concentrators have been considered, for convenience, most of the parametric analyses conducted for the compact STG design have utilized the truncated cone type concentrator. In addition to the solar concentrators, the present study has also in a cursory manner addressed itself to the use of a Fresnel lense. It must be emphasized that a number of possibilities obviously exist for the type of solar concentrator used in conjunction with the STG and that the present study has not considered the advantages or disadvantages of each type; the selection of a truncated cone type concentrator for the bulk of the studies conducted on the present program is more a matter of convenience than the result of exhaustive investigation of solar concentrators. It is envisioned that prior to the selection of any one type of concentrator for an actual STG, considerable further studies will have to be performed in terms of concentrator effectiveness, its weight and the amenability of its use in a space application. The thermal insulation assumed to surround the thermoelectric generator is a fibrous insulation of the type also assumed for the flat-plate type STG, namely an insulation of the Min-K series. The compact STG design concept is schematically illustrated in Figure 2.

The number of thermocouples,  $N$ , needed in the STG in order to produce a load voltage of  $E_L$  is given by:

$$N = \frac{n(1+m)E_L}{mS\Delta T} \quad (22)$$

## Compact STG Concept

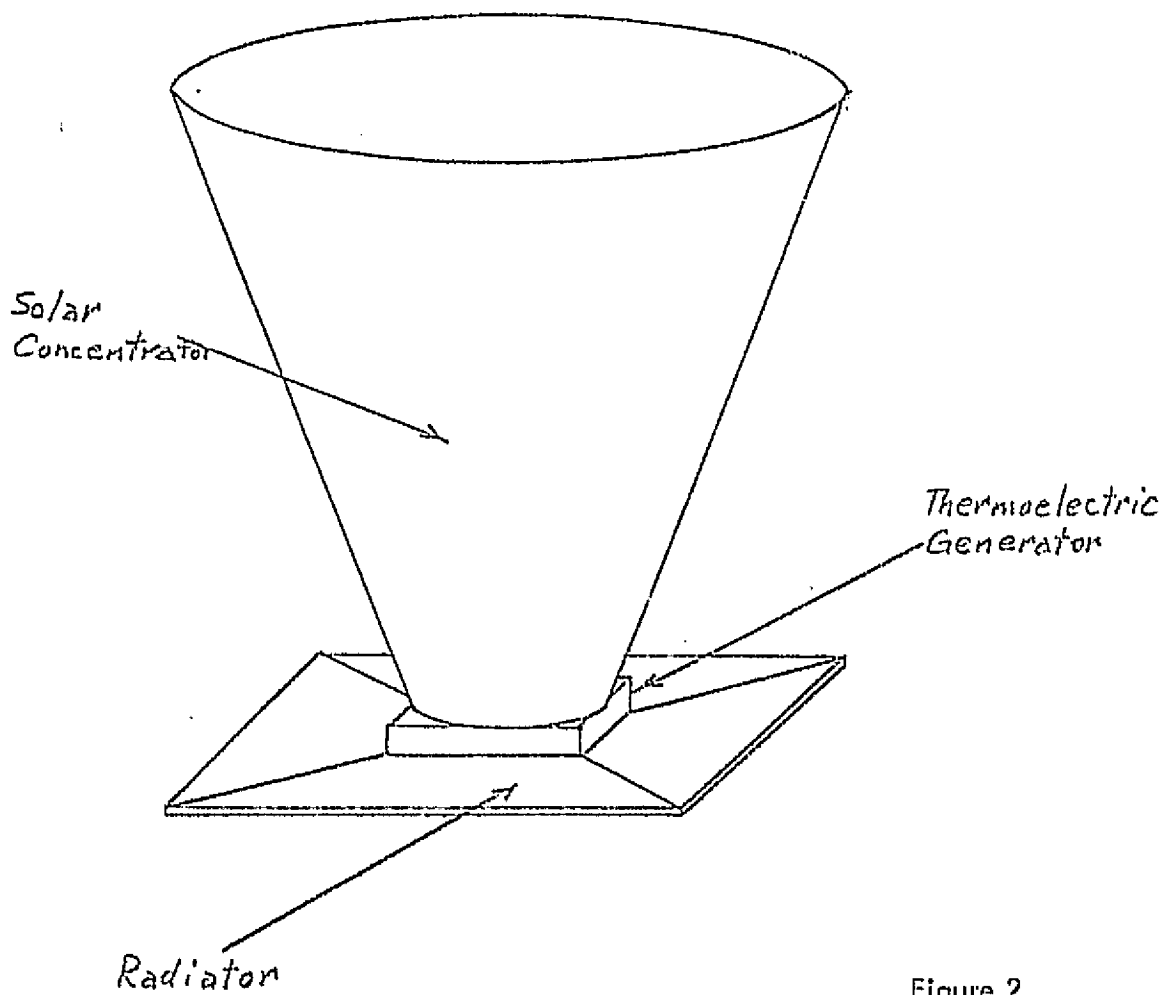


Figure 2

It is noted in Eq. (22) that the temperature difference across the thermoelements within each thermopile is an input quantity in the determination of the number of thermocouples. The ratio of thermoelement length to cross-sectional area is calculated as follows:

$$\frac{\ell}{A} = \frac{n E_L S \Delta T_M}{P(1+m) \rho R_E} \quad (23)$$

In this relationship, the power output of the STG is input as an independent parameter, as is the number of thermopiles comprising the generator. For any assumed length of the thermoelements, Eq. (23) enables the calculation of thermoelement cross-sectional area,  $A$ . A number of the other geometric parameters of the interelement insulation within each thermopile and the thermopiles themselves have already been expressed in connection with the preceding section on the Flat-Plate STG concept. These geometric parameters are subsequently used in the heat balance equations and the weight calculations of a compact STG. The electrical current flowing through the load may be expressed as:

$$I_L = \frac{P}{E_L} \quad (24)$$

The value of the load current is used in some of the following heat calculations. First, the heat conducted through the STG, including the thermoelements and the interelement insulation,  $Q_K$ , is calculated as follows:

$$Q_K = \frac{MN}{\ell} \left[ Ak + 2 A_{IS} k_{IS} \right] \Delta T \quad (25)$$

The shunt heat,  $Q_S$ , conducted through the thermal insulation surrounding the thermoelectric generator within the STG is calculated by means of the following relationship:

$$Q_S = \frac{4\omega_S [S_G + \omega_S] k_S \Delta T}{l} \quad (26)$$

where  $\omega_S$  is the width of the thermal insulation and  $S_G$  is the side width of the thermoelectric generator. The side width of the thermoelectric generator is calculated from a knowledge of the total number of thermopiles within the generator and the side width of each thermopile by means of the relationship:

$$S_G = \left[ M \right]^{\frac{1}{2}} S_{TP} \quad (27)$$

The amount of heat absorbed at the hot junctions of the thermocouples by means of the Peltier effect,  $Q_P$ , is expressed as follows:

$$Q_P = \frac{N I_L S T_H}{n} \quad (28)$$

The Joule heat,  $Q_J$ , generated within the thermocouples is taken into account by assuming that one-half of it flows to the thermocouple hot junctions:

$$Q_J = \frac{I_L^2 E_L}{2m} \quad (29)$$

A combination of the various heat terms yields the total heat required to be input to the STG. This may be expressed as:

$$Q_T = Q_K + Q_S + Q_P - Q_J \quad (30)$$

It was noted above that the calculation is performed by assuming a value of temperature difference across the thermoelements within the STG. An inspection of Eq. (28) indicates that the hot junction temperature of the thermoelements is also an assumed input variable. Using these two quantities it is now possible to define the temperature of the radiator which rejects waste heat from the STG:

$$T_R = T_H - \Delta T(1 + \beta) \quad (31)$$

Using the thus determined radiator temperature, along with the total heat required of the STG and its power output, the cross-sectional area of the radiator is immediately obtained by means of the relationship:

$$A_R = \frac{Q_T - P}{\sigma \epsilon_R [T_R^4 - T_A^4]} \quad (32)$$

If it is assumed that the radiator possesses a square cross-section, the side width of the radiator is simply the square root of the radiator area:

$$S_R = [A_R]^{\frac{1}{2}} \quad (33)$$

The effective cross-sectional area of the solar concentrator in the plane perpendicular to the solar flux,  $A_C$ , may be written as:

$$A_C = \frac{Q_T + \sigma \epsilon_G A_G [T_H + \gamma \Delta T]^4}{\alpha W} \quad (34)$$

where  $\epsilon_G$  is the emissivity of the top surface of the thermoelectric generator and  $A_G$  is the total cross-sectional area of the thermoelectric generator.

Assuming the generator to also possess a square cross-section, the latter quantity may be expressed as:

$$A_G = S_G^2 \quad (35)$$

Assuming the solar concentrator to be of the form of a truncated cone, its large diameter,  $D_C$ , may be expressed as:

$$D_C = 2 \left[ \frac{A_C}{\pi} \right]^{\frac{1}{2}} \quad (36)$$

The above equations essentially characterize the determination of an overall STG configuration for required performance. Using this configuration, it is now possible to calculate the weight of the STG. However, before this is done, it is necessary to establish a few more configurational terms for the solar concentrator. These terms do not pertain to STG performance but rather enable the determination of concentrator weight and thereby also the total weight of the STG. The small diameter of the solar concentrator,  $D_{CS}$ , may be expressed as:

$$D_{CS} = F_C D_C \quad (37)$$

where  $F_C$  is the ratio of the small diameter to the big diameter of the concentrator cone. The height of the solar concentrator,  $H_C$ , is given by

$$H_C = \frac{1}{2} [D_C - D_{CS}] \tan \theta \quad (38)$$

where the angle  $\theta$  pertains to the angle that the sides of the concentrator make with respect to the plane of the thermoelectric generator.

The weight of the solar concentrator,  $W_C$ , is calculated by means of the following relationship,

$$W_C = \frac{\pi}{2} [D_C + D_{CS}] t_C H_C \delta_C , \quad (39)$$

where  $t_C$  is the thickness of the concentrator and  $\delta_C$  is the density of the material used for the concentrator. The weight of the thermoelectric generator,  $W_G$ , is identical to the total weight of the thermopiles,  $W_{TP}$ , as given by Eq. (16) if  $M_D$  and  $N_D$  are replaced by  $M$  and  $N$ . Assuming that the thickness of the thermal insulation,  $t_S$ , is identical to the height of each thermopile,  $H_{TP}$ , as given by Eq. (7), the total weight of the thermal insulation,  $W_S$ , may be expressed as:

$$W_S = \left[ \left( \frac{D_{CS}}{D_C} \right)^2 A_C - A_G \right] H_{TP} \delta_S , \quad (40)$$

where  $\delta_S$  is the density of the thermal insulation. The weight of the radiator is given by the expression:

$$W_R = \frac{\delta_R}{2} \left\{ A_G [t_{RI} - t_{RO}] + A_R [t_{RO} + t_{RI}] \right\} , \quad (41)$$

where  $t_{RI}$  represents the thickness of the outer periphery of the radiator and  $t_{RO}$  represents the thickness of the radiator immediately underneath the thermoelectric generator. It is noted that Eq. (41) implies a tapered radiator which possesses different thicknesses in different locations; this assumption enables an approximate weight optimization of the radiator. The total weight of the STG,  $W_T$ , is obtained through the summation of the individual weight terms given above:

$$W_T = W_G + W_R + W_C + W_E + W_S \quad , \quad (42)$$

where  $W_E$  represents an extraneous weight that accounts for components and structural members not included within the above calculations of individual components weights. The specific power,  $P_S$ , of the STG is defined as:

$$P_S = \frac{P}{W_T} \quad . \quad (43)$$

### C. Multi-Concentrator Distributed STG Design

The third STG design concept combines some of the features of each of the other two designs, the flat-plate STG design and the compact STG design. In this design concept, the thermopiles are distributed across the face of the radiator much as they are in the flat-plate STG design. On the other hand, rather than using flat solar collectors as they do in the flat-plate STG design, each thermopile in the third STG design concept utilizes an individual solar concentrator. The advantage of this design concept is that the single, relatively large solar concentrator is reduced to a great number of small concentrators and therefore the STG assumes an overall panel-like configuration. At the same time, however, it retains all of the advantages associated with a STG design based on the use of a solar concentrator. The main advantage of a solar concentrator is that the reradiation area of the hot face of the STG is essentially reduced to the combined area of all of the thermopiles rather than the total area of the STG facing the sun, as it is in the flat-plate STG design. The reduction of the reradiation area enables a

significant decrease in the total cross-sectional area of the STG and therefore results in a more compact and a lighter weight system. It is especially the last consideration that is important because it is weight that is the primary optimization criterion in the design of components intended for use on spacecraft.

The mathematical model to be used in conjunction with the distributed type STG design that uses a large number of individual solar concentrators is essentially identical to that of the compact STG design discussed in Part B of this section. In fact, exactly the same model can be used if the number of thermopiles,  $M$ , is set to unity and if the power output,  $P$ , is divided by the number of thermopiles that it is intended to use in the total STG. The latter number can be selected randomly in such a way that the power produced by each thermopile is set at some reasonable value or it can be determined through an optimization of the specific power of the STG in terms of the number of thermopiles contained by it. The use of the previously defined mathematical model thereby enables the optimization of a unit cell or distributed type STG that uses a great number of individual solar concentrators. The overall STG design is determined by using the required number of such cells to provide the desired total output power of the STG. This STG design is schematically illustrated in Figure 3.

### Cross-Section of STG Unit Cell

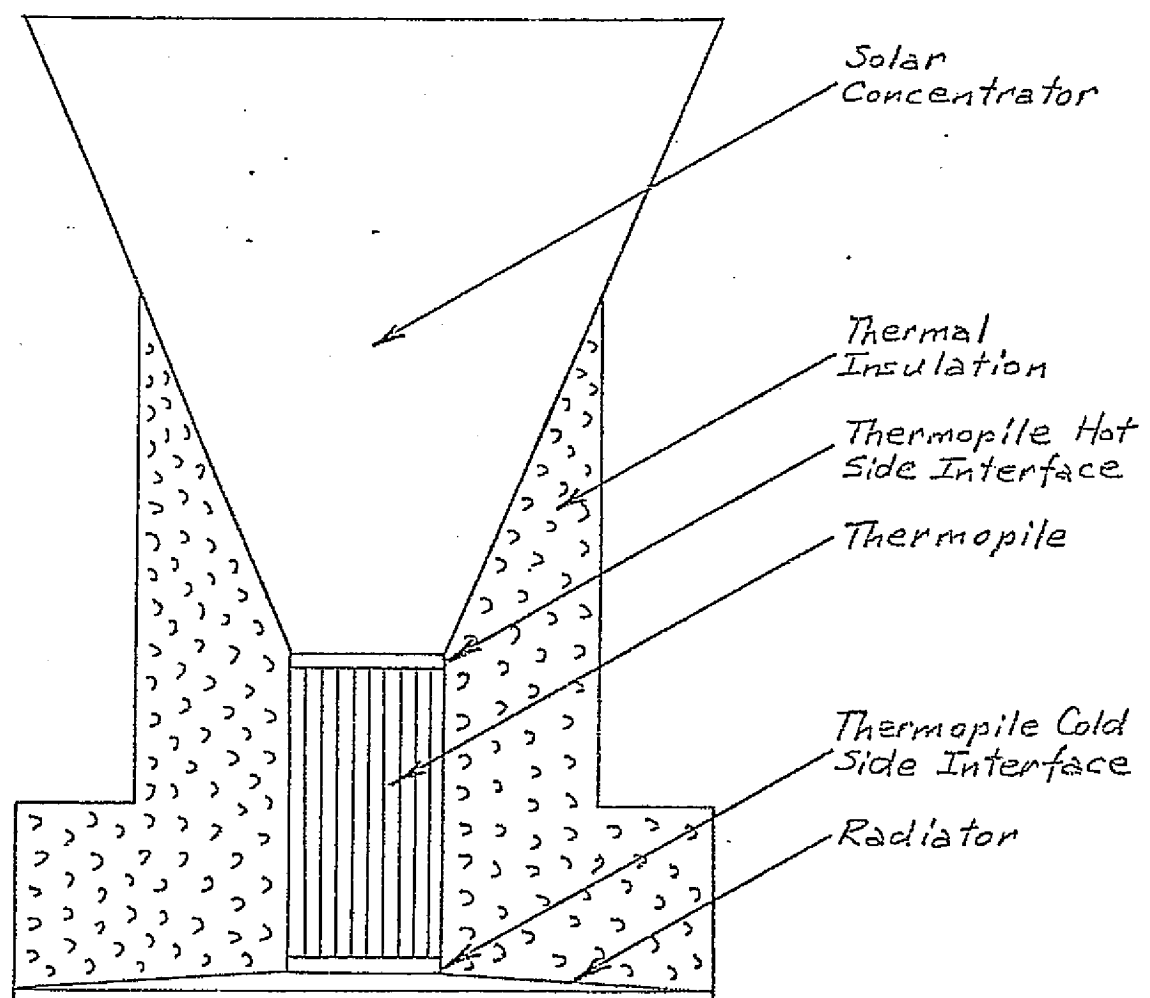


Figure 3

### III. THERMOELECTRIC TECHNOLOGY SELECTION

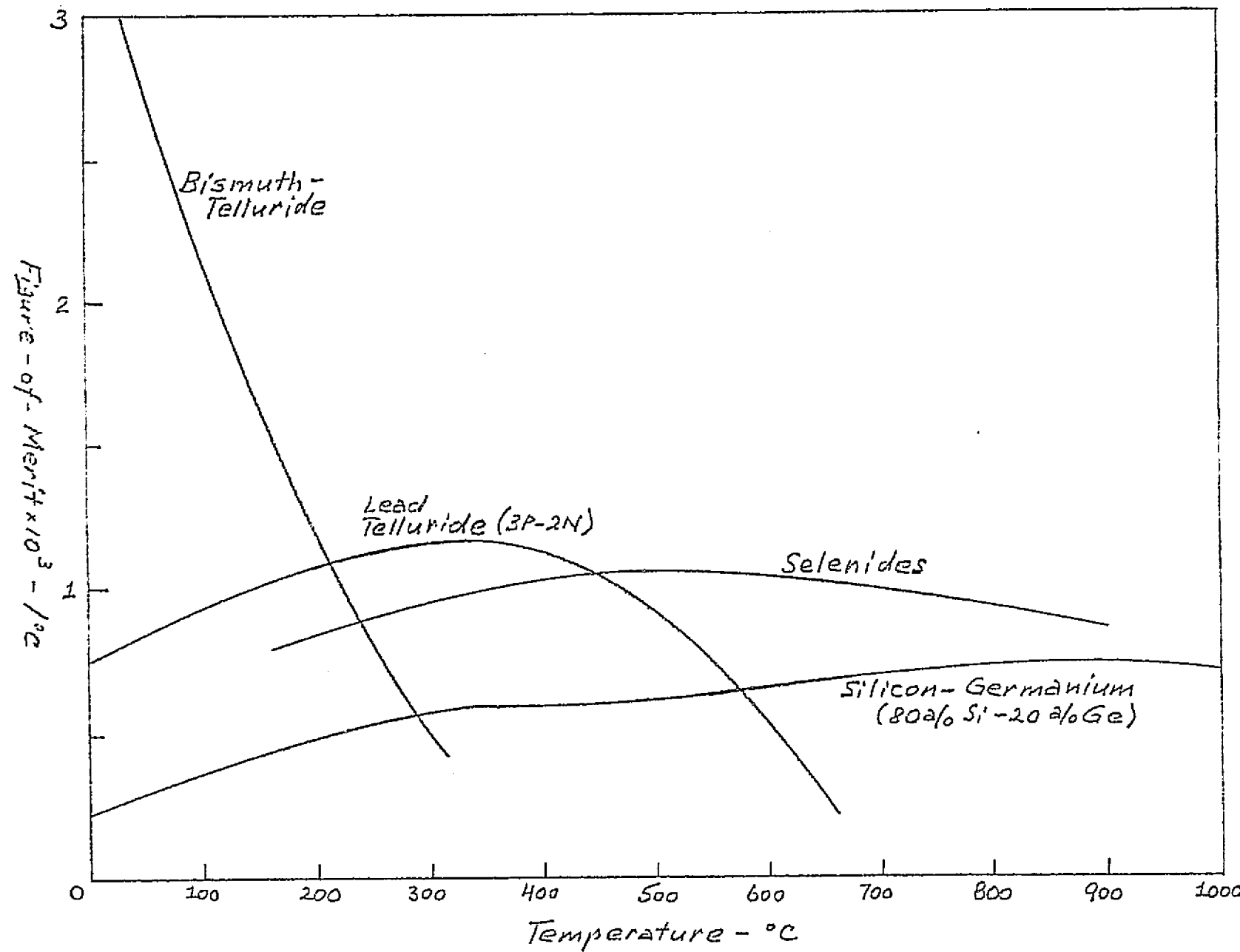
A number of different thermoelectric technologies exist and are currently in use in various thermoelectric energy conversion applications. Probably the three most common of such technologies involve the use of either bismuth telluride, lead telluride or silicon-germanium alloys. A fourth technology, that based on copper and gadolinium selenide, is presently in development and should be available for practical use within the foreseeable future. Each of these thermoelectric technologies is characterized by a range of operating temperatures under which the material can be considered useable. Considerable differences exist in the temperature ranges of useability of these materials. For example, bismuth telluride is useful at temperatures around room temperature and up to some 300°C. Lead telluride can be used from room temperature up to about 500°C if it is not hermetically sealed in order to prevent material loss by sublimation; if sealed, the material may be used up to about 600°C. Silicon-germanium alloys are useful from room temperature up to 1000°C. Although the selenides are not useable as at high temperature as are silicon-germanium alloys, they do exhibit an inherently higher capability of converting heat to electricity, i.e. they are inherently more efficient than the silicon-germanium alloys.

The conversion efficiency of a thermoelectric material can be approximately expressed by:

$$\eta = \frac{1}{4} Z \Delta T \quad , \quad (44)$$

where  $Z$  is a quantity known as the figure-of-merit that solely depends on basic material characteristics, and  $\Delta T$  is the temperature difference across which the material is operated. The most widely used thermoelectric materials are depicted in Figure 4 in terms of their figures-of-merit as a function of temperature. It is noted in Figure 4 that bismuth telluride possesses the highest figure-of-merit

Figure 4



of any of the commonly used thermoelectric materials. Inasmuch as its useful temperature range, however, is relatively small, an inspection of Eq.(44) indicates that the conversion efficiency available with this material is only of the order of four to five percent, and even this only if the cold side of the material is operated essentially at room temperature. As will be apparent below, in a space system, which is normally optimized for minimum weight, it is impractical to operate a thermoelectric system with its cold side much below 300 to 400°C. For this reason it is concluded immediately that bismuth telluride is not a viable candidate for use in a STG. Essentially the same conclusion can be drawn in regard to lead telluride. Even though the material is capable of higher temperature operation than is bismuth telluride, its useful range of operating temperatures in a STG spans only some 200 to 300°C. Even though this range of operating temperatures enables the obtainment of conversion efficiencies approximately comparable to those possible with the best bismuth telluride systems, this can only be accomplished by maintaining the cold side temperature at relatively low values. At those cold side temperature values, the overall system weight is not comparable to that possible with the remaining two thermoelectric materials, namely the selenides and silicon-germanium alloys. For this reason lead telluride can also be rejected as a possible candidate for use in STGs.

Having rejected bismuth telluride and lead telluride as non-competitive in a STG application, it remains to consider the two other thermoelectric materials, namely the selenides and the silicon-germanium alloys. Inasmuch as both of these materials are capable of fairly high temperature operation, it is necessary to consider both materials in some detail prior to the selection of one as the material on which to base the remainder of the study. As already noted, silicon-germanium alloys are capable of operation up to temperatures of 1000°C. The selenides are normally considered to be capable of operation up to about 800°C, although efforts are being made to use this material at temperatures up to 900°C. If possible, this can only be done by using sublimation inhibition coatings

on the material. Even though the use of the selenides at  $900^{\circ}\text{C}$  is questionable, the present consideration of thermoelectric materials does include that temperature, as well as the temperature of  $800^{\circ}\text{C}$ , as possible hot side temperatures of STGs based on selenides. For comparison, silicon-germanium alloys are considered at both of these temperatures, as well as at the temperature of  $1000^{\circ}\text{C}$ . It should be noted that the operation of silicon-germanium alloys at  $1000^{\circ}\text{C}$  has been repeatedly proven in practice and therefore no question exists as to the usability of silicon-germanium alloys at all of the hot side operating temperatures considered in this study.

Using simplified STG design concepts, parametric performance and weight analyses were conducted on STGs based on silicon-germanium alloys and the selenides. These analyses were performed for a STG operating distance of 0.45 AU from the sun because it is this distance that represents the worst-case operating environment for the STG in the contemplated Mercury orbiter mission. The simplified design concepts are essentially based on a STG that uses a solar concentrator, such as the STG represented by the mathematical model in Section II B. The simplified model assumes a solar concentrator, a thermoelectric converter, surrounding thermal insulation and a radiator. Extraneous structural members were not included in this preliminary comparative analysis because the intent of the analysis is only to select a thermoelectric technology which will then be subjected to detailed analyses which consider all structural members in detail. It was assumed for the purposes of the preliminary analysis that STGs using both technologies must be capable of producing 300 watts of electrical power at a load voltage of 32 volts when operating at 0.45 AU from the sun. For convenience of analysis, it was assumed that neither system experiences electrical losses; this assumption tends to penalize the silicon-germanium alloy system because it is this system that generally exhibits lower electrical losses. It should also be noted that the neglect of structural members from the weight analysis also penalizes the silicon-germanium alloy system because unlike the selenides, in

a STG application, silicon-germanium alloy thermopiles act as their own structural members without the need for extraneous supporting components. The selenide system, on the other hand, requires fairly elaborate extraneous structural support because the selenides cannot be metallurgically bonded and are normally used in conjunction with springs and various retainers. The simplified analysis, therefore, is interesting from the standpoint of showing the relative merits of the two technologies, with a heavy bias towards the selenides. If, in fact, the silicon-germanium alloy STG concept proves more advantageous under these conditions, then there is absolutely no question as to its superiority because in actual practice the superiority will be considerably greater than here illustrated. This is especially true in view of the far superior mechanical characteristics of silicon-germanium alloys and their insensitivity to operating environment and fabrication techniques. This latter factor of course lends itself towards enhanced system reliability. The results of the aforementioned parametric analyses on the two types of STG are illustrated in Figure 5 in terms of specific power as a function of STG cold side operating temperature for various values of hot side operating temperature. Figure 5 shows the results for both the silicon-germanium alloy as well as the selenide STG. The specific power has been normalized to 100 percent for the maximum value found throughout the analysis because actual specific power values are somewhat meaningless in view of the simplifying assumptions. It is noted in Figure 5 that the analyses for the silicon-germanium alloy STG have been performed at the three hot side operating temperatures of 800, 900 and 1000°C. The analyses for the selenide STG have been performed at 800 and 900°C. For each hot side operating temperature for both technologies, the specific power optimizes for a definite cold side operating temperature in the range of about 300 to 450°C. For the same hot side operating temperature, it is noted that the specific power of the silicon-germanium alloy STG optimizes at a higher cold side temperature than it does for the selenide STG. Similarly, it is noted that at identical hot side operating temperatures, the silicon-germanium alloy STG invariably results in higher optimum

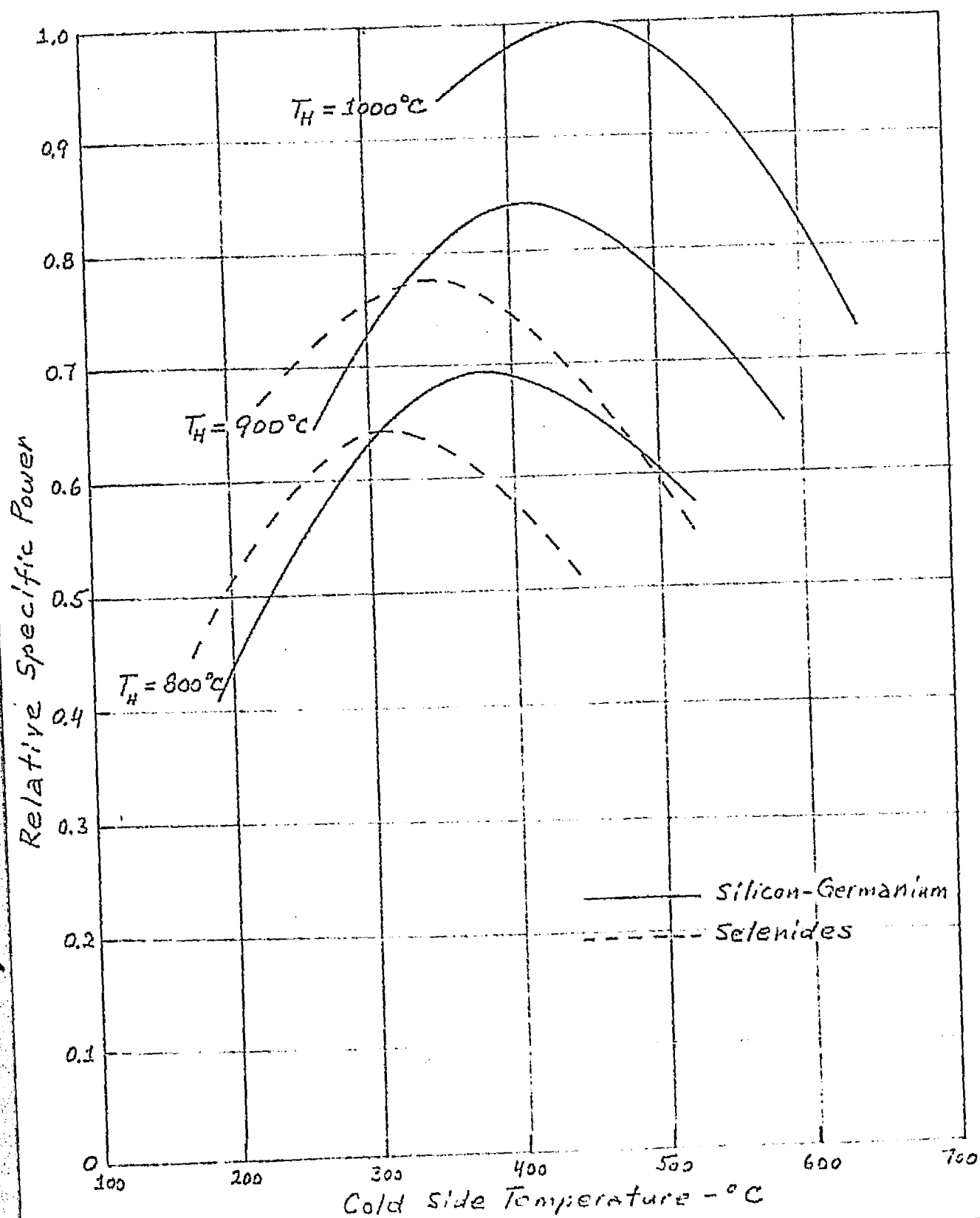


Figure 5

specific power. This is primarily due to the fact that the thermoelectric generator of the silicon-germanium alloy STG is significantly lower in weight than the selenide generator; the densities of silicon-germanium alloys and the selenides differ by more than a factor of two. In fact, even though the selenides exhibit a higher figure-of-merit and hence conversion efficiency, the consequent reduction in radiator and concentrator weights are more than compensated by the increased thermoelectric generator weight. If attention is now drawn to the fact that the silicon-germanium alloy STG is capable of operation at a hot side temperature of  $1000^{\circ}\text{C}$ , it is obvious that the optimum specific power attainable with a silicon-germanium alloy STG is far in excess of that obtainable with a selenide STG, especially if it is remembered that the results of the analysis depicted in Figure 5 are somewhat biased towards the selenides. It is of interest, and possibly somewhat surprising, that even at the same hot side operating temperature the results indicate the superiority of the silicon-germanium alloy STG.

In view of the above findings and the fact that silicon-germanium alloys have been repeatedly proven in actual practice at operating temperatures as high as  $1000^{\circ}\text{C}$ , it is obvious that a STG intended for use in a space mission should be based on silicon-germanium alloys and not on the selenides or any other currently available thermoelectric technology. For this reason, all subsequent analyses performed in detail on the present study pertain strictly to the use of silicon-germanium alloys in STGs. Silicon-germanium alloy technology is not only superior to any other thermoelectric technology when used in a STG, but it is fully developed and therefore a STG based on it may be essentially considered as state-of-the-art; no basic development effort is necessary, as it would be with the selenides, if it is decided to use such a STG in space.

IV. SILICON-GERMANIUM ALLOY STG DESIGN ANALYSES

In the preceding section it was concluded that the state-of-the-art thermoelectric technology that offers the greatest potential of a high reliability and low weight STG is that based on the use of silicon-germanium alloys. As a result of this conclusion, the bulk of all design analyses performed on the present program have been restricted to the silicon-germanium alloy thermoelectric technology. Inasmuch as even within the silicon-germanium technology, different state-of-the-art approaches exist, it was further necessary to decide which of these would be most applicable for use in a STG. The two main approaches involved the so-called Air-Vac technology in which silicon-germanium alloy thermocouples are fairly bulky and are used singly. This means that a large number of such thermocouples are required in a system that is intended to provide fairly substantial amount of power at a load voltage typically required in space applications. The other approach uses multi-thermocouple thermopiles which are monolithic units and contain a large number of silicon-germanium alloy thermocouples that possess extreme geometrical configurations; each thermocouple possesses an extremely small cross-sectional area, while its height is comparable to that used for Air-Vac type thermocouples. Even though each thermopile of this latter approach provides fairly small values of power output, in fact, values comparable to those produced by each Air-Vac thermocouple, the load voltage produced is that required of the spacecraft. This means that all of the thermopiles of the latter type can be connected in a parallel arrangement, thereby providing extreme electrical redundancy within the STG. Moreover, the thermopiles of the latter approach being monolithic units, are totally encased within an electrical insulator. This means that the possibility of electrical shorting between thermopiles is practically impossible and, moreover, the sublimation of the thermoelectric material that is sometimes experienced in high temperature operation is practically totally eliminated. Based on these considerations, it is fairly obvious that the second mentioned design approach for silicon-germanium technology is more advantageous and is therefore the

preferred design approach; namely, that in which each thermopile contains a great number of individual thermocouples. All detailed design analyses on the present program have therefore been performed with thermopiles of that type. It should be mentioned, that thermopiles of this type have previously been extensively used in a variety of thermoelectric applications, most notably in radioisotope thermoelectric generators. As a consequence, considerable practical experience exists with thermopiles of this type and it may be considered that the use of such thermopiles in a STG does not require any technology development.

The object of the present study is the design of a STG for use on a spacecraft in a Mercury orbiter mission. It is envisioned that the spacecraft is launched from Earth to Mercury and placed in a near circular orbit around Mercury in the plane of the orbit of Mercury around the sun. The orbit of the spacecraft around Mercury is assumed to have a period of 90 minutes, with the spacecraft at an altitude of 105 kilometers above the surface of the planet. Being in that type of an orbit, the spacecraft therefore will experience direct sunlight during one half of each of its orbits and will be shielded from the sun by the planet during the other half of each orbit. While the spacecraft is within direct sunlight, the STG will be facing the sun on its one side and will be facing the relatively hot surface of the planet on its other side. While the spacecraft is eclipsed by the planet, one side of the STG faces the planet, while the other side faces outer space; inasmuch as the back side of the planet Mercury is very cold in comparison to the side facing the sun, for calculational convenience, it may be assumed that both sides of the STG are facing a heat sink representative of outer space. In view of these considerations, it is apparent that during each of its orbits around Mercury, the STG is subjected to a variety of thermal conditions. Moreover, inasmuch as the orbit of Mercury around the sun is quite eccentric, varying between 0.30 and 0.45 AU from the sun, the variability of the thermal conditions imposed upon the spacecraft and the STG is even more extreme. Design analyses of STG performance must therefore address themselves to each of these conditions in order that the selected

design satisfy performance requirements under all conceivable STG operating conditions.

Most of the parametric design studies conducted on the present program have assumed steady state operating conditions for the STG at distances of 0.30 to 0.45 AU from the sun. The results of these analyses have enabled the selection of a recommended STG design. This design has then been subjected to the dynamic conditions envisioned not only during each orbit of Mercury, but also for the orbital period of Mercury around the sun. The results of the steady state parametric design analyses are discussed in this section; the dynamic performance of a selected STG design are discussed in a subsequent section. All studies have been based on the assumption that the STG must be capable of producing 300 watts of electrical power at a load voltage of 32 volts. Inasmuch as efficiency is of no importance to a STG, incident solar heat does not represent any economic considerations, all design analyses of the present study have been optimized with respect to total STG weight; it is obviously important for the STG to possess as low a weight as possible.

Power source designs are generally evolved for the worst-case operating conditions envisioned for the life of the power source. This is true of STGs as well as other types of power sources. Although during its total mission to Mercury, the worst-case operating conditions of the STG may be considered the transit period from Earth to Mercury because of the great distances from the sun, the present study considers the worst-case to be the operation of the STG at a distance of 0.45 AU from the sun. This is the farthest distance attained by Mercury from the sun and therefore during the orbital period of Mercury, the STG may be considered to produce less power at that distance from the sun than during other portions of its mission. The STG design evolved from the present study, therefore, has been optimized for operating at 0.45 AU from the sun. For completeness, however, the study has also addressed itself to all other distances between 0.30 and 0.45 AU from the sun. Moreover, for selected designs, the study has also considered the

performance of the STG at all distances between 0.30 and 1.0 AU from the sun; the latter distance, of course, represents the distance of the Earth from the sun.

In the parametric STG design studies, it is assumed that all surfaces exposed to the incident solar flux, except the surface of the concentrator, possess absorptivity values of 0.85 and emissivity values of 0.10. This applies to the solar collectors used in the flat-plate STG design as well as to the surface of the thermopiles in both the compact type and the distributed solar concentrator STG designs. The surface of the concentrator is assumed to possess perfect reflectivity. All surfaces of the STG facing away from the sun are assumed to possess emissivity values of 0.85 and absorptivity values of 0.10. In particular, these values are assigned to the radiator of each STG design.

As mentioned in the Introduction and discussed in some detail in Section II, the present study on the use of a STG in a Mercury orbital mission addresses itself primarily to three separate STG designs. These designs include a flat-plate type STG which uses flat solar collectors for each individual thermopile within the STG, a compact STG that uses a single solar concentrator for all of the thermopiles within the STG and a STG design that utilizes individual solar concentrators for each thermopile within the STG. Although the bulk of the present study has been devoted to these three STG designs, a fourth design has also been considered, although in not as great a detail as the three designs just mentioned. The fourth STG design considers a cylindrical thermoelectric generator which obtains its heat by means of a centrally located heat pipe that is attached to a solar concentrator. Whereas the results of the design analyses on the first three STG designs are presented in detail, consideration of the last mentioned design is somewhat cursory.

#### A. Flat-Plate STG Design

Using the equations given in Section II A, parametric analyses were conducted on the flat-plate STG. The parametric analyses were conducted with respect to the solar collector area of the STG, individual thermoelement cross-sectional area and thermoelement length. The

optimization parameter in each case is the specific power (watts per pound) of a STG that produces 300 watts of electrical power at a load voltage of 32 volts. Each optimization parameter is varied independently, with the overall optimization accomplished in a step wise procedure. In other words, values are assumed initially for thermoelement length and cross-sectional area and the collector area is optimized for these values. Maintaining the collector area at its optimized value, the individual thermoelement cross-sectional area is then varied across a range of values. The cross-sectional area that yields the highest value of specific power for the STG is then selected and subsequent analyses maintain thermoelement cross-sectional area at that value. With the solar collector area and thermoelement cross-sectional area maintained at their optimized values, the thermoelement length is varied in order to determine its optimum value. After this last optimization step, the STG is considered to have been optimized and the design that uses the thus optimized values of collector area, thermoelement cross-sectional area and thermoelement length represent the optimized design. It should be noted that a variety of other parameters of the STG could also be varied in an even more rigorous optimization procedure. This has not been done because most of the other parameters contribute less to the optimization of STG design than the three parameters here considered. It should be noted that prior to the final optimization procedure, some analyses were conducted with respect to the number of thermopiles to be used within the STG. Inasmuch as overall STG weight is only slightly dependent on the number of thermopiles, this number was fixed at 100 for the final optimization. It should also be mentioned that the absorptivity and emissivity characteristics of the hot and cold side heat exchangers of the STG have a drastic effect on overall STG size and weight. The absorptivity and emissivity values used in the

present study, 0.85 and 0.10 for the hot side heat exchanger and 0.10 and 0.85 for the cold side heat exchanger, represent the most optimistic values that can be fairly conveniently obtained in practice. These values also result in the smallest and lowest weight STG design.

The optimization of the flat-plate type STG is illustrated in Figures 6 to 8 in terms of plots of STG specific power as a function of total solar collector area, individual thermoelement cross-sectional area and thermoelement length. All of the points on each plot in Figures 6 to 8 pertain to a STG power output of 300 watts at a load voltage of 32 volts, with the STG directly facing the sun at a distance of 0.45 AU from it. It is noted in Figure 6 that the specific power maximizes for a total solar collector area of about  $100 \text{ cm}^2$  per thermopile. Assuming the STG to possess a square configuration, this area corresponds to a panel having a side width of about 84.4 inches. The total thickness of the STG is less than one inch. The optimization of STG with respect to thermoelement cross-sectional area in Figure 7 indicates that optimum thermoelement cross-sectional area is approximately  $1.0 \times 10^{-3}$  square centimeters. Inasmuch as this very value was used in the collector area optimization shown in Figure 6, it is noted that the optimum specific power in Figures 6 and 7 is essentially identical. Figure 8 shows STG specific power as a function of thermoelement length. It is noted that STG weight is a minimum for a thermoelement length of about 0.8 centimeter. Inasmuch as a thermoelement length of 1.0 centimeter was used in the optimization depicted by Figures 6 and 7, it is noted that fixing of thermoelement length at 0.8 centimeter results in a slight weight saving or higher value of specific power than the value obtained in Figures 6 and 7.

The results of the optimization of the flat-plate STG have been translated into a reference design for operation at a distance of 0.45 AU from the sun. The performance and configurational parameters of the

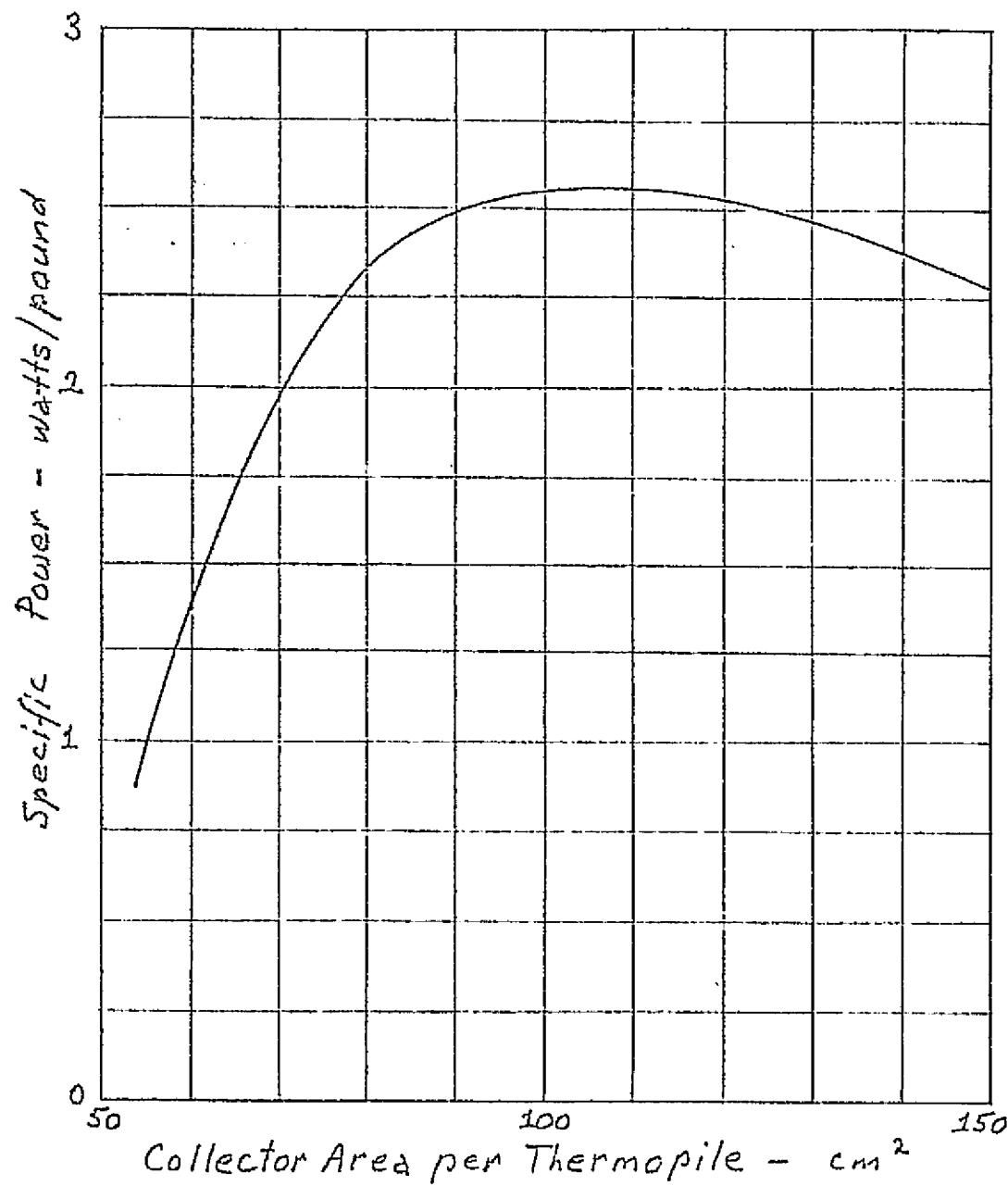


Figure 6

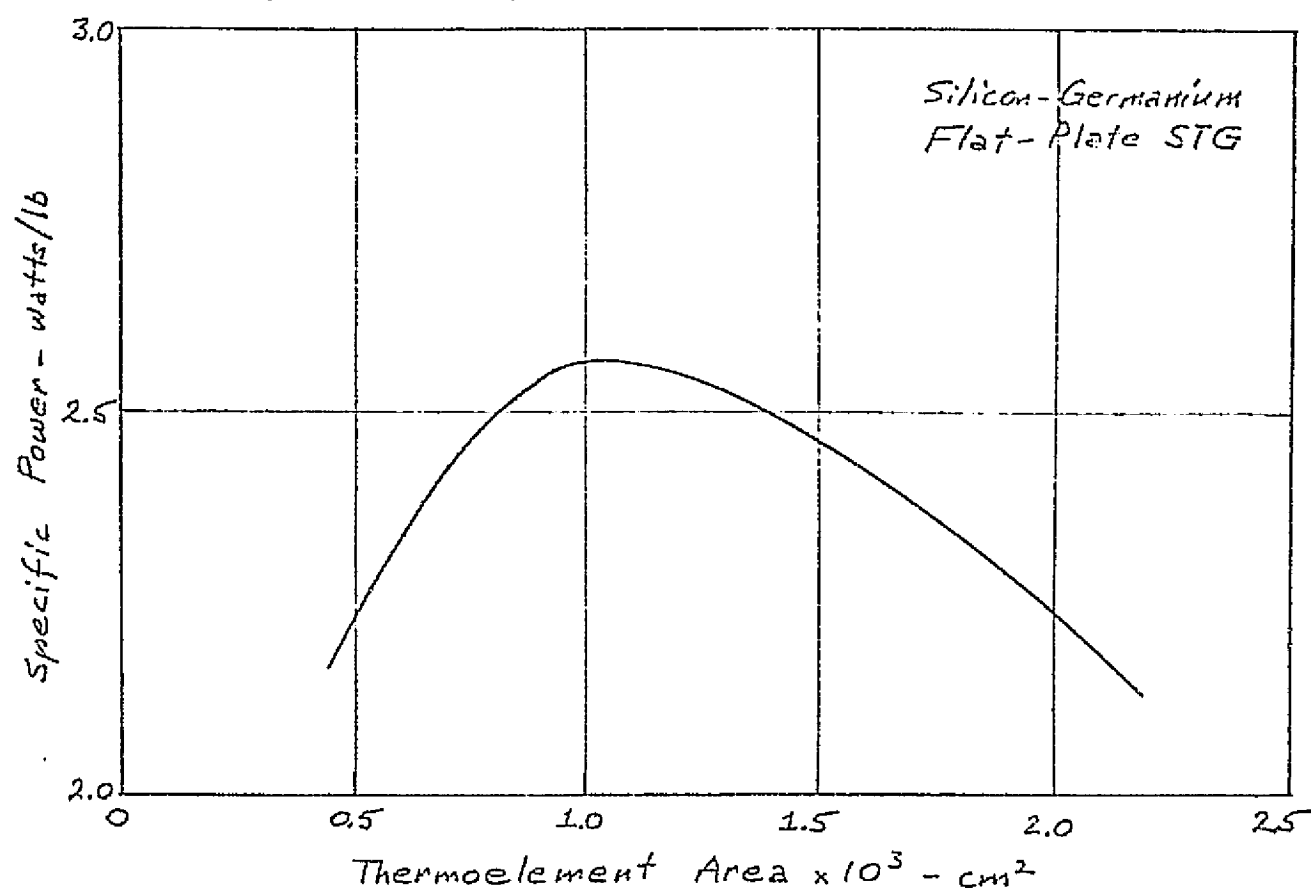


Figure 7

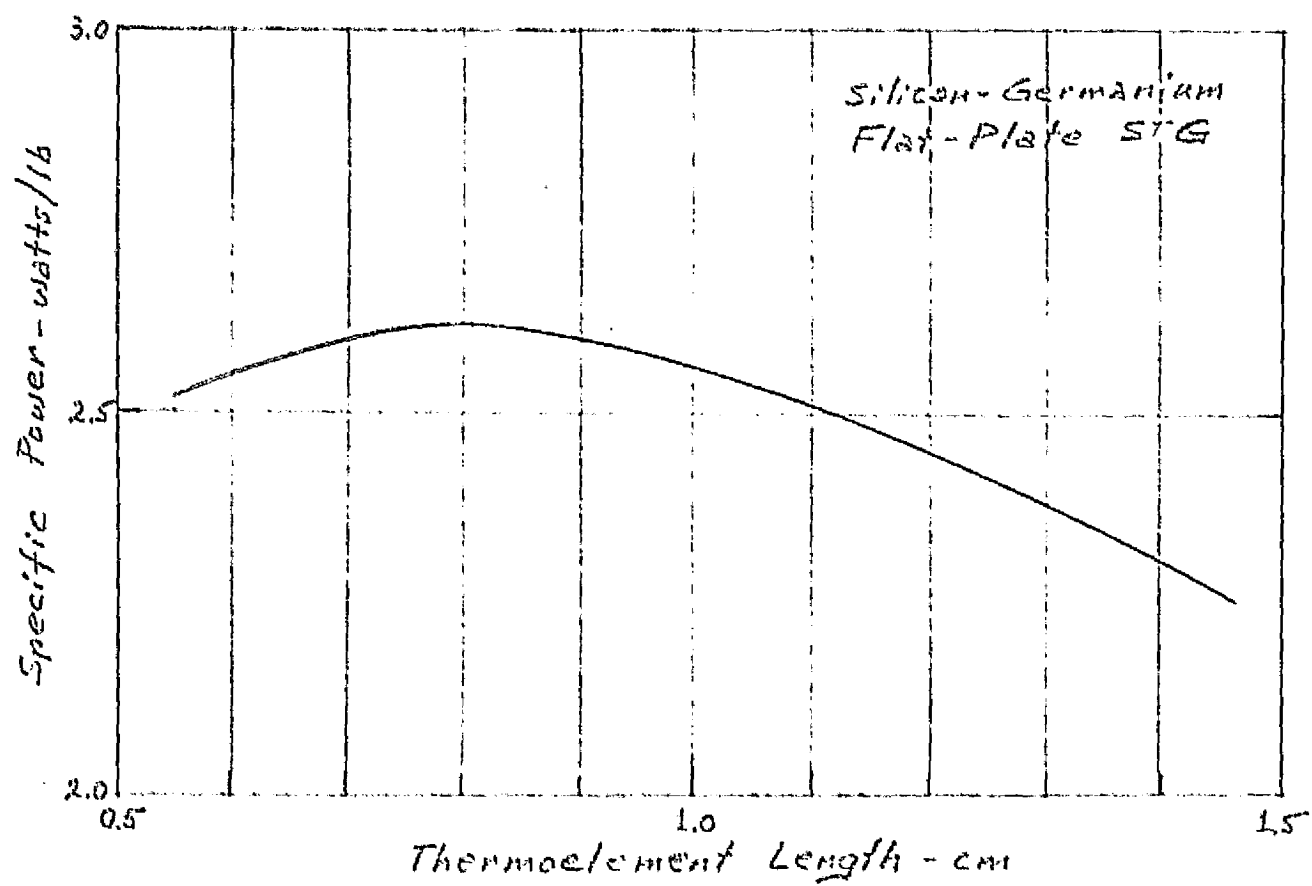


Figure 8

resultant STG design are summarized in Table 1. It must be emphasized that the performance values given for the flat-plate type reference design STG in Table 1 only pertain to the operation of the STG at a distance of 0.45 AU from the sun. At other distances from the sun, the STG performance considerably differs from that at 0.45 AU from the sun. This is illustrated in Table 2 in terms of the peak power output, load voltage at peak power, power output at 32 volts and the hot and cold side temperatures of the reference design STG as a function of the distance from the sun. It is noted in Table 2 that although the STG produces 300 watts of electrical power at its designed distance of 0.45 AU from the sun, at other distances from the sun, the power output is considerably different. For example, upon leaving the Earth, the STG produces only some 20 watts of electrical power. As the STG approaches the sun, this value rapidly increases until at its closest approach to the sun, 0.30 AU from the sun, it produces in excess of 900 watts of electrical power. A similar observation can be made in regard to the load voltage at peak power output of the STG. The temperatures of the STG are also significantly dependent on the distance of the STG from the sun. Upon leaving the Earth, the hot and cold side temperatures of the STG are both between 100 and 200°C. At the design distance of 0.45 AU from the sun, the hot side temperature of the STG has risen to about 470°C, while the cold side temperature is about 270°C. At its closest approach to the sun, these two temperatures have risen to in excess of 700 and 350°C, respectively. It should be noted that all of the performance and temperature values shown in Table 2 assume that the STG faces the sun directly. It is of interest, therefore, that even at its closest approach to the sun, the STG operates at temperatures significantly below the maximum permissible temperatures of silicon-germanium alloy thermoelectric technology. It is considered that this fact provides a considerable reliability margin to a flat-plate STG.

TABLE 1

Flat-Plate STG Dimensions and Performance

Number of Thermocouples per Thermopile	1200
Number of Thermopiles per Generator	41
Thermoelement Side Width - mils	12.4
Thermoelement Length - cm	0.8
STG Surface Area - square feet	53.4
STG Side Length - inches	87.7
STG Thickness - inch	0.45
STG Weight - pounds	114.7
Optimum Specific Power* - watts per pound	2.64
Optimum Power per Unit Area* - watts per square foot	5.65
Power Output* - watts	300.0
Load Voltage* - volts	32.0
Collector Temperature* - Centigrade	470.5
Radiator Temperature* - Centigrade	273.0

\* All Performance Values pertain to STG operation at a distance of 0.45 AU from the sun with the STG directly facing the sun.

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TABLE 2

Flat-Plate STG Performance\* As a Function Of Its Distance From The Sun

Distance From Sun AU	Collector Temperature °C	Radiator Temperature °C	Peak Power Output watts	Voltage at Peak Power volts	Power Output at 32 Volts watts	Optimum Specific Power watts/lb	Optimum Area Power watts/ft <sup>2</sup>
1.00	157.8	107.5	19.6	7.4	-	0.17	0.37
0.70	272.3	176.9	70.7	14.1	-	0.62	1.32
0.50	416.7	249.4	216.8	24.8	198.8	1.89	4.05
0.45	470.5	273.0	302.5	29.3	300.0	2.64	5.65
0.40	535.5	299.8	430.6	35.0	427.4	3.75	8.05
0.35	615.6	330.5	630.2	42.3	592.6	5.49	11.78
0.30	716.4	366.0	951.9	52.1	810.3	8.30	17.79
0.25	846.9	407.9	1494.6	65.2	1107.5	13.03	27.94
0.20	1022.7	458.9	2464.2	83.7	1524.7	21.49	46.06

\* Optimized for STG operation at a distance of 0.45 AU from the sun; all values pertain to a STG orientation in which the STG directly faces the sun.

Its only drawback is the relatively low specific power of between 2.5 and 3.0 watts per pound. Finally, in order to obtain the calculated performance of the STG, it is obvious that the STG must always be aimed directly at the sun. For this reason, if use is made of a flat-plate type STG, it will be necessary to also make use of a sun sensor that maintains the orientation of the STG in such a way that it always points to the sun.

From the performance figures shown in Table 2, it is apparent that the flat-plate STG will produce considerably in excess of its design power throughout much of its mission. This, however, is quite acceptable because nearly half of its mission time is spent behind the planet Mercury and during that portion of its mission, the STG will produce very little electrical power\*. Inasmuch as the spacecraft power requirement quite likely remains relatively constant at the design power value, the excess power produced by the STG during its mission can be used to keep storage batteries charged so that the storage batteries can be used to power the spacecraft while it is eclipsed by the planet during each of its orbits. If the STG produces excess power even after spacecraft and storage battery requirements are met, it is possible to dump such excess by means of a shunt. Moreover, by operating the STG at the fixed voltage of 32 volts, the power production of the STG will be maintained relatively fixed at the 300 watt power output level. In fact, if this is done, and it is also necessary to obtain power for storage batteries, it will be necessary to increase the size of the STG. Whatever course is actively followed depends on the detailed power requirement profile of the mission and the type of power storage used by the spacecraft.

\* The dynamics of STG performance within any given orbit of Mercury are discussed in Section V.

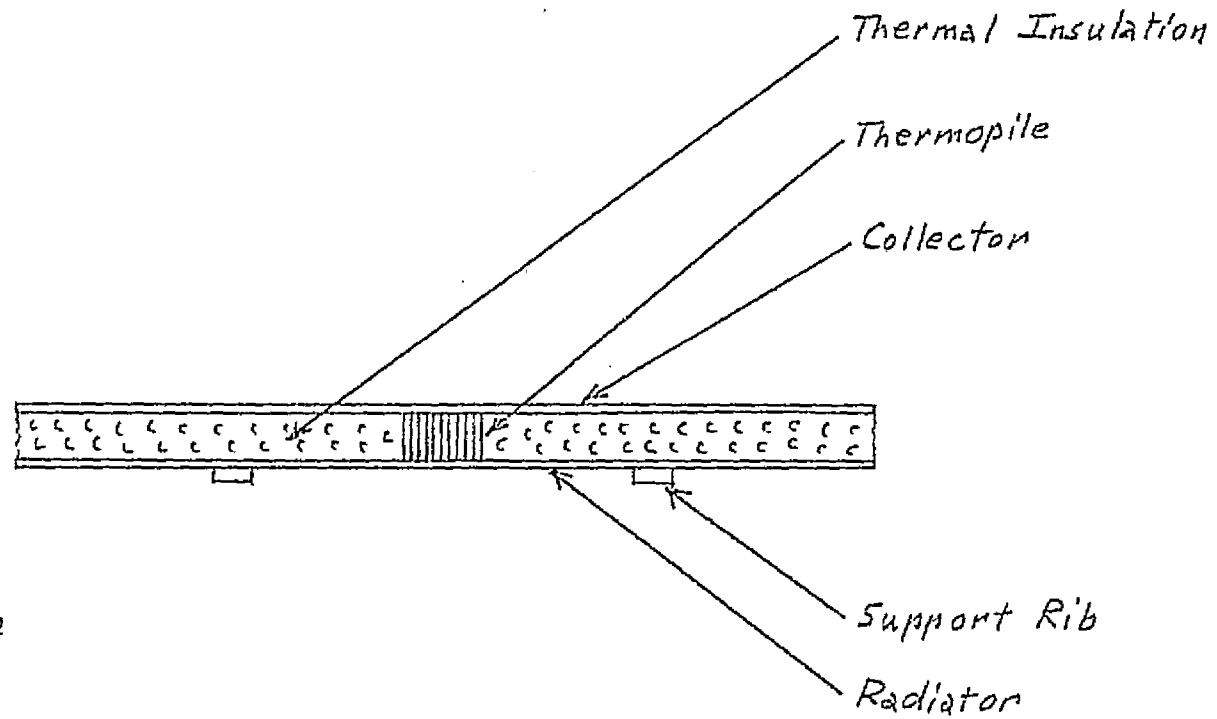
The configuration of the reference design STG is illustrated in Figure 9 in terms of the cross-section of a unit cell of the STG. The dimensions of the STG components shown in Figure 9 are in their proper proportion, although all dimensions have been scaled.

#### B. Compact STG Design With Single Concentrator

Using the mathematical model discussed in Section II B, parametric design analyses were conducted for the compact STG design that utilizes a single solar concentrator. Just as in the case of the flat-plate STG, the assumption was made that the compact STG with a single solar concentrator of the STG always directly faces the sun. Similarly, the design calculations have been performed for the STG operating at a distance of 0.45 AU from the sun, the farthest distance from the sun attained by the STG while orbiting the planet Mercury. Additionally, the assumption that the hot and cold side heat exchangers possess absorptivity and emissivity values identical to those assumed for the flat-plate STG has also been made. The solar concentrator has been assumed to possess perfect reflectivity.

The optimization procedure involves optimization with respect to thermopile hot side operating temperature, the temperature difference across the thermopile and the thickness of the solar concentrator. The reason for detailed optimization with respect to the thickness of the solar concentrator is that it was found that it is the solar concentrator that contributes the bulk of the weight of the STG. Assuming the solar concentrator and the radiator to be made of beryllium and maintaining the concentrator thickness at 0.25 inch, it was found that STG specific power does not possess an optimum for operating temperatures up to 1000°C; the specific power increases as a function of increasing hot side operating temperature. This is illustrated in Figure 10 in terms of a plot of STG specific power as a function of STG hot side temperature. The reason

Cross-Section of Flat-Plate STG Cell



Approximate  
Scale 1:2

Figure 9

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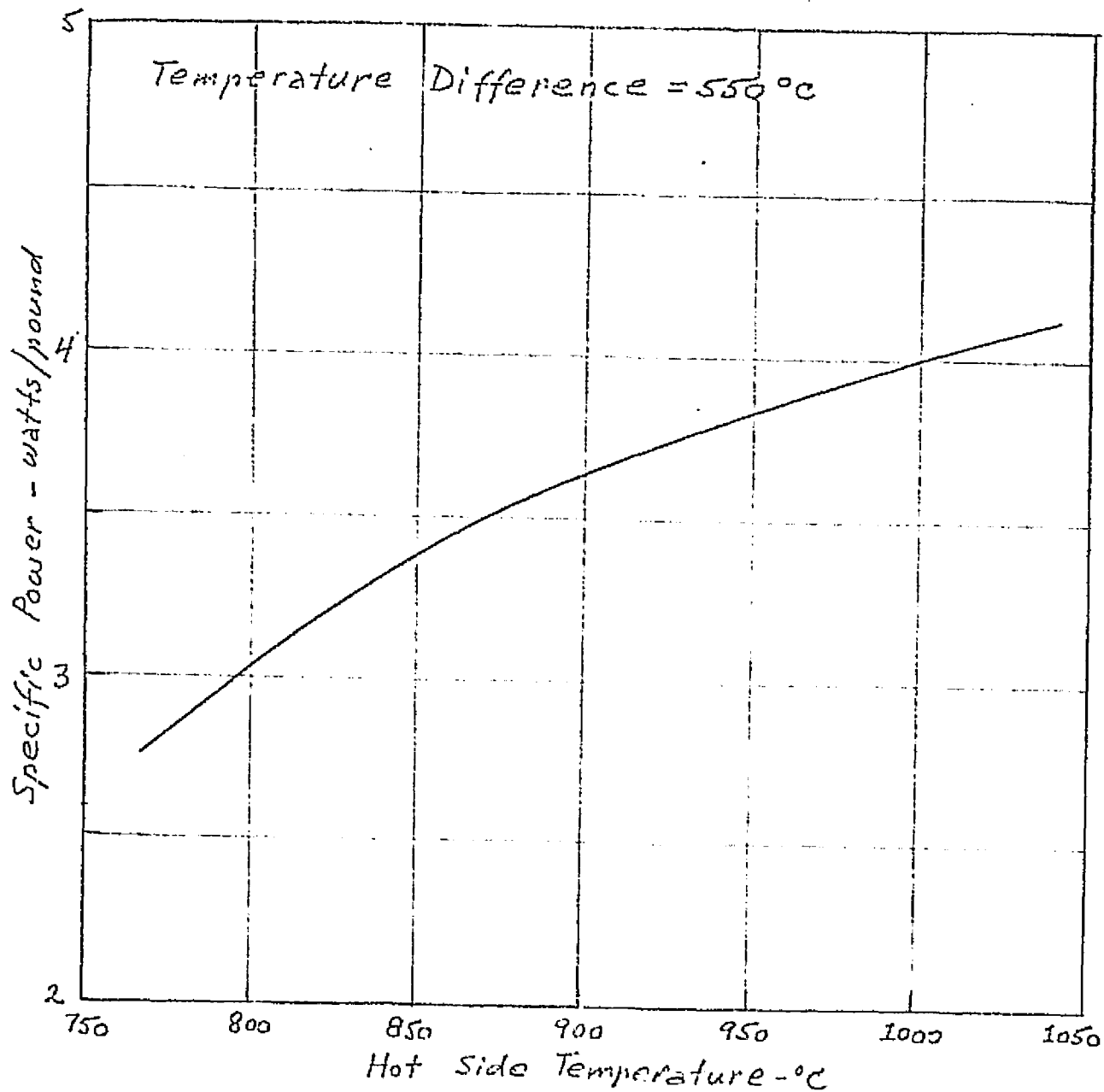


Figure 10

that the optimization procedure has not been taken beyond the hot side operating of  $1000^{\circ}\text{C}$  is that it is this temperature that is considered the maximum operating temperature for silicon-germanium alloy thermopiles in reliable long life operation. It is noted in Figure 10 that the specific power values possessed by the STG are considerably higher than those determined for the flat-plate STG in the preceeding section.

Still assuming that the solar concentrator possesses a thickness of 0.25 inch, the optimization of STG specific power as a function of the temperature difference across individual thermopiles within the STG is shown in Figure 11. Figure 11 shows a plot of STG specific power as a function of the temperature difference across the STG for the fixed hot side operating temperature of  $1000^{\circ}\text{C}$ . It is noted that the specific power values obtained by the variation of the temperature difference across the STG are somewhat higher than those determined by the variation of hot side operating temperature in Figure 10. The reason for this is that the latter optimization assumed the fixed temperature difference of  $550^{\circ}\text{C}$  across the STG. As seen in Figure 11, STG specific power actually optimizes for the temperature difference of nearly  $650^{\circ}\text{C}$ . As already mentioned, inasmuch as the bulk of the weight of the compact STG with a single concentrator resides within the concentrator, the final optimization for this type of a STG was performed with respect to concentrator thickness. Figure 12 shows STG specific power as a function of the temperature difference across the STG for different values of concentrator thickness. In all cases, it has been assumed that the STG operates at a hot side temperature of  $1000^{\circ}\text{C}$ . It is noted in Figure 12 that STG specific power has a considerable dependence on the thickness of the solar concentrator. This is in keeping with the finding that the bulk of the STG weight resides within the concentrator. In fact, it is noted that a four-fold reduction in concentrator thickness to one-sixteenth inch results in more than doubling the specific power of the STG. It

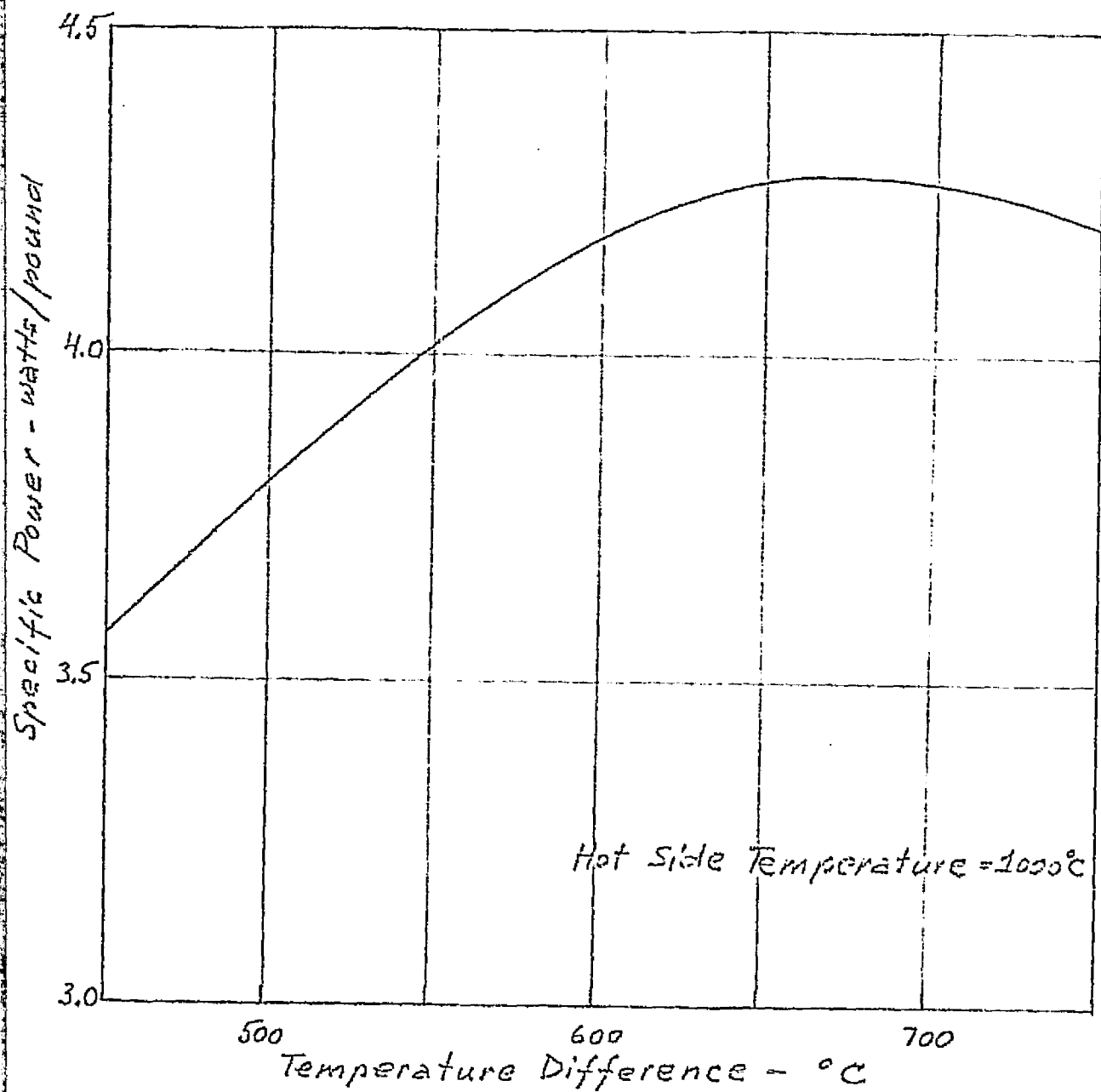


Figure 11

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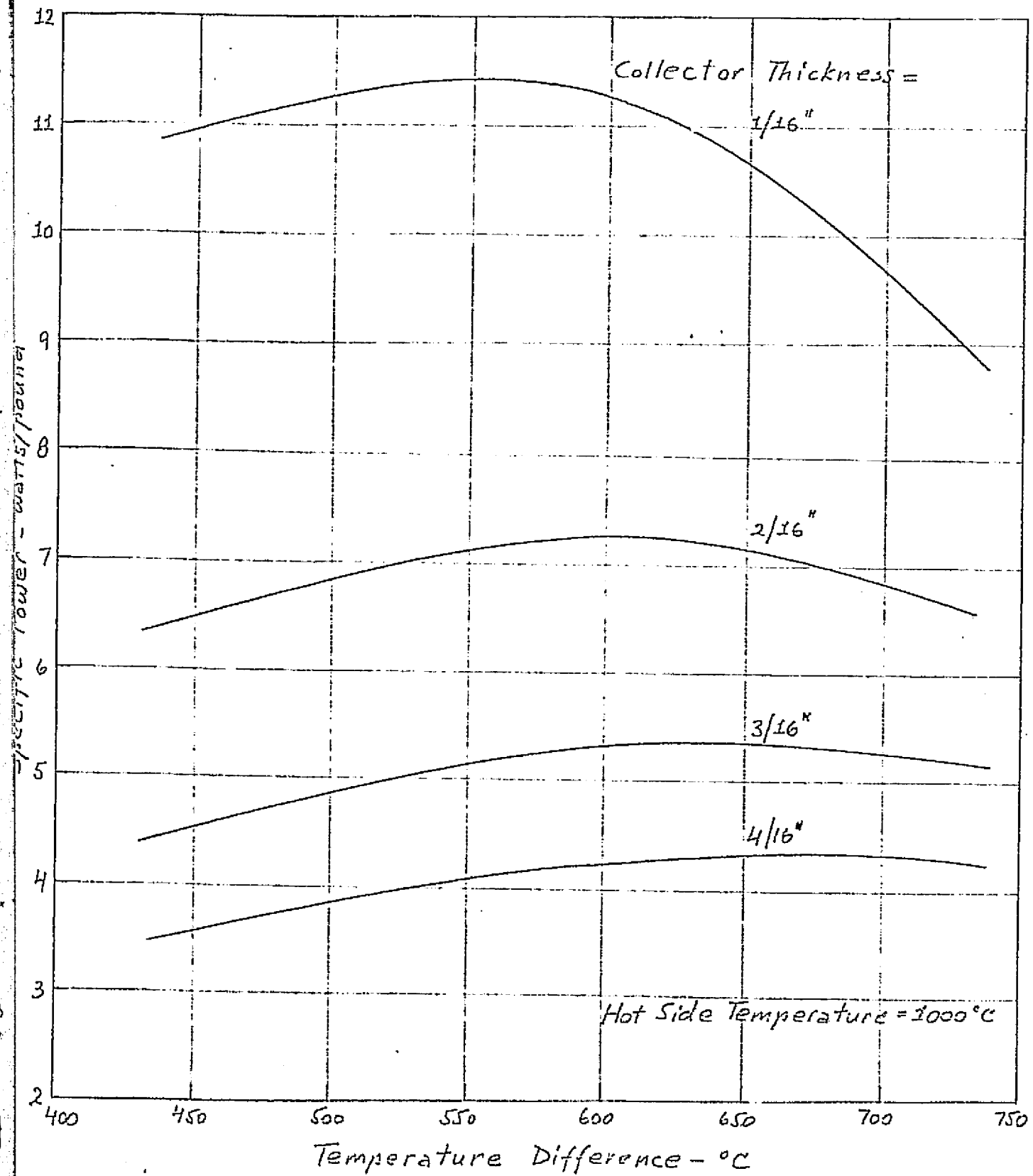


Figure 12

is believed that a concentrator thickness of one-sixteenth inch is quite practicable from the standpoint of the rigidity and strength of the concentrator. It is also noted in Figure 12 that the reduction of concentrator thickness is accompanied by a decrease in the temperature difference across the thermopiles within the STG that results in optimum specific power. With a solar concentrator having a thickness of one-sixteenth inch, the optimum specific power of the STG occurs at a temperature difference of about  $550^{\circ}\text{C}$ . At that temperature difference, the STG possesses a specific power of about 11 to 12 watts per pound. It is conceivable that an even higher value of specific power is possible if the concentrator does not possess a uniform thickness, but rather is constructed with a rib-type structure that has a very thin reflector attached to it. It must be emphasized that the detailed solar concentrator design is beyond the scope of the present study and therefore has not been included in the present treatment.

The detailed performance and configurational characteristics of the compact STG design derived from the parametric analyses illustrated by the data in Figures 10 to 12 are summarized in Table 3. It is noted that all of the performance values given in Table 3 pertain to STG operation at a distance of 0.45 AU from the sun, with the STG directly pointed at the sun. Inasmuch as closer approaches to the sun, such as those experienced by the STG during much of its mission time around Mercury, result in an increased incident solar flux, the hot side of the STG will heat to temperatures higher than  $1000^{\circ}\text{C}$  if the STG is maintained in a position directly pointing to the sun. Because the silicon-germanium alloy technology contemplated for use on the STG possesses a maximum temperature capability of  $1000^{\circ}\text{C}$  in long term operation, it will be necessary to tilt the STG with respect to the sun to keep it from overheating. This means that only at its farthest distance from

TABLE 3

Compact STG Dimensions and Performance

Number of Thermocouples per Thermopile	339
Number of Thermopiles per Generator	400
Thermoelement Side Width - mils	9.4
Thermoelement Length - cm	1.0
Concentrator Cross-sectional Area - square feet	10.67
Concentrator Diameter - inches	44.23
Concentrator Height - inches	40.95
STG Weight - pounds	26.31
Specific Power* - watts per pound	11.40
Power per Unit Collector Area* - watts per square foot	28.12
Power Output* - watts	300.0
Load Voltage* - volts	32.0
Generator Hot Side Temperature* - Centigrade	1000
Radiator Temperature* - Centigrade	422

\* All performance values pertain to STG operation at a distance of 0.45 AU from the sun and will remain constant at closer distances to the sun if STG is tilted (see Table 4).

the sun, at 0.45 AU, will the STG directly face the sun. At all other points in the orbit of Mercury around the sun, the STG will be tilted in such a way that the tilt angle increases with decreasing distance from the sun. Table 4 summarizes the tilt angles necessary to maintain the STG at the hot face temperature of  $1000^{\circ}\text{C}$  at all distances from the sun during its orbit around Mercury. If the STG operating temperatures are thus maintained at fixed values throughout its intended mission, it is obvious that the performance of the STG will also remain constant. In view of this, it will be necessary to overdesign the STG in order that not only adequate power be available to the spacecraft directly from the STG but also in order that power be available for maintaining adequate charge in the storage batteries that are used during the eclipse portions of each spacecraft orbit around Mercury.

A comparison of the performance and configurational data for the flat-plate and compact STG indicates that the latter type STG possesses a considerably higher specific power, and hence, a considerably lower total weight, than the former type STG. Although this is certainly true, the margin of weight difference reflected by the numbers in Tables 1 and 3 is somewhat excessive because the flat-plate STG produces considerable excess power during most of the mission, while the compact STG produces the design power throughout. As already stated, in the actual mission, it will therefore be necessary to increase the size of the compact STG in order that excess power for battery storage be available. This, of course, increases overall STG weight and reduces the effective specific power of the compact STG. By the same token, it may be possible to reduce the size of the flat-plate STG because of the considerable amount of excess power it produces during most of its mission time. This, of course, results in the two types of STG being closer in size and weight than the numbers in Tables 1 and 3 indicate.

TABLE 4

STG Orientation With Respect to Sun  
Required for Constant Performance

<u>Distance from Sun</u> <u>AU</u>	<u>STG Tilt Angle with</u> <u>Respect to Sun</u> <u>degrees</u>
0.25	72.0
0.30	63.6
0.35	52.8
0.40	29.0
0.45	0.0

Even with this being the case, it is nevertheless true that the compact STG will in all cases possess a lower weight than the flat-plate STG. The primary reason for this is the considerably smaller reradiation surface area of the hot face of the compact STG as compared to the hot face of the flat-plate STG and the consequent ability to operate the former STG at the maximum permissible operating temperature of the silicon-germanium alloy technology; the latter STG in all cases operates considerably below that temperature, as can be seen in Table 2.

Inasmuch as the use of a solar concentrator is clearly beneficial to the overall performance and weight of a STG it was decided that the most advantageous STG configuration of all might be one that combines some features of both the flat-plate and the compact STGs. Namely, the utilization of individual miniature solar concentrators for each thermopile in place of the flat solar collectors used in the flat-plate STG design would enable the retention of the attractive features of a compact STG in a panel-like configuration. A panel-like configuration of course offers an advantage over the compact type STG configuration in its ability to be stored during transit from Earth to Mercury and ease of deployment upon injection into Mercury orbit. A STG that utilizes individual solar concentrators in a panel-like configuration will be considered in terms of performance and configuration in the next section.

### C. Panel-like STG Design With Multiple Solar Concentrators

In view of the findings of the parametric analyses performed on the flat-plate and compact STGs, it was concluded at the end of the preceding section that conceivably a STG design of considerable interest would be one that combines certain features of both of the other designs. This design is similar to the flat-plate STG design in that the thermopiles are uniformly distributed across one face of the radiator, with

thermal insulation separating the individual thermopiles. In place of the flat solar collector attached to each thermopile, however, this design concept makes use of miniature solar concentrators. These concentrators have the very same configuration as the solar concentrator assumed for the compact STG design, except that each concentrator is scaled to the heat input requirements of each individual thermopile; there are, therefore, as many miniature solar concentrators as there are thermopiles within the STG. The attachment of each concentrator to its respective thermopile is very likely metallurgical.

Parametric design analyses on the distributed STG design that utilizes individual solar concentrators for each thermopile were conducted by using the mathematical model discussed in Section II C. As noted in Section II C, the mathematical model is very similar to that used for the compact STG design discussed in Section II B. The parametric analyses essentially confirm the belief that the distributed STG design combines the best features of both the flat-plate and the compact designs and, in fact, results in the highest values of specific power of any of the three different designs. The design calculations on the distributed STG are more conservative than those made for the compact STG because although it is assumed that the solar concentrators possess perfect reflectivity, the thermopile hot faces are assigned absorptivity values of 0.50 and emissivity values of 0.50. The radiator is assumed to possess an emissivity of 0.85 and an absorptivity of 0.10. It is also assumed that individual solar concentrators represent truncated cones having the same relative dimensions, in a scaled-down version, as the single solar concentrator used in the compact STG design. Furthermore, it is assumed that Min-K 2020 thermal insulation separates the individual thermopiles within the STG. The thermopiles are the mult-thermoelement monolithic-type structures previously discussed, i.e. each thermopile produces the load voltage of 32 volts

directly. Both the solar concentrator and the radiator are assumed to be constructed of beryllium. The solar concentrator is assumed to possess a thickness of 0.010 inch. It should be noted that even though this is a very small thickness, in the small sizes anticipated for the individual solar concentrator, it is believed that the concentrator possesses adequate rigidity and strength characteristics to withstand any environmental loading that it may be subjected to during launch, transit to Mercury and deployment in Mercury orbit. If it is decided that beryllium is not suitable for the concentrator, it can be easily replaced with a concentrator made of some other metal. For example, tungsten or molybdenum would be suitable materials for that purpose. It is recognized, of course, that any replacement of beryllium as the solar concentrator material would result in a somewhat heavier concentrator because beryllium possesses a lower density than any of the other materials that might be contemplated for use. Inasmuch as in the distributed STG design the solar concentrator weight is actually very small, the overall effect on the specific power of the STG will be only nominal, with the STG weight increasing slightly and the specific power decreasing accordingly.

The parametric design analyses on the distributed-type STG were conducted as a function of STG hot side temperature, the temperature differential across each thermopile within the STG and the operating distance of the STG from the sun. In all cases it was assumed that the STG faces the sun directly. As noted in connection with the compact STG design, if over-temperaturing of the STG should present a problem upon closer approaches to the sun than the design distance, it is possible to tilt the STG in order to reduce the incident solar flux impinging upon it. Figures 13 to 15 show plots of STG specific power as a function of the temperature differential across individual thermopiles within the STG for different values of operating distance from the sun. Figure 13

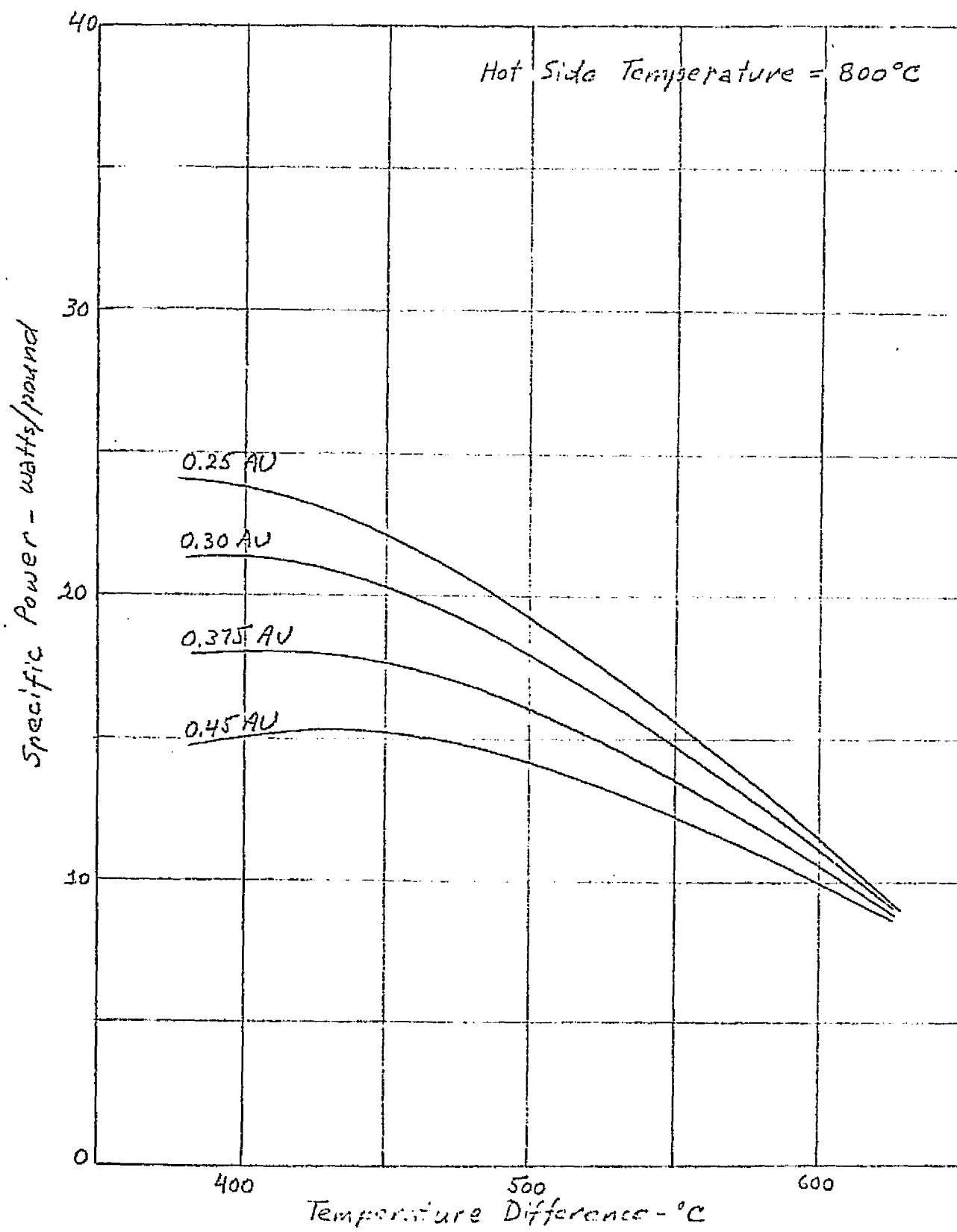


Figure 13

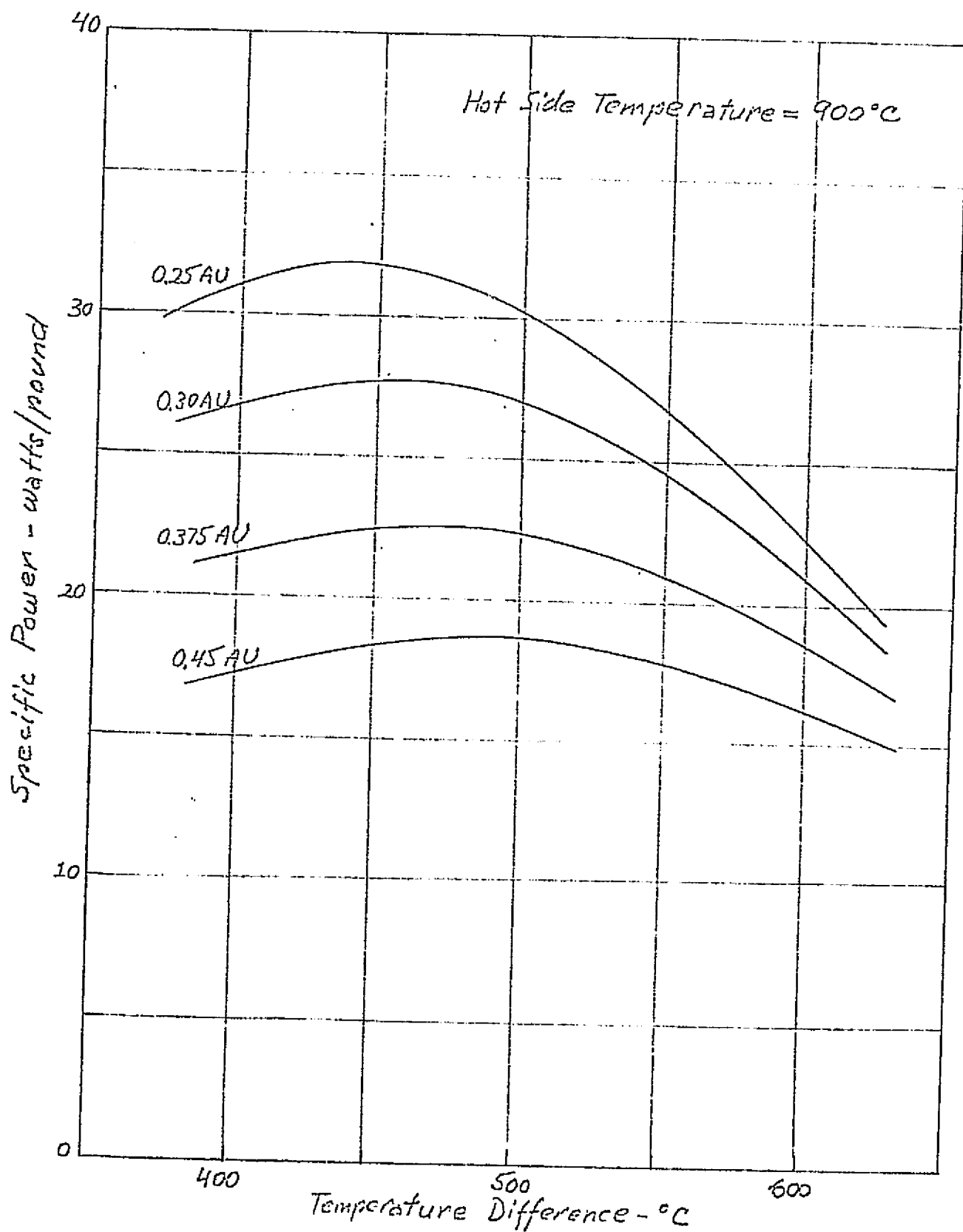


Figure 14

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shows STG operation at a hot side temperature of  $800^{\circ}\text{C}$ . Figures 14 and 15 show performance at hot side operating temperatures of  $900$  and  $1000^{\circ}\text{C}$ . It is noted in Figures 13 to 15 that the specific power increases with decreasing distance from the sun, as is to be expected. It is also noted that for each hot side operating temperature and operating distance from the sun specific power maximizes for a very definite value of temperature differential across the thermopiles within the STG. At any given operating hot side temperature, the closer the STG is to the sun, the lower is the temperature differential across the thermopiles at which optimum specific power occurs. On the other hand, as the hot side operating temperature of the STG increases, the temperature differential at which the optimum specific power occurs increases towards higher values. Similarly, the specific power of the STG increases with increasing hot side operating temperature. An inspection of Figure 15 reveals that specific power values for a STG hot side operating temperature of  $1000^{\circ}\text{C}$  are considerably higher than those previously found for either the flat-plate or the compact STG designs. For example, operating at a distance of 0.45 AU from the sun, the STG possesses an optimum specific power in excess of 22 watts per pound.

It is of interest to cross-plot the results of the parametric design analyses on the distributed STG design that uses multiple solar concentrators. This has been done in Figures 16 to 18 for the optimum specific power point on each of the curves shown in Figures 13 to 15. Figure 16 shows plots of optimum specific power as a function of STG operating distance from the sun at three different values of hot side operating temperature, namely,  $800$ ,  $900$  and  $1000^{\circ}\text{C}$ . As is to be expected, it is noted in Figure 16 that STG specific power increases with increasing hot side operating temperature and decreases with increasing distance from the sun. Figure 17 shows plots of STG cold side operating temperature

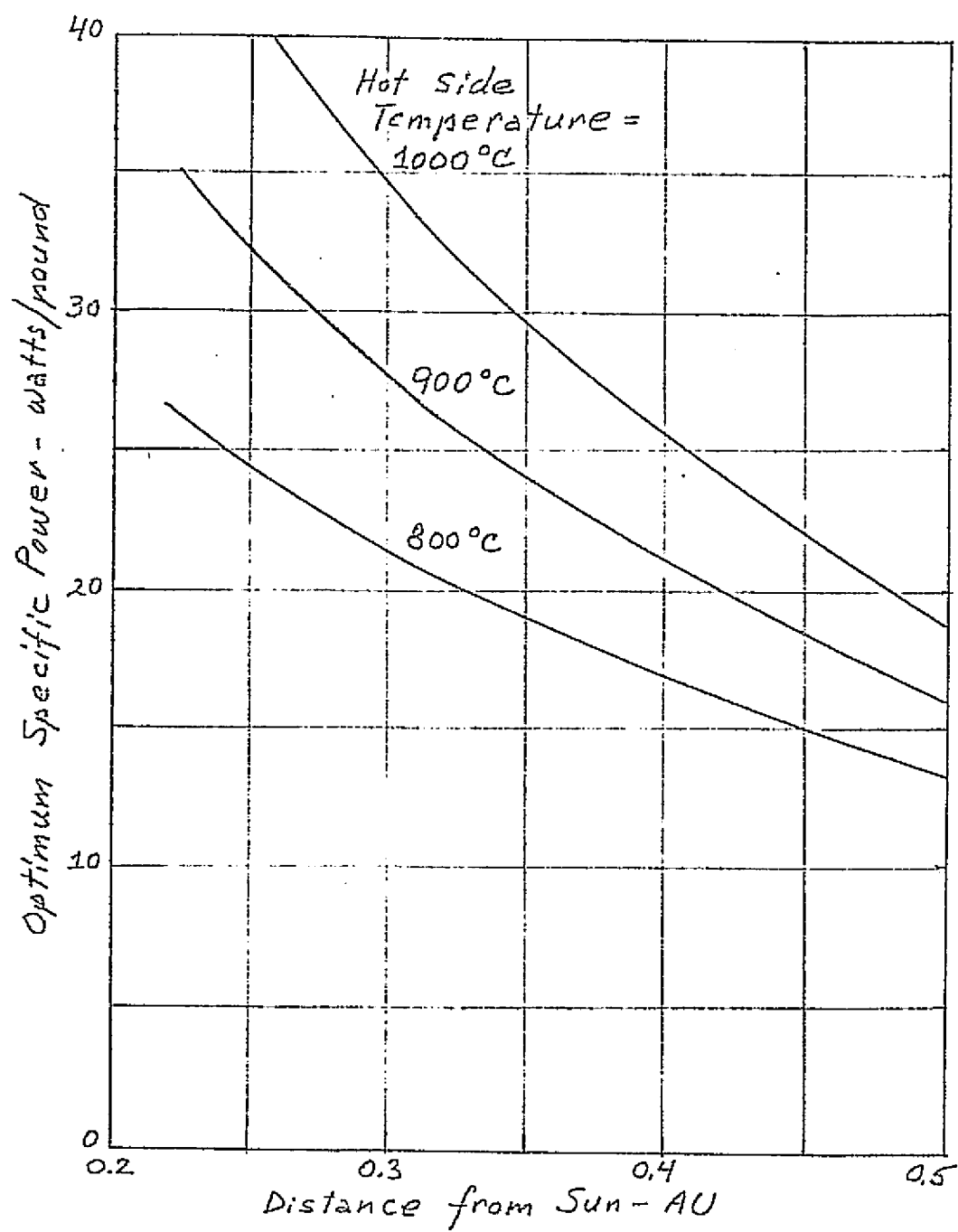


Figure 16

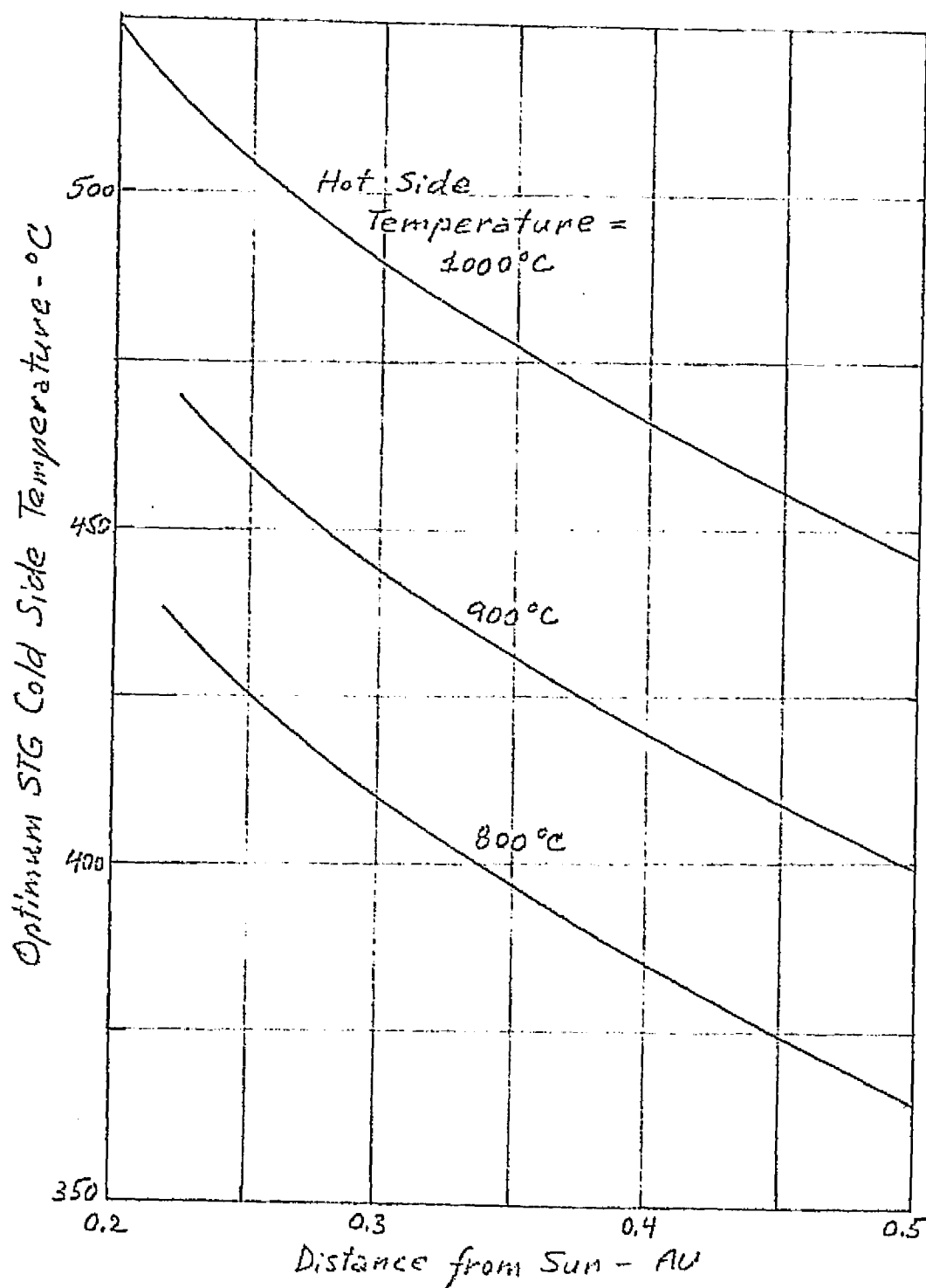


Figure 17

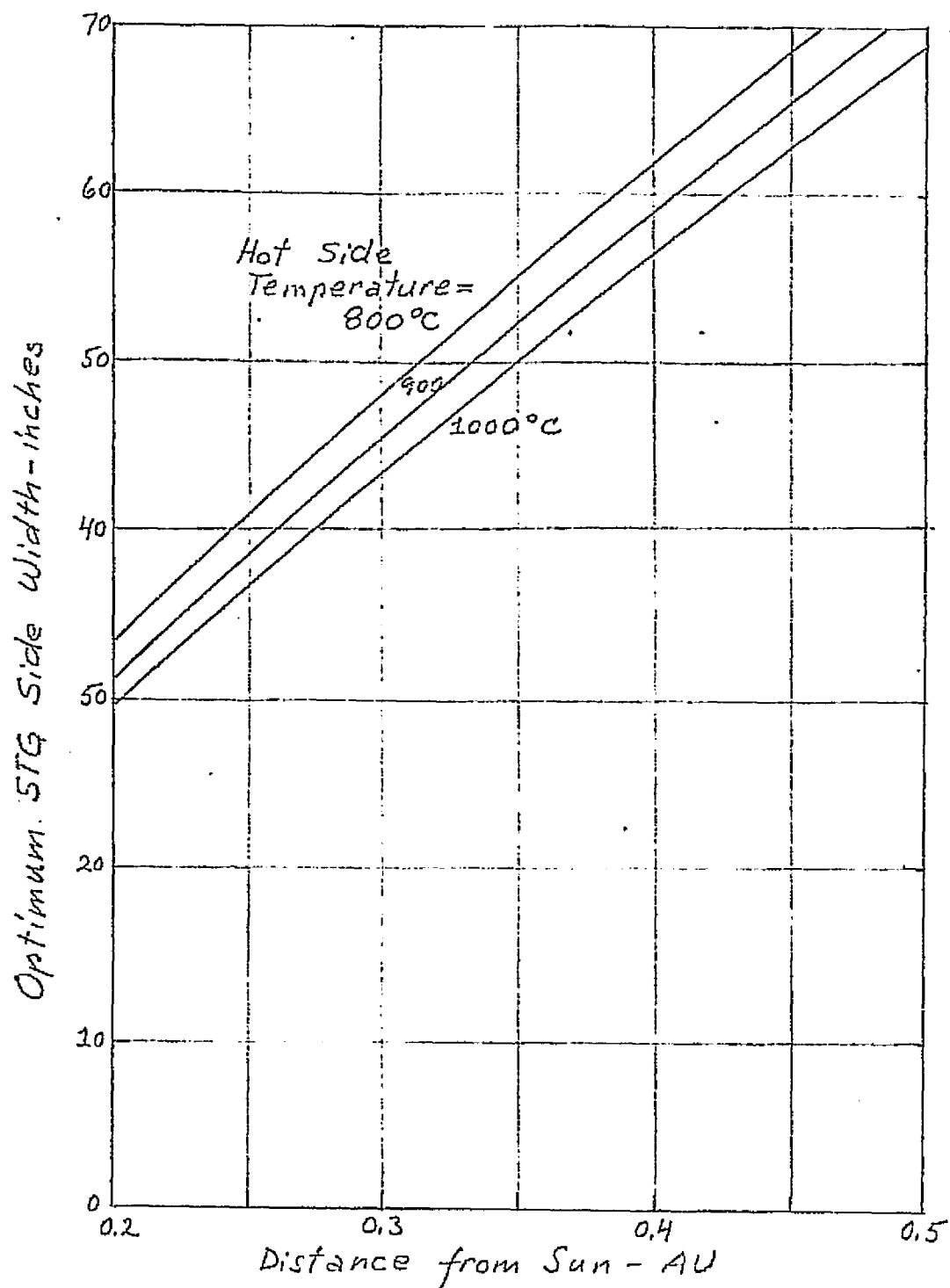


Figure 18

at its optimum specific power point as a function of distance from the sun for the same three hot side operating temperatures displayed in Figure 16. It is noted in Figure 17 that STG cold side operating temperature is proportional to its hot side operating temperature. It is also noted that the STG cold operating temperature that corresponds to optimum specific power decreases with increasing distance from the sun. Finally, Figure 18 shows plots of STG side width, assuming the STG to possess a square cross-section, as a function of distance from the sun at different hot side operating temperatures; the data in Figure 18 again pertain to optimum specific power of the STG.

Just as in the cases of the flat-plate and compact STGs, all of the data shown in Figures 13 to 18 pertain to a STG power output of 300 watts at a load voltage of 32 volts. Also, as in the previous cases, the STG design selected for the distributed-type STG has been optimized for operation at a distance of 0.45 AU from the sun. This represents the farthest distance attained by the STG from the sun during its orbital mission around Mercury. As previously stated, the hot side operating temperature of the STG increases as the STG approaches the sun and therefore if the design operating hot side temperature at 0.45 AU is maximized, as it is done here, the STG must be tilted at all points in the orbit of Mercury except when Mercury is at its farthest distance from the sun. The maximized hot side operating temperature assumed for the STG is  $1000^{\circ}\text{C}$ . As can be seen in Figure 16, this hot side operating temperature corresponds to an optimum specific power of about 22 watts per pound at a STG operating distance of 0.45 AU from the sun.

The selected distributed-type STG design is summarized in terms of configurational and performance parameters in Table 5. The operating and performance data in Table 5 correspond to STG operation at 0.45 AU from the sun and these data remain constant at closer approaches to

TABLE 5

Distributed STG Dimensions and Performance

Number of Thermocouples per Thermopile	388
Number of Thermopiles per Generator	144
Thermoelement Side Width - mils	14.6
Thermoelement Length - cm	1.0
Individual Concentrator Diameter - inches	5.27
Individual Concentrator Height - inches	4.88
STG Cross-sectional Area - square feet	27.73
STG Height - inches	5.41
STG Weight - pounds	13.53
Specific Power * - watts per pound	22.17
Power per Unit Area - watts per square foot	10.82
Power Output* - watts	300.0
Load Voltage* - volts	32.0
STG Hot Side Temperature* - Centigrade	1000
Radiator Temperature* - Centigrade	422

\* All performance values pertain to STG operation at a distance of 0.45 AU from the sun and will remain constant at closer distances to the sun if STG is tilted (see Table 4).

the sun if the STG is tilted according to the tilt angles given in the preceding section in Table 4. It is noted that the STG produces its designated output only while it is exposed to the solar heat flux. Inasmuch as approximately one-half of its operation occurs in an operating mode in which the STG is eclipsed by the planet Mercury, it is very likely necessary to upscale the size of the STG if the STG is to provide power for storage batteries in addition to providing power for the operation of the spacecraft while it is not in eclipse. Therefore, the configurational parameters shown in Table 5, just as those shown in Table 3 for the compact STG, are quite likely conservative. In fact, if it is assumed that in the non-eclipse mode of operation the STG produces twice its required power, one-half of it being directly used by the spacecraft and one-half being used to charge the storage batteries, the total size, i.e. the cross-sectional area, of the STG will be double that shown in Table 5.

Of course, the STG may be optimized for an operating distance from the sun other than 0.45 AU or for an operating temperature that is less than  $1000^{\circ}\text{C}$  at 0.45 AU in order that the STG produce more power at distances in-between its closest and farthest approach from the sun. In this way, it will be possible to attain a STG design that is even more optimum than a design that is based on simply increasing the size of the STG reflected by the configurational parameters in Table 5. Although this has not been done here, it must be considered prior to a final STG design selection. The design given in Table 5 is representative and is directly comparable to the compact STG design given in Table 3.

The distributed-type STG is schematically illustrated in Figure 19. As it is shown in the figure, it corresponds to the design specified in Table 5. The STG consists of 144 thermopiles in a twelve by twelve array. Each thermopile is affixed on its cold side to the radiator of the STG. Min-K 2020 thermal insulation separates the thermopiles. Individual

Distributed STG with Concentrators

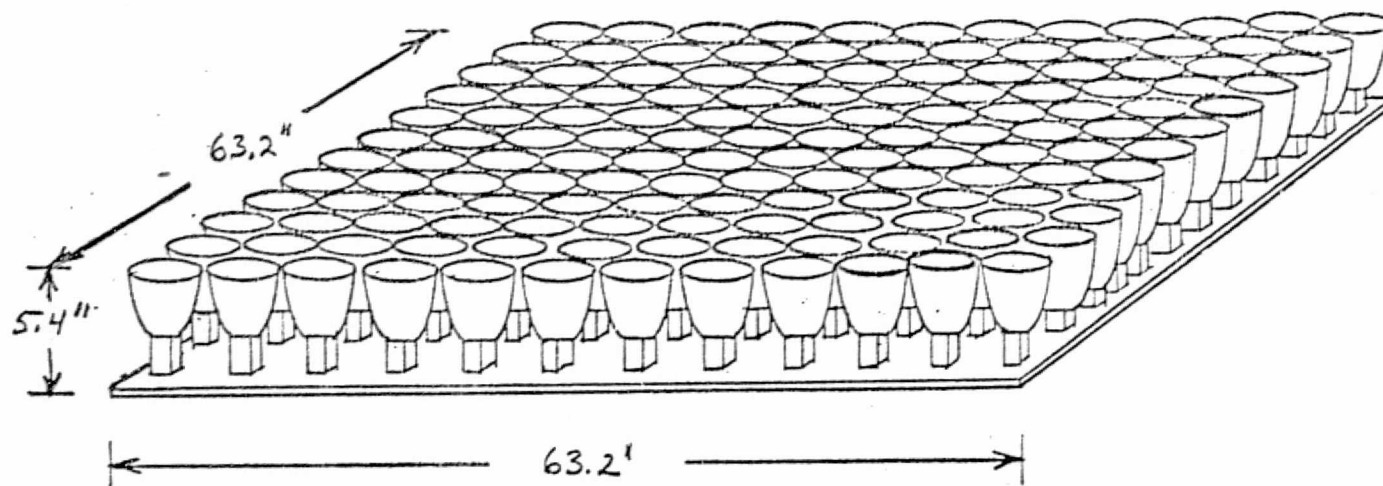


Figure 19

truncated cone configured solar concentrators are attached to the hot side of each thermopile. It should be noted that even though the radiator of the STG in Figure 19 is shown to be the same size as the total array of side-by-side concentrators, the radiation area needed to reject the heat according to the specific design is actually smaller than this area. This means that the radiator of the STG actually consists of a frame into which are fitted unit STG cells consisting of the solar concentrator, thermopile and radiator section. The various radiator sections of the unit cells do not completely fill the frame. For this reason, the total radiating surface of the STG is actually smaller than the size of the STG as defined by the twelve by twelve array of solar concentrators. In other words, a direct view of the cold side of the STG shows the total area to be partially filled with unit radiator sections. Individual unit radiator sections are separated from each other, being interconnected by the frame structure, such that the Min-K 2020 thermal insulation is directly visible in-between the various sections.

Inasmuch as the STG design specified in Table 5 pertains to a STG that has been optimized for operation at a distance of 0.45 AU from the sun, it is of interest to consider the performance of this design at other distances from the sun. As already stated, as the STG approaches the sun more closely, it tends to heat up and consequently provide greater performance. Because the optimized design at 0.45 AU pertains to STG hot side operating temperature of  $1000^{\circ}\text{C}$ , a closer approach to the sun would result in higher operating temperatures and therefore the STG must be tilted with respect to the sun if it is to be kept from overheating. As a consequence, STG performance and operating temperatures will remain constant as the STG approaches the sun more closely than 0.45 AU. Table 6 shows values of STG performance and temperatures if the STG operates in a mode in which it always directly faces the sun.

For this reason it is seen that performance and operating temperature values at distances closer to the sun than 0.45 AU exceed those calculated for it at 0.45 AU. The performance and operating temperature of the STG rapidly decrease as the distance from the sun increases to values beyond 0.45 AU and, in fact, at a distance of 1.0 AU these values are considerably less than the design values. It is obvious that the STG does not provide adequate power between the time of its launch from Earth and the time that it arrives at Mercury. It is suggested that during that time period use be made of storage batteries or solar cells for spacecraft power needs. Upon injection of the spacecraft into Mercury orbit, all of the power needs can be satisfied by the STG. This is especially true if the STG is tilted such that at all times in Mercury orbit it provides a fixed amount of power output. Inasmuch as the STG produces power only during the time that it faces the sun, approximately half of each of its orbits around Mercury, it will be necessary to increase the overall size of the STG in order that the storage batteries be charged while the STG faces the sun. It is the storage batteries that provide the total spacecraft power during the times that the spacecraft is eclipsed by the planet. Even though the data in Tables 5 and 6 pertain to a 300 watt electrical power output of the STG, it is completely valid to linearly upscale the STG design for any power need that not only satisfies spacecraft requirements, but also enables the maintenance of the storage batteries in a fully charged condition. For example, if it is decided that this can only be accomplished by having the STG produce twice the stated amount of power during the time that it faces the sun, then the overall STG size doubles, including its weight. As a result it is a simple matter to generalize the results given for the distributed STG design in this section to any performance or operating requirements. This, of course, is also true of the flat-plate and compact STG designs discussed in the preceeding sections.

#### D. Cylindrical STG With Single Concentrator

Although most of the effort on the present study was devoted to the design and performance analysis of the three STG designs discussed in the preceeding sections of this part of the report, an additional STG design concept was investigated in somewhat more than just a cavalier fashion. This STG design concept pertains to a configuration that is very similar to most radioisotope thermoelectric generator configurations. Namely, the thermopiles within the STG are enclosed in a cylindrical outer case, with the cold ends of the thermopiles in intimate contact with the outer case. The hot ends of the thermopiles are located around a central heat source, which provides heat to the thermopiles either by direct conduction or by radiation. Whereas the central heat source in a radioisotope thermoelectric generator is a radioisotope fuel capsule, in the present instance it is a heat pipe that receives its heat from the sun by means of a solar concentrator attached to one end of it. This means that the heat pipe penetrates the outer case and is mated to the solar concentrator outside the outer case. In the space between the solar concentrator and the outer case of the STG, the heat pipe is thermally insulated to minimize heat losses. For the same reason, the space between individual thermopiles within the STG is also thermally insulated, as are the ends of the STG. Waste heat is radiated from the outer case of the STG directly into space. In order to enhance heat transfer from the outer case to the space environment, the STG outer case possesses radiation fins. In its most optimum mode of operation, the solar concentrator at one end of the STG directly faces the sun. This means that the body of the STG is shielded from a direct view of the sun by the solar concentrator, with a result that relatively little, if any, sunlight impinges on

the outer case of the STG. This, of course, is important in order to enhance the effectiveness of the STG; in other words, the cold side heat exchanger, i.e. the outer case and radiation fins, is only heated by heat that has passed through the heat pipe and the STG. If it becomes necessary to tilt the STG, as it is in the case of the other STG design concepts, then it may no longer be true that the cold side heat exchanger does not receive any direct heating from the sun. In order to minimize the effects of such occurrences on the performance of the STG, it is desirable to coat the outer case and the radiation fins of the STG with a low absorptance coating. Needless to say, the outer case and the radiation fins must in any case possess a high emissivity coating in order to enhance the rejection of waste heat. The cylindrical STG design concept is schematically illustrated in Figure 20. Figure 20 shows the outer case, with its radiation fins, and the solar concentrator of the STG interconnected by means of a heat pipe. Inasmuch as Figure 20 depicts a view of the outside of the STG, it is not apparent how the hot sides of the thermopiles are connected to the heat pipe heat source. It is envisioned that this will be accomplished by means of a sleeve that on its inside exactly fits the heat pipe and makes intimate contact with it. On its outer surface, the sleeve has a polygonal cross-section. The hot faces of the thermopiles are impressed against the flat faces of the outer surface of the sleeve. The compression of the thermopiles against the sleeve is provided by spring assemblies at the cold side of the thermopiles. In using such a method, it is of course important to also provide adequate thermal conductance between thermopile cold sides and the STG outer case. This can be accomplished either by using a spring-follower type approach previously used in various thermoelectric generators or by making use of braided flexible conductors in between thermopile cold sides and the outer case.

# Cylindrical STG Design

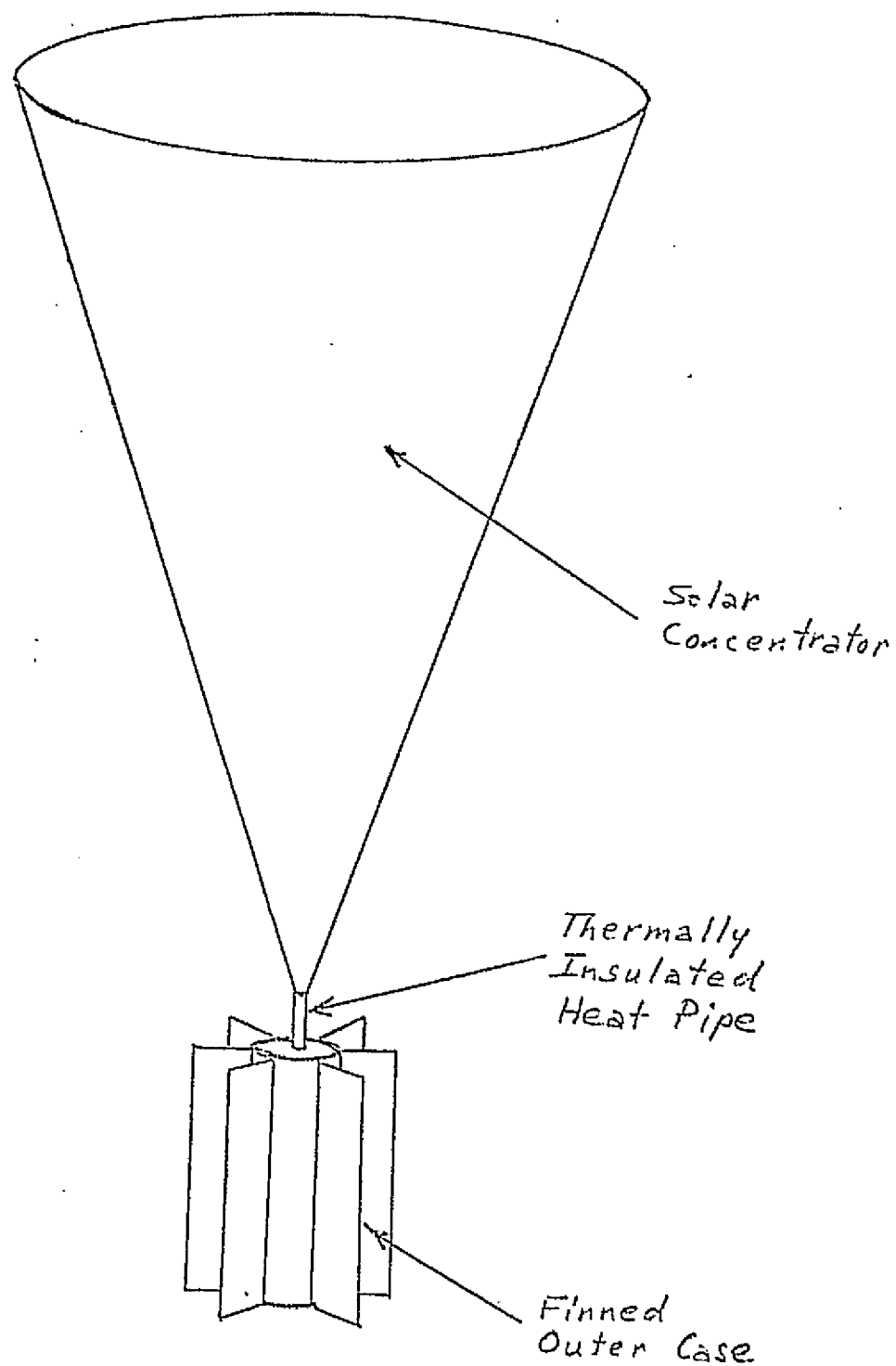


Figure 20

The mathematical model used to conduct parametric analyses on the design of a cylindrical STG with single solar concentrator is very similar to that used for the analyses pertaining to the compact STG discussed in section II B. In fact, the only difference in the two mathematical models is a geometric term that modifies the compact STG with its flat geometry into a cylindrical STG. The primary effect of this is a reshaping of the thermal insulation, the addition of radiation fins to the radiator which has been folded into a cylindrical configuration and the addition of the heat pipe that interconnects the solar concentrator and the hot sides of the thermopiles. The calculational sequence of the mathematical model, however, has not been altered.

A number of parametric analyses were conducted with this model in order to derive a design of a cylindrical STG that can be directly compared to the designs previously developed for the other STG concepts. It should be mentioned, however, that the analytical procedure was not carried out as extensively for this STG design concept as it was for the other concepts. Nevertheless, it is felt that the design derived for the cylindrical STG represents an optimum and therefore can be directly compared to the other STG designs. The design thus derived pertains to STG operation at a distance of 0.45 AU from the sun. Essentially, the same assumptions as those made for the other STG designs were made for the physical characteristics of various STG components. For example, it was assumed that the solar concentrator possesses perfect reflectivity and that the cold side heat exchanger has an emissivity of 0.85 and a solar absorptivity of 0.10. The results of the parametric analyses indicate that an optimum cylindrical STG design operates at a hot side temperature of  $1000^{\circ}\text{C}$  and a cold side temperature of about  $450^{\circ}\text{C}$ . The number of thermopiles within the STG is 144, just as it was in the case of the earlier STG designs. These thermopiles are arranged in

such a way that eight rows of 18 thermopiles are located circumferentially around the polygonal interface member surrounding the centrally located heat pipe. For a power output of 300 watts at a load voltage of 32 volts, the outer case of the STG has a length of 11.34 inches, a diameter of 1.61 inches and possesses eight radiation fins extending the length of the outer case and having heights of about eight inches. This means that the overall radial extension of the body of the STG is approximately 18 inches. The solar concentrator possesses a diameter slightly less than 40 inches and has a height of almost the same numerical value. A gap of about 2.7 inches separate the base of the solar concentrator from the body of the STG, this gap being spanned by one end of the heat pipe. The total height of the STG, including the cylindrical outer case, the heat pipe extension and the solar concentrator, is about 54 inches.

For purposes of calculation, it has been assumed that the heat pipe is made of a refractory material such as tungsten. The polygonal hot side heat exchanger within the STG is assumed to be made of molybdenum. The thermal insulation within the STG and surrounding the heat pipe outside the STG is assumed to consist of Min-K 2020. The STG outer case, the radiation fins and the solar concentrator are all assumed to be made of beryllium. Using these assumptions, it is calculated that the total weight of the STG is about 38 pounds. This means that the optimum specific power for the cylindrical STG in operation at 0.45 AU is about 7.89 watts per pound. It is noted that a STG of this type accordingly possesses a lower specific power than either a compact STG or a distributed STG with multiple solar concentrators. It does possess a slightly higher specific power than the flat-plate STG.

Although the specific power of the cylindrical STG with a single solar concentrator is lower than the specific power of either the compact or the distributed STGs, the cylindrical STG does present a few advantages.

cylindrical STG have not been as complete and exhaustive as on the other types of STGs. For this reason, a STG of the cylindrical type should not be totally precluded from future consideration of STGs in near sun missions.

#### E. Alternate Hot Side Heat Exchanger Design

The STG designs considered on the present study have utilized either solar concentrators or flat-plate collectors; the solar concentrator design considered on the program is based on a truncated cone. Prior to the completion of the discussions on STG designs, it is necessary to mention that another type of hot side heat exchanger can be considered. Namely, either a single solar concentrator or multiple solar concentrators can be replaced by Fresnel lens. A relatively large single solar concentrator can be replaced by a Fresnel lens having the same extension in the plane perpendicular to the sun as the total cross-sectional area of the solar concentrator. The lens would be located such that it focuses the incident solar flux onto the total area of the thermoelectric generator. This means that the lens would be located at a distance from the generator that is slightly either greater than or less than the focal length of the lens. In this way the total surface area of the generator would be relatively uniformly heated.

Fresnel lens can also be used to replace the multiple concentrators of the distributed STG. In this case, the Fresnel lens can be either individual lens or can consist of a single large sheet with individual lens inscribed into it. Each lens will act as the hot side heat exchanger for a single thermopile within the STG. As with the compact STG, the lens would be located such that the total cross-section of each thermopile is heated by incident solar flux being concentrated by the lens. This means that the lens would be located at a distance from the thermopiles slightly other than the focal length of the lens.

Although the present study did not address itself to analyses on the weight of STGs that use Fresnel lens, it may be postulated that STGs of that type would very likely possess weights of the same general order as STGs using solar concentrators. Possibly the main disadvantage of the use of Fresnel lens is the mechanism of mounting of the lens in such a way that proper heating of the thermopiles within the STG is always possible. Misalignment or shifting of the lens at any time might make the whole STG inoperable. Of course, this is also true of a STG that uses a single solar concentrator; it is not, however, true of a distributed STG with multiple concentrators. Another possible disadvantage of Fresnel lens as the hot side heat exchanger of STGs is the criticality of the alignment with the sun. Fresnel lens would be essentially inoperable if they are not at all times pointing directly to the sun. In other words, very likely the cosine law that can be applied to solar concentrators and flat-plate collectors does not apply to Fresnel lens. This presents a design problem because the STG cannot be designed to operate at its maximum permissible temperature at a distance of 0.45 AU and over-temperaturing prevented by the tilting of the STG on closer approaches to the sun. It will be necessary to design the STG to operate at its maximum permissible temperature at its closest approach to the sun. This means that the STG must be oversized in order that adequate power be available at distances other than its closest distance to the sun. Finally, if Fresnel lens are used they will have to be fabricated from a material, such as quartz, that is capable of high temperature operation without the lens being distorted or even destroyed.

## V. DYNAMIC STG PERFORMANCE CHARACTERISTICS

Various STG design concepts were investigated in the preceeding section. It was seen that the STG design that utilizes distributed thermopiles, with each thermopile having an individual solar concentrator, possesses the lowest weight and highest specific power. It also possesses a panel-like configuration that enables its convenient storage and deployment during mission. For these reasons it is this STG design that has been selected as the recommended design for a STG intended for use in a Mercury orbiter mission. The characteristics of the selected STG design are detailed in Table 5 in Section IV. The STG performance data shown in that table pertain to STG operation at 0.45 AU from the sun, with the STG directly facing the sun. At closer distances to the sun, while still directly facing the sun, STG performance and operating temperatures will remain at the values shown for 0.45 AU if the STG is tilted according to the tilt angels given in Table 4 in Section IV. Of course, this presupposes the STG being exposed to direct sunlight during the whole time. In its intended mission, however, the STG will be orbiting the planet Mercury and therefore will be periodically shielded from the sun by the planet. Even though the planet itself emits radiation, the amount thus emitted is negligible to that emitted by the sun and during the eclipse portion of its orbit the STG can be considered, for all practical purposes, to receive no incident heat. Inasmuch as the spacecraft requires power not only during the times that it is without direct sunlight, but also while it is eclipsed by the planet, it is necessary to consider the effect of periodic eclipsing on the output of the STG. For this purpose, a mathematical model was adopted and utilized to calculate STG temperatures and performance as a function of time when it is no longer receiving incident heat from the sun.

The mathematical model used to perform time dependent analyses of STG temperatures and performance assumes a parallelepipedal configuration with insulated sides. Only the two heat exchanger surfaces are assumed to interact

with the environment. In fact, at the moment that the STG is eclipsed by the planet Mercury, it is assumed that both heat exchangers of the STG radiate freely into the environment. At the moment of the start of eclipse, the STG is assumed to possess a linear temperature gradient between its hot and cold faces. This temperature gradient,  $f(x)$ , is defined in terms of STG cold side temperature,  $T_C$ , the temperature difference,  $\Delta T$ , across the STG and the thickness of the STG as follows:

$$f(x) = T_C + \left( \frac{\ell + x}{2\ell} \right) \Delta T \quad (45)$$

where  $2\ell$  is the total thickness of the STG and  $x$  indicates the location within the STG between  $-\ell$  and  $+\ell$ . Thus, the cold side heat exchanger of the STG is designated by  $x = -\ell$  and the hot side heat exchanger by  $x = +\ell$ . A second order partial differential equation in the space and time variables provides STG temperatures as a function of location within the STG as a function of time. The boundary conditions assumed for the solution of the differential equation pertain to radiative heat loss from the two heat exchangers; even the hot side heat exchanger loses heat during the eclipse mode of STG operation. Inasmuch as the differential equation is soluble in closed form only for linear boundary conditions, it is assumed that radiative heat loss can be represented by the so-called Newton's law of cooling. This law assumes heat transfer to occur proportionately to the product of the third power of temperature and the temperature difference between the cooling body and its environment. Although this boundary condition will be applied to the case of the STG in a situation where relatively large temperature variations occur, it must be noted that it is strictly rigorous only when temperature variations are small. Nevertheless, the use of the boundary condition enables the solution of the problem and enables an approximate assessment of STG performance and temperatures as it enters the shade portion of its orbit around Mercury. The solution of the appropriate differential equation, along with the stated boundary conditions

may be represented as\*:

$$T = \sum_{n=1}^{\infty} e^{-D\alpha_n^2 t} \frac{[c_n \cos \alpha_n x + d_n \sin \alpha_n x]}{(\alpha_n^2 + h^2)^{1/2}} \int_{-l}^l f(x) [c_n \cos \alpha_n x + d_n \sin \alpha_n x] dx, \quad (46)$$

where

$$c_n = h \sin \alpha_n l + \alpha_n \cos \alpha_n l, \quad (47)$$

$$d_n = h \cos \alpha_n l - \alpha_n \sin \alpha_n l, \quad (48)$$

and the  $\alpha_n$  are the positive roots of

$$\tan 2\alpha l = \frac{2\alpha h}{\alpha^2 - h^2}. \quad (49)$$

In the above relationships, it should be noted that  $h$  is the heat transfer coefficient from the hot and cold side heat exchangers and  $D$  is the effective thermal diffusivity of the STG. Inasmuch as the STG consists of a number of different materials, each having unique temperature dependent values of thermal diffusivity, it has been necessary to average these values in order to obtain a single effective thermal diffusivity for the STG. This has been done by calculating the total heat capacity of the STG, reducing this to a net specific heat and then using average values of density and thermal conductivity, calculating thermal diffusivity as follows:

$$D = \frac{k}{\rho C}, \quad (50)$$

\* H. S. Carslaw and J. C. Jaeger, Conduction of Heat In Solids, Second Edition, Oxford University Press, 1959.

where  $k$  is the thermal conductivity,  $\sigma$  is the density and  $C$  is the specific heat of the panel.

Substituting the temperature gradient,  $f(x)$ , into Eq. (46) and solving for temperature,  $T$ , for  $x = -l$  and  $x = +l$  gives STG hot and cold side temperatures as a function of time. The difference of these temperatures as a function of time gives the temperature differential across the thermopiles within the STG. Utilizing this temperature differential, it is possible to calculate the performance of the STG, such as power output and load voltage, as a function of time. This has been done for the distributed STG with multiple solar concentrators and results are displayed in Table 7 in terms of the power output, the load voltage and the temperature differential across the thermopiles as a function of time; initial time pertains to the instant at which the STG becomes eclipsed by the planet. It is noted in Table 7 that the temperatures and performance of the STG rapidly decrease when the STG is no longer exposed to direct sunlight. Within some three to four minutes after entering shade, the STG performance has decreased to approximately one-half the value it possesses in direct sunlight. This means that during the eclipse portion of its orbit, the spacecraft cannot depend on the STG for electrical power and such power must be obtained from a storage battery that is recharged during the sunlit portion of the spacecraft orbit. Upon re-emergence from behind the planet, the STG rapidly heats and once again starts to provide electrical power. The STG heat-up curve, as concerns temperatures and performance is a mirror image of the cool-down curve as represented by Table 7. This means that during each orbit, the STG provides nearly its design performance for 45 minutes and very little performance during the remaining 45 minutes. If a storage battery is used to provide spacecraft power during the eclipse portion of each spacecraft orbit, the STG must be capable of providing twice its required performance during the portion it is in direct sunlight. This means that the STG design must be formulated for 600 watts rather than for 300 watts, if 300 watts are required to power the spacecraft. Total STG weight and size therefore double over the values shown

TABLE 7

Time Dependent STG Performance in Eclipse Mode

<u>Time/Seconds</u>	<u>Temp. Differential °C</u>	<u>Load Voltage/Volts</u>	<u>Power Output/Watts</u>
0	550	32.0	300
60	473	27.5	222
120	414	24.1	170
180	361	20.9	129
240	312	18.2	97
300	272	15.9	74
360	238	13.8	56

in Table 5 of Section IV. A total power system weight analysis, must also consider the weight of the storage batteries. The actual specific power of the total power system is therefore considerably less than the value shown in Table 5.

The above thermal calculations on the transient behavior of STG temperatures and performance indicate that the total heat capacity of the STG is relatively low. In other words, the STG is incapable of maintaining heat and therefore its performance while it is eclipsed by the planet. It is for this reason that the use of storage batteries in conjunction with the STG must be considered for the overall spacecraft power system. As a part of the present study, another possibility was investigated. Namely, it was considered that possibly the total heat capacity of the STG could be increased by the use of a high heat capacity molten salt bath located at the hot side of the STG. In other words, a molten salt bath, enclosed in a metal container, was considered to be placed in intimate thermal contact with the hot sides of the thermopiles within the STG; this is most conveniently done in the compact STG design in which all of the thermopiles are placed adjacent to each other. The face of the molten salt bath opposite to the one in contact with the thermopiles is envisioned to be directly heated by the solar concentrator. The sides of the molten salt bath are insulated such that heat enters from the concentrator side of the bath and leaves through the thermopiles. Considering the heat capacities of various molten salts, it was found that various alkali metal hydrides possess the highest values of specific heat. By using the specific heat capacities and densities of typical alkali metal hydrides and by knowing the total heat requirements of the STG as a function of its operating temperature difference, it is possible to write an equation for the specific power of the STG as a function of the amount of cooling that takes place upon the entry of the spacecraft into the shade portion of its orbit. Assuming a value of specific power of 22.17 watts per pound for the distributed STG design given in Table 5, and assuming that power output varies as the square of the temperature difference across the thermopiles, the following expression may be written for the specific power of the same STG if it uses molten

salt baths for power flattening:

$$P_s = \frac{300 \left[ \frac{550 - \Delta T_L}{550} \right]^2}{\left[ \frac{550}{550 - \Delta T_L} \right]^2 13.533 + \frac{7114.79}{\Delta T_L}} \quad (51)$$

where  $\Delta T_L$  is the decrease in the temperature difference across the thermopiles. It is noted in Eq. (51) that the second term in the denominator represents the weight of the molten salt required to provide adequate heat to power flatten the STG if the molten salt is permitted to decrease by  $\Delta T_L$  in temperature during the shade portion of the orbit. It is seen that if the change in temperature is minimized to as small a value as possible, the weight of the molten salt rapidly increases. On the other hand, if the temperature is permitted to considerably vary the STG itself rapidly increases in size in order that the required power output be maintained. A variation of  $\Delta T_L$  yields an optimum in terms of a specific power of the total power flattened STG. Table 8 shows the results of this optimization procedure. It is seen that a temperature variation of  $125^\circ\text{C}$  on a total initial temperature difference of  $550^\circ\text{C}$  results in the lowest weight system. As seen in Table 8, unfortunately, however, the specific power of the optimum STG has decreased by an order of magnitude when compared to the STG without power flattening (See Table 5). What makes this result even more significant is the fact that the molten salt weight in Eq. (51) does not include the weight of any structural or containment members required to contain the molten salt. This means that the specific power reduction will be even more drastic than is apparent in Table 8. For this reason, it is concluded that the use of a molten salt bath to power flatten STG performance is not realistic and severely penalizes total system weight. It is more realistic to consider the STG as defined in Table 5 and to use storage batteries for purposes of power flattening. If storage battery weight is appreciable, this conclusion may have to be somewhat modified. In any case, however, it is thought that the use of storage batteries in conjunction with a distributed STG results in a power conversion system of lowest weight and it is this kind of a system that is therefore recommended for the Mercury orbiter mission.

TABLE 8

Effect of Power Flattening By Molten Salt Bath  
On STG Specific Power

<u><math>\Delta T</math></u>	<u><math>P_S</math></u>
0	0
10	0.40
25	0.91
50	1.56
75	1.98
100	2.20
125	2.25
150	2.17
175	2.00
200	1.76
225	1.49
250	1.21

## VI. SUMMARY

The present study on the use of a STG in a Mercury orbiter mission has addressed itself first to the selection of the most appropriate thermoelectric technology, second to the selection of the most appropriate STG design configuration and third to the weight optimization and performance characterization of the final STG design. This design has then been analyzed in terms of its steady state and dynamic performance in a Mercury orbit, both as regards a given orbit around Mercury as well as its performance at different times while Mercury orbits the sun; the latter consideration is of course important because of Mercury's extremely eccentric orbit that varies between 0.30 and 0.45 AU from the sun.

The main conclusions of the present study may be summarized as follows:

1. The most appropriate thermoelectric technology applicable to STGs intended for use in space is one based on silicon-germanium alloys. The use of this technology enables the design and fabrication of the lightest weight and most reliable STG.
2. The most appropriate STG design concept is based on one in which the individual thermopiles are uniformly distributed over the radiator and in which the thermopiles are heated by individual solar concentrators. This design concept enables the fabrication of a STG that possesses a specific power in excess of 22 watts per pound when operating at a distance of 0.45 AU from the sun. Although an exhaustive investigation of various concentrators was not included in the study, it is seen that a concentrator configured as a truncated cone results in an attractive STG design. Possible other concentrator configurations can only enhance the design.
3. A STG in a Mercury orbit is incapable of producing appreciable power while eclipsed by the planet. This means that power must be stored during the time the STG is exposed to direct sunlight; it is this stored

power that powers the spacecraft during the eclipse mode. This being the case, the STG must be designed for at least twice the spacecraft power requirement. While in direct sunlight, half of the power produced by the STG goes into storage; the STG is directly exposed to the sun during essentially one-half of each of its orbits around Mercury.

4. STG energy storage must be performed electrically rather than thermally because it is the former mode of storage that enables very likely the lowest overall system weight to be obtained. Thermal storage in terms of the use of various liquid metal baths penalizes total STG weight by nearly a factor of ten. System weight increases as a result of the use of chemical storage batteries is very likely considerably less.
5. The state-of-the-art of silicon-germanium alloy technology is such that a STG intended for space application in a near sun mission does not require any special development. All of the components required of the STG presently exist in a developed form. Although this is true, the optimum STG design concept herein selected has never been assembled or tested.
6. Additional design effort is recommended in the area of solar concentrators in order that the most optimum concentrator design be available for a STG. The present study has been based on the use of a solar concentrator configured as a truncated cone; this configuration may not necessarily represent the most optimum concentrator shape.
7. Assumptions pertaining to the absorptivity, reflectivity and emissivity of the solar concentrator and the hot side of the thermoelectric generator and/or thermopiles are based primarily on conjecture and therefore must be established in detail prior to a final STG design. The emissivity and absorptivity values assumed for the radiator of the STG are considered reasonable and easily attainable.

## VII. APPENDIX

In order to illustrate the method used in the design of the various STGs considered by the present study, sample calculations for typical flat-plate and compact STGs are given in this appendix. The equations underlying the sample calculations have been given in Section II and are not repeated here.

The typical design calculation of the flat-plate STG includes Eqs. (1) to (12) in Section II. As explained in that section, the calculational sequence is iterative and for purposes of illustration, the sample calculation has been iterated 19 times. Most of the input parameters are listed for the sample calculation, although material property values have not because those are generally temperature dependent variables that are introduced into the calculational sequence at the beginning of each iteration; the values are selected on the basis of the temperatures calculated in the iteration preceeding the one for which the properties are input. It is noted that in the flat-plate sample calculation all of the variables converge towards final values. It is also noted that convergence is not complete after 19 iterations, although the variation between parameters at that point is relatively small. The calculation is generally continued until convergence is obtained within any arbitrarily predetermined convergence criterion.

The sample calculation for the compact STG corresponds to Eqs. (22) to (43) in Section II. Unlike the flat-plate STG calculation, that for the compact STG is noniterative. For this reason, the calculation is complete after all of the parameters have been calculated once. As with the flat-plate STG calculations, some of the input parameters used in the sample calculation of the compact STG are given.

Flat-Plate STG  
Sample Calculation

Input Parameters:

$N=400$        $\gamma=\beta=0.05$        $W=0.69136$   
 $M=400$        $\alpha=\epsilon_R=0.85$        $T_A=0$   
 $A=1 \times 10^{-3}$        $\epsilon_C=0.10$        $W_{gl}=0.00508$   
 $\ell=0.8$        $R_F=1.1$        $A_C=10,000$

Iteration Number

	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19
$Q_T$	5877	2916	4127	3706	4726	4042	4524	4196	4425	4268	4378	4300	4354	4316	4344	4324	4338	4328	4335
$T_R$	590.9	489.3	570.4	522.9	558.3	535.4	551.9	540.8	548.6	543.3	546.0	544.4	546.2	544.9	545.9	545.2	545.7	545.4	545.6
$T_G$	590.9	501.7	575.3	533.0	564.9	544.4	559.2	549.3	556.3	551.6	554.9	552.6	554.2	553.0	553.9	553.3	553.7	553.4	553.6
$s$	$3.16 \times 10^{-2}$ —————→																		
$A_{gl}$	$3.47 \times 10^{-4}$ —————→																		
$S_{TP}$	1.043 —————→																		
$H_{TP}$	0.848 —————→																		
$A_s$	9565 —————→																		
$\Delta T$	246.8	96.6	201.2	131.6	180.3	147.1	170.2	154.3	165.3	157.6	163.0	159.3	162.1	160.1	161.4	160.5	161.1	160.7	161.0
$E_L$	14.59	5.71	11.89	7.77	10.65	8.69	10.06	9.12	9.77	9.31	9.64	9.42	9.58	9.46	9.54	9.48	9.52	9.49	9.51
$P$	153.1	23.4	101.8	43.5	81.6	54.4	72.8	59.9	68.7	62.4	66.8	63.8	66.0	64.4	65.5	64.7	65.2	64.9	65.1
$T_H$	837.7	598.2	776.5	664.6	745.2	691.5	729.4	703.6	721.6	709.2	718.0	711.9	716.3	713.1	715.3	713.8	714.8	714.1	714.6

$$N_D = \frac{32.00}{9.5} \times 400 = 1347$$

$$M_D = \frac{300.00}{65.00} \times \frac{9.5}{32.00} \times 400 = 548$$

$$A_{CD} = \frac{548}{400} \times \frac{1347}{400} \times 10,000 = 46,155$$

Equations (16) to (21) in Section II can be used to calculate the weight and specific power of the STG.

Compact STG  
Sample Calculation

Input Parameters:

$$\epsilon_R = 0.85$$

$$\alpha = 0.85$$

$$\epsilon_G = 0.10$$

$$T_H = 1273^\circ\text{K}$$

$$W = 0.691358 \text{ watt/cm}^2$$

$$\Delta T = 550^\circ\text{K}$$

N=	339.3	Q <sub>J</sub> =	125.00
$\frac{l}{A}$ =	1769.0	Q <sub>T</sub> =	5453.90
A=	$5.6529 \times 10^{-4}$	T <sub>R</sub> =	695.50
A <sub>gl</sub> =	$2.6737 \times 10^{-4}$	A <sub>R</sub> =	4570.31
s =	$2.3776 \times 10^{-2}$	S <sub>R</sub> =	67.604
S <sub>TP</sub> =	0.75681	W <sub>R</sub> =	3325.44
H <sub>TP</sub> =	1.04826	A <sub>C</sub> =	9913.12
W <sub>TP</sub> =	1.6787	D <sub>C</sub> =	112.35
W <sub>G</sub> =	671.48	D <sub>CS</sub> =	41.61
S <sub>G</sub> =	15.1362	t <sub>S</sub> =	1.04826
A <sub>G</sub> =	229.105	W <sub>S</sub> =	558.23
I <sub>L</sub> =	10.7143	H <sub>C</sub> =	104.02
Q <sub>K</sub> =	4263.14	W <sub>C</sub> =	7380.07
Q <sub>S</sub> =	42.77	W <sub>T</sub> =	11,935.22
Q <sub>P</sub> =	1272.99	P <sub>S</sub> =	11.403