THERMOELECTRIC
SOLAR ENERGY CONVERTER

by

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STATEMENT BY AUTHOR

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THERMOELECTRIC SOLAR ENERGY CONVERTER

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ABSTRACT

The first three chapters of this study deal with the selection of the best type of solar collector for use with thermoelectric material and a discussion of thermoelectric material to be used. A proposed design incorporating a shielded focal tube receiver is presented. This entire device is for research purposes only and the cooling necessary for controlled results is quite elaborate. This would not be necessary in some climates and certainly not in outer space. A small refrigeration unit has also been incorporated for research measurements.

The general conclusion of the study is that the present cost of thermoelectric material makes the cost of the proposed device prohibitive. If at some future date, the price of thermoelectric material is drastically reduced, as is expected, the proposed device may become feasible to construct.

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CHAPTER 1

INTRODUCTION

1.1 Statement of the Problem. The problem is to design a device for the conversion of solar energy into electrical energy and electrical energy into refrigeration, incorporating thermoelectric generation in both conversions.

1.2 Justification of the Proposed Device. If the present rate of consumption of fossil fuels is maintained this energy source may be exhausted before many years have passed. Even nuclear energy may be insufficient to close the gap. It is, therefore, imperative that some other sources of energy be found and all known possibilities be investigated. One ever-present source is solar energy. At Tucson, Arizona, for instance, the average insolation is about 275 Btu per hr per sq ft.[17a] An economical means of converting this energy to usable power would be a boon to all mankind. In particular it would benefit those who live and work in remote areas where power is unavailable or prohibitively expensive. The proposed device is a step in this direction.

Furthermore, in this present age of space flight we are faced with existence in outer space. To exist in a new environment man must investigate ways and means of using the natural elements of that environment and protecting himself from them. One natural element provided in these new surroundings of space is solar energy.

At the outer fringes of the earth's atmosphere the source of usable energy averages out to about 442.4 Btu per hr per sq ft.[18a] Obviously, the more of this available energy which is used in space travel the less fuel will have to be carried in other forms. However, this source is limited in that there is only a given amount per unit area available. Therefore, the conversion must be efficiently done to be of any appreciable value. To this end, then, the design of this device is undertaken to investigate ways and means of tapping this vast source of energy.

1.3 Procedure. First, a discussion will be included regarding the choice of solar energy collector type. Next the results of an analysis will be presented concerning the thermoelectric elements which were selected for the final design. Then, a design will be proposed for use on the earth's surface, and finally, a short discussion of present-day costs and future uses will be presented.

The literature is so replete with history on this subject that only so much as is necessary for clarity within the body of the thesis will be covered. Allais, in his thesis, gives an excellent historical background of solar collectors.[1a]

As for thermoelectricity, the article by Zener, "The Impact of Thermoelectricity on Science and Technology" gives a very fine historical background.[4a]

CHAPTER 2

SOLAR COLLECTORS

2.1 Introduction. Of the many varieties of solar collectors, three main configurations will be briefly discussed. These are the flat plate, the spherical parabola, and the cylindrical parabola type collectors.

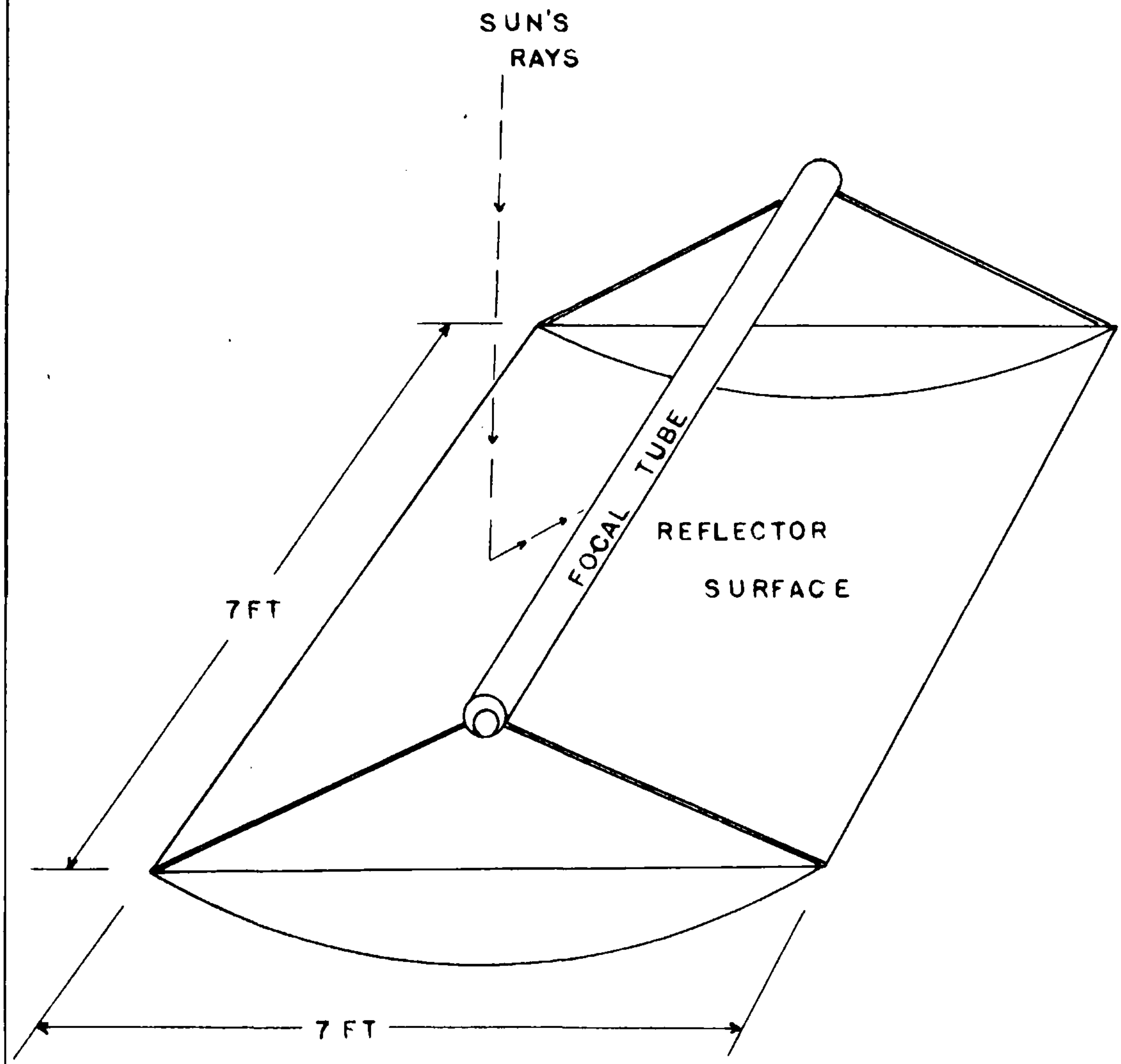
2.2 The Flat Plate Collector. The flat plate type of collector is the simplest and least expensive type to build but has no concentration capability. As mentioned, in Tucson, Arizona, the average insolation is about 275 Btu per hr per sq ft.[17a] This then is the average energy available per square foot with a flat plate collector. However, as will become apparent later, thermoelectric elements are most effective at temperature differences within certain limits. The upper limit of the most efficient material to date is in the vicinity of 300°F. Previous work with flat plate collectors indicates that at temperatures in this range the efficiency of collection is only about 20% to 40% under the best conditions.[16a] For this reason the flat plate collector was not selected for use in the energy conversion device.

2.3 The Spherical Parabolic Collector. As opposed to the flat plate, the spherical parabola type is an excellent concentrator of solar energy. Practical concentrations in excess of 10,000 to 1 can be obtained with an efficiency of 70%.[15] In practice, large collectors are mounted in fixed positions and tilted, flat orientator mirrors are used as heliostats to focus the radiation into the spherical parabola. As an example of the operating temperatures obtained in these solar collectors, the Quartermaster Research and Engineering Center in Massachusetts has a 28 square foot (projected area) furnace in which temperatures up to 4,000°F. were achieved.[12]

This degree of concentration and such high temperatures are more than are needed. As a matter of fact, the highest temperature that is possible to use with any solid thermoelectric material is about 1400°F.[4a]

2.4 The Cylindrical Parabolic Collector. The cylindrical parabolic type of collector is not as simple to construct as the flat plate type, but the problem of orientation with respect to the sun is far less complicated than for the spherical parabolic collector. When using the cylindrical parabola type of concentrator with its axis placed in a North-South direction, it is necessary to provide a sun tracking mechanism which will provide east to west rotation only. The effect of the sun's declination may be corrected for by making periodic manual adjustments. However, this correction may be neglected. Any energy loss incurred due to lack of correction for sun's declination can be recovered by use of a mirror compensator. The mirror compensator consists of a plane mirror which would be installed perpendicular to the axis of and at the ends of the parabolic cylinder mirror. This compensator would intercept the otherwise lost radiation and re-direct it to the receiver. Refer to reference 1 or 2 for a complete description. A thorough discussion of the problem of tracking in two dimensions is given by Bowman in his thesis. This discussion gives complete construction details of the solar collector which was used at the University of Arizona.[2a]

The type of collector selected for use in the energy converting device to be proposed here will be of the cylindrical parabola type. A diagrammatic sketch of this device is shown in Fig. 2.1.



SKETCH OF
CYLINDRICAL PARABOLIC SOLAR COLLECTOR
SHOWING SOLAR RAY REFLECTION

FIG 2.1

CHAPTER 3

THERMOELECTRIC ELEMENTS

3.1 Theory.[9a] Seebeck, in 1821, noted that a magnetic needle held near a circuit of two different conductors was deflected when the conductors were heated. Peltier, in 1834, discovered that a current passed through a junction of two different conductors had a thermal effect. Although these two men gave their names to the phenomena discovered, neither realized what he had found. Actually, there are four distinct thermoelectric actions: the Joule effect, the Thomson effect, the Peltier effect, and the Seebeck effect. These actions have to do with the conversion of electrical energy to heat and heat to electrical energy.

3.1.1 The Joule Effect. Heating, which is caused by the Joule effect, is found in all electric circuits and is controlled by I^2R where I is the amount of current and R the resistance. This heating effect is produced in the conductor regardless of which way the current travels. Unlike the other three effects to be discussed, heating a simple conductor material will not produce a current.

In a thermoelectric element this heating is always present and is distributed throughout the material. Therefore, when considering the "hot" end of a conductor, the Joule heat often is assumed to have no preferential directional effect.

3.1.2 The Thomson Effect. The Thomson effect is a result of the change of entropy as a current in a simple homogeneous conductor passes from an area of one temperature, T, to another area of a different temperature, $T + \Delta T$. Thus, energy in the form of heat is either absorbed or evolved depending upon the direction of the current.

3.1.3 The Peltier Effect. The Peltier effect is also a result of the entropy change as current flows, but it takes place when the current flows from one type of conductor material to another type of conductor material. Since this entropy is roughly related to specific heat, when the current flows from one material to another, heat must either be evolved or absorbed to account for the change of atomic structure at the junction of the two materials.

Both the Peltier and Thomson effects are similar in nature and may produce either a heating or a cooling process, depending upon the direction of current flow.

3.1.4 The Seebeck Effect. The Seebeck effect results when a temperature differenct is established between the junction of the two dissimilar materials. An electromotive force appears between the junctions. This is the Seebeck emf.

3.1.5 Figure of Merit. Many books, notably among them, Thermoelectricity[4] and Direct Conversion of Heat to Electricity[9], give the detailed development of the figure of merit, Z, for semiconductor thermoelectric materials. Therefore it will be merely stated that the figure of merit is a parameter which involves the thermoelectric power, or Seebeck coefficient, S, in volts per ^{o}F, the thermal conductivity of the material used, k, in Btu per hr ft ^{o}F, and the electrical resistivity of the material used, ρ, in ohms. The relation between these terms is

$$Z = \frac{S^2}{k\rho} \text{ deg}^{-1} \tag{3.1}$$

The thermoelectric power, S, is related to the rate of change of the Seebeck emf, θ, with the temperature, T, or

$$S = \frac{\partial \theta}{\partial T} \tag{3.2}$$

3.1.6 The Thermocouple Operating Efficiency. For generation of electricity, the optimum efficiency, η, incorporates the figure of merit, Z, and the temperature difference between the hot junction, T_h, and the cold junction, T_c, as follows:

$$\eta_{max} = \frac{(T_h - T_c)}{T_h} \left[\frac{\sqrt{1+Z(T_h+T_c)/2}-1}{\sqrt{1+Z(T_h+T_c)/2}+T_c/T_h} \right] \qquad (3.3)$$

This is the maximum electrical output for a given thermal input.

An inspection of the equations for Z and η indicates that if a high figure of merit and a high efficiency are desired, the product, $k\rho$, and T_c must be minimized and the Seebeck coefficient, S, and T_h should be maximized. Unfortunately thermal conductivity and electrical resistivity cannot be separated and as the magnitude of one is reduced the other will increase and vice versa. Thus, little change can be made in the $k\rho$ product. However, the figure of merit can be improved by changing S and this is a squared quantity.

3.2 Element Construction. The basic thermoelectric element is constructed of an n-type material, one in which the electric current is electron transported, and a p-type material, one in which the current is hole transported. The n-type exhibits a negative voltage at the cold end with respect to the hot end. The p-type exhibits a positive voltage at the cold end with respect to the hot end. Thus the elements are arranged thermally in parallel and electrically in series. The metal electrodes used for mounting have such small resistances that they can be ignored.

Fig. 3.1 shows such an element.

3.2.1 Optimum Sizes of Thermoelectric Elements.

As shown in Fig. 3.1 the length of the n-type and p-type elements is the same in normally constructed units. The cross-sectional area relationships for optimized design is[9c]

$$\frac{A_n}{A_p} = \sqrt{\frac{k_p \rho_n}{k_n \rho_p}} \qquad (3.4)$$

where

A = cross-sectional area of each leg

k = thermal conductivity

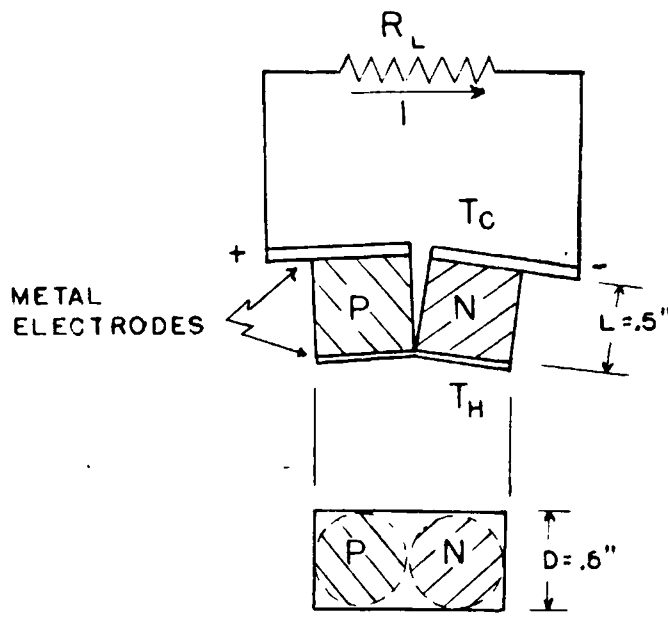
ρ = electrical resistivity

n = subscript referring to n-type material

p = subscript referring to p-type material

This equation indicates that with a construction of equal length legs of n-type and p-type material, the relative areas are the only optimizing factors. Going one step further, it may be concluded that utilizing the smallest elements in a device would be best in terms of material economy. The length-to-area ratios resulting from optimization are given by[9c]

$$\frac{L_n}{A_p} = \frac{R_g/N}{\rho_n\left(\frac{\rho_p\, A_n}{A_p}\right)} \qquad (L_n = L_p) \qquad (3.5a)$$



BASIC THERMOELECRIC CIRCUIT

FIG 3.1

and

$$\frac{L_p}{A_p} = \frac{R_g/N}{\rho_p \frac{(\rho_n A_p)}{A_n}} \qquad (L_n = L_p) \qquad (3.5b)$$

where

L_n and L_p = lengths of n-type or p-type material

N = the total number or elements used

R_g = internal resistance, ohms

As long as these ratios are maintained, the limits to dimensions for and fabrication of thermoelectric elements are only mechanical in nature. Special care must be taken during the joining process between the semiconductor material and metal electrodes used for mounting. The electrical resistance which is inherent in any such connection must be minimized. This makes miniaturized elements very difficult to fabricate.

3.3 Material Selection. By proper doping, a semiconductor material can be made to perform as an n-type or a p-type. There are of course an infinite variety of possibilities. The General Electric Company, under Department of the Army Contract No. DA-130-115-ORD-960, May 13, 1958-March 1, 1959, has published A Study of Thermoelectric Generator Materials and included a chart, Fig. 3.2. This chart was reproduced with permission of Harold F. Gibson of

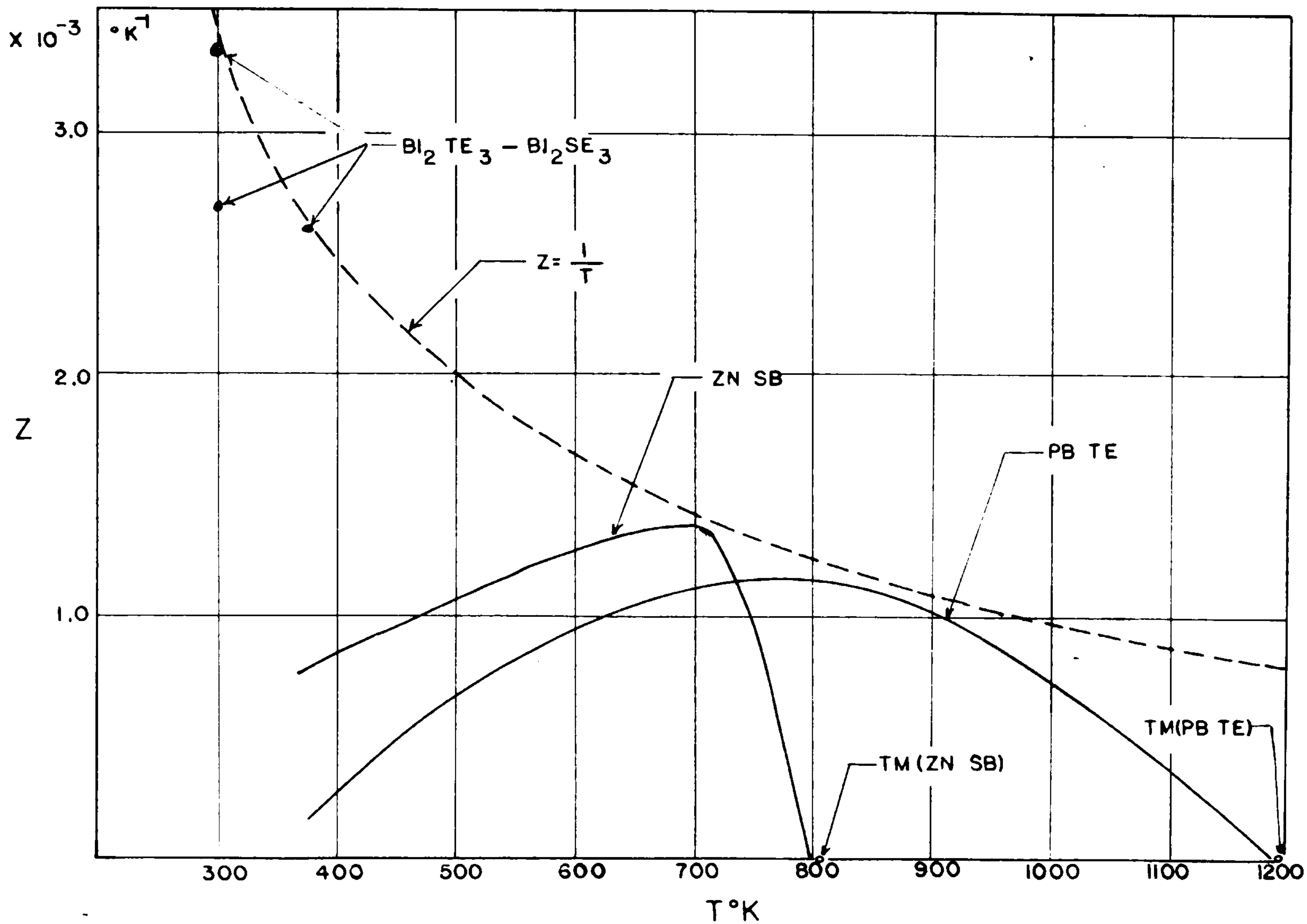


FIG 3.2 TEMPERATURE DEPENDENCE OF SOME FIGURES OF MERIT

The Diamond Ordnance Fuze Laboratories, Washington 25, D. C. Fig. 3.2 presents data which were obtained experimentally and are plotted as Figure of Merit vs. Temperature for several different substances.[5] The temperatures given on this chart are those for the hot junction. It is interesting to note that the dotted line, $ZT = 1$, indicates a limit beyond which no substances have been found to operate, as of this writing.

From the standpoint of the present design problem the temperature of 300°K (80.7°F) to 500°K (440°F) is most important since this is the general temperature zone in which the cylindrical parabolic solar radiation collector operates. Fortunately this is also the temperature zone in which the highest figure of merit is found which in this case is for bismuth telluride (Bi_2Te_3); a Z of about $3 \times 10^{-3} deg^{-1}$ is indicated. This would, therefore, indicate that the material to be used in the presently proposed design should be bismuth telluride.

CHAPTER 4

PROPOSED DESIGN FOR SOLAR ENERGY CONVERTER

4.1 General Design. The general design of this solar energy converter will incorporate a solar concentrator of cylindrical parabolic shape and thermoelectric elements of bismuth telluride as the conversion material. The design of the converter will now be discussed in the following general sequence:

a. The concentrating mirror

b. The focal tube receiver

c. The cooling system

d. The refrigeration unit

An element of length of one foot will be used in initial calculations for the concentrating mirror and receiver.

4.2 Assumptions. Before discussing the design in detail the assumptions upon which the design is based will be listed. These assumptions are as follows:[17]

a. Ambient air temperature = 68°F (530°R)

b. Average wind velocity = 8 mph (11.75 fps)

c. Average daylight hours per year =

$$\frac{4442}{365} = 12.16 \text{ hrs per day}$$

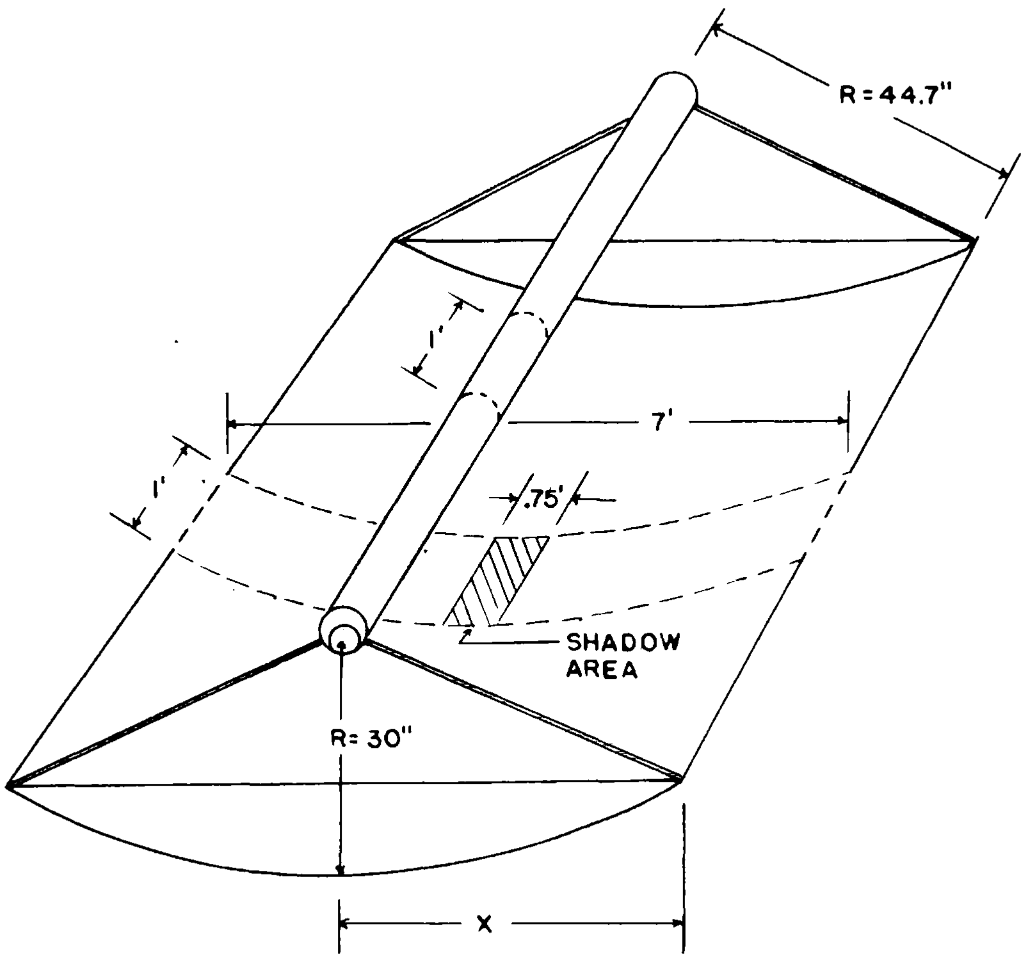
d. Average sunshine hours per day =

$12.16 \times 0.85 \approx 10$ hrs per day

e. Average insolation is about 275 Btu per hr per sq ft. All values listed refer to the region around Tucson, Arizona.

4.3 The Concentrating Mirror. The concentrator proposed for use consists of a 7 ft. by 7 ft. parabolic cylindrical mirror mounted with its axis inclined at 32 degrees 13 minutes from the horizontal in a north and south line.[17b] The north end of the mirror is elevated above the south end. The entire mirror surface is rotated by 1/150 hp motor at the rate of 15 deg per hr to maintain east to west solar focus during operation. The surface is constructed in accordance with the equation $x^2 = 120y$.[1b] The sunlight incident on the mirror surface will be concentrated at a point which is R distance from that surface. The value of R varies from 30 inches, directly above the vertex of the parabola, to 44.7 inches from the outer edge.[1c] See Fig. 4.1. This reflector was first built according to the design of Dr. C. G. Abbot and later modified.[2c]

The reflecting surface is of bright aluminum with a reflectivity, $r = 80\%$.[1d] The presently modified mirror surface has an effectiveness of 88.4%.[2d] This means that due to inherent imperfections in the construction of the concentrating mirror surface a ray of light impinging on



DIAGRAMMATIC SKETCH OF
SECTIONALIZED COLLECTOR

FIG 4.1

the surface has an 88.4% chance of being accurately reflected to the focal region of the receiver.

Considering a segment of the mirror surface 1 ft along the axis by 7 ft across, as indicated in Fig. 4.1, an area of 7 sq ft, less the shadow area caused by the focal tube shadow, will be available to intercept the rays of the sun. The shadowed area cast by the insulated focal tube will be .75 sq ft. This will allow 6.25 x 275 = 1720 Btu per hr per ft of mirror length as the energy received on the mirror surface. Now, applying the mirror reflectivity, the amount of energy reflected is 1720 x 0.80 = 1377 Btu per hr per ft of length. This quantity, after being multiplied by the mirror effectiveness, is reduced to a value of 1377 x 0.884 = 1217 Btu per hr per ft of length. This is the average quantity of energy received at the focal region of the cylindrical parabolic mirror for the foot of length midway between the ends of the focal tube receiver.

4.4 <u>The Focal Tube Receiver</u>. A "heat trap" similar to that which is provided by a "hohlraum" is found by using the inside surface of a tube for interception of solar energy. A slotted opening on the mirror side of the tube allows radiation to pass inside.

4.4.1 Focal Tube Receiver Size. The problem of focal tube receiver size will now be considered, starting with the slotted opening on the mirror side of the solar energy receiver. This opening is to be covered with a pyrex shield.

The sun's rays, due to the size of the sun, will impinge upon the surface of the reflector with an angle of 32 minutes between the outside rays as depicted in Fig. 4.2A.[7a] This angle of incidence will be reflected to the focal tube receiver and the receiver opening must be at least large enough to receive these rays. Therefore, in Fig. 4.2B, consider a ray coming from the outside edge of the mirror to focal point F. Construct a perpendicular to this ray, FT. The sine of the angle θ that this ray makes with the perpendicular from the vertex of the parabola is

$$\sin \theta = \frac{42''}{44.7''} = 0.94 \text{ and } \theta = 70^\circ$$

Now

$$\tan 16' = \frac{TF}{44.7} = 0.005$$

$$TF = 44.7 \times 0.005 = 0.225 \text{ in.}$$

and

$$\cos 70^\circ = \frac{TF}{FS} = \frac{0.225}{FS} = 0.342$$

$$FS = 0.66 \text{ in.}$$

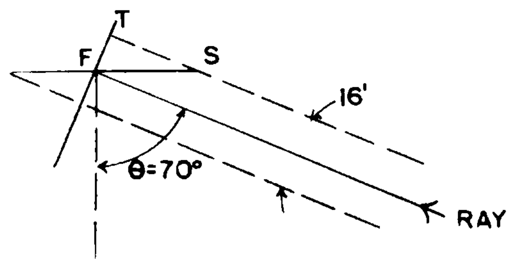
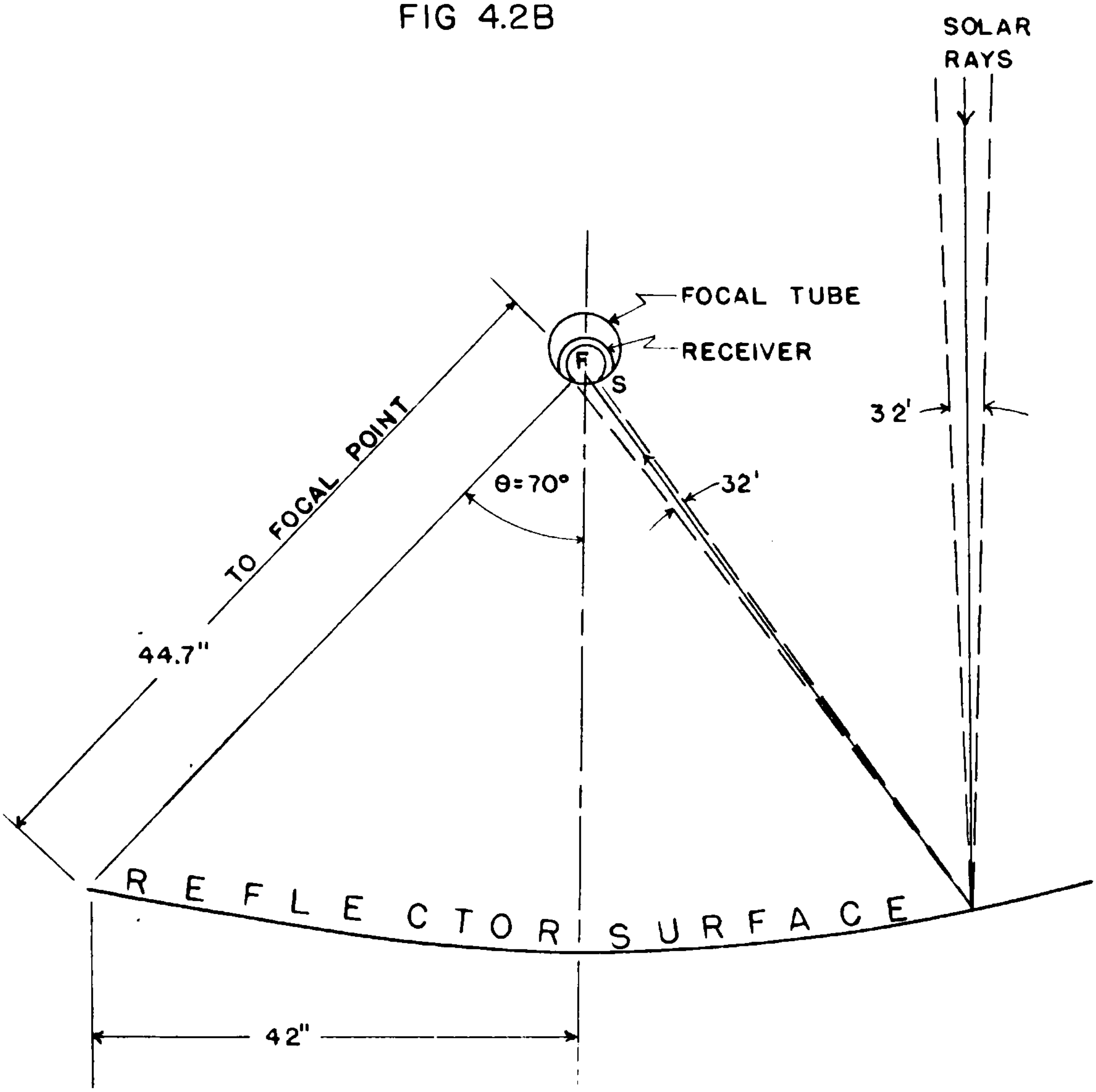


FIG 4.2B



FOCAL TUBE ORIENTATION

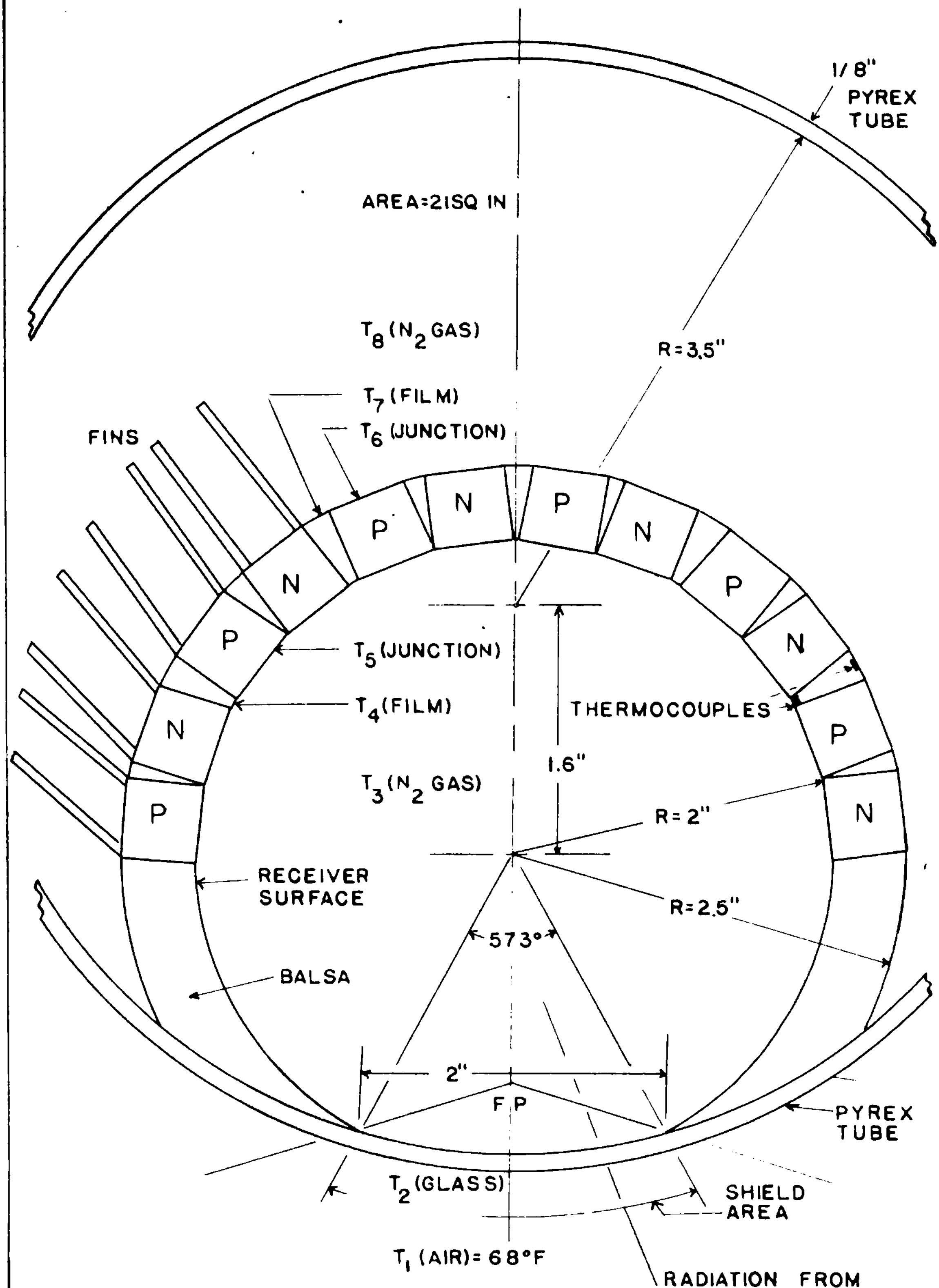
FIG 4.2A

Since 16 minutes accounted for only half of the opening, the entire opening must be 2 x 0.66 = 1.32 inches. This is the minimum opening for the focal tube receiver. The mirror effectiveness arrived at by Bowman (88.4%) was based on some reflector inaccuracies and to accomodate these inaccuracies an opening of 2.0 in. will be used. This will provide a focal tube slotted area of

$$\left(\frac{2.0}{12}\right) \text{ ft} \times 1 \text{ ft} = 0.1667 \text{ sq ft per ft of length}$$

The diameter of the receiver of the focal tube will be 4 inches. The pyrex tube containing the receiver, Fig. 4.3, will have an inside diameter of 7 inches. This size will provide sufficient area for the cooling fins which will be described later.

4.4.2 <u>Heat Loss Through Shield</u>. Now, referring to Fig. 4.3 and the stated assumptions at the beginning of the chapter, an ambient air temperature, T_1, of 68°F and an average wind velocity of 8 mph across the focal tube will be used. This is a safe-side assumption since, if there were no wind, there would be less heat lost through the pyrex shield and this would be advantageous. Also, it will be assumed that the temperature, T_3, of the gas in the receiver is 400°F since temperatures in this range have been attained in similar apparatus. The part of the pyrex tube which



CROSS-SECTION OF FOCAL TUBE

FIG 4.3

shields the slotted opening will be considered a flat plate with air blowing over one side of it. Then the magnitude of the surface coefficient, h_c, may be computed from[3a]

$$h_c = 0.055 \frac{k}{L} \left(\frac{LV\rho}{\mu}\right)^{0.75} \qquad (4.1)$$

For air at average film temperature of 200°F:

Thermal conductivity, k, = 0.0182 Btu per hr ft°F

Density, ρ, = 0.0601 lb_m per cu ft

Absolute viscosity, μ, = 0.052 lb_m per ft hr

Velocity, V, = 8 x 5280 = 42,250 fph

Length, L, = 0.1667 ft

Substituting these values in Eq. 4.1, the surface coefficient is calculated to be 5.1 Btu per hr sq ft °F.

Now, using[3c]

$$q_c = h_c A(t_2 - t_1) \qquad (4.2)$$

where q_c = rate of heat flow due to convection, Btu per hr

A = surface area, sq ft

$t_2 - t_1$ = difference in temperature between pyrex shield and ambient air

A temperature, t_2, of 341°F was obtained by a method of trial and error using Eqs. 4.1, 4.2, and 4.3. Details of sample calculations are presented in Appendix A. Thus q_c = 226.5 Btu per hr, which is the heat lost by convection from a one foot length of the shield to outside air.

Next, the equation for the net rate of heat transfer by radiation from the receiver surface inside the focal tube to the pyrex shield will be used:[17c]

$$q_r = \frac{\sigma \epsilon_2 \gamma (2\pi - \gamma)}{4\pi} \left[\frac{\epsilon_1}{\epsilon_2} \quad T_5^{\,4} - T_2^{\,4} \right] \qquad (4.3)$$

where σ = Stefan-Boltzmann constant

$\sigma = 0.173 \times 10^{-8}$ Btu per hr per sq ft per $^{\circ}F^4$

ϵ_1 = emissivity of the lampblack coating on the internal receiver surface = 0.97

ϵ_2 = emissivity of pyrex[3d] = 0.94

γ = arc in radians = 57.3° = 1.000r

$2\pi - \gamma = 302.7^{\circ}$ = 5.283r

T_5 = Temperature = $302^{\circ}F + 460 = 762^{\circ}R$

T_2 = Temperature = $341^{\circ}F + 460 = 801^{\circ}R$

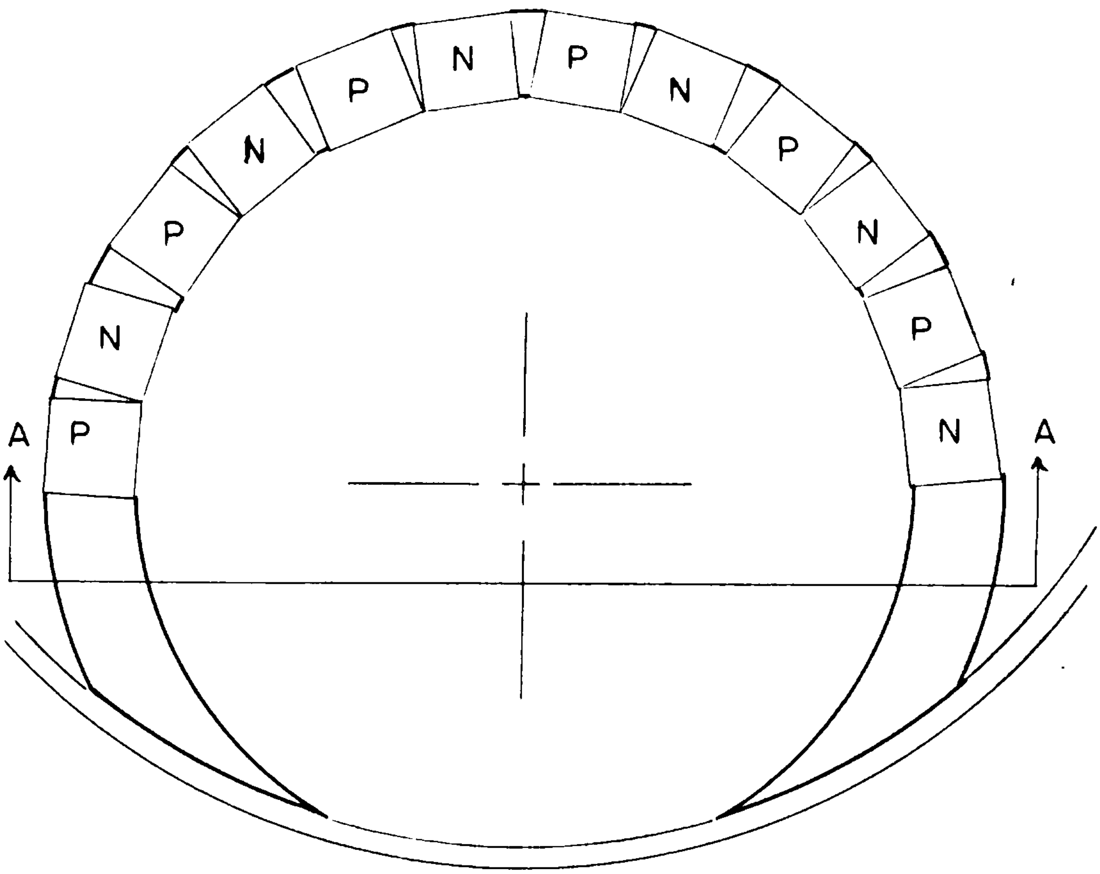
Substituting the values into Eq. 4.3, q_r is -35.4 Btu per hr due to radiation. The loss due to convection is q_c = 334 Btu per hr.

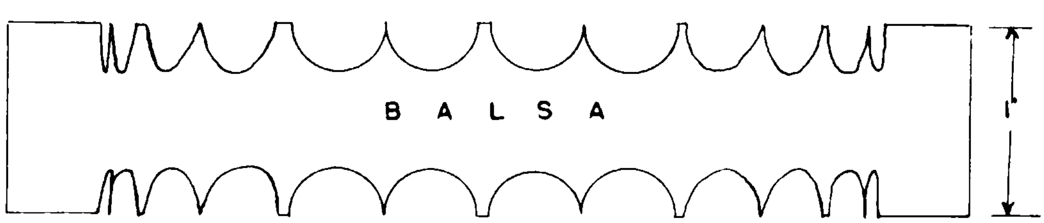
The solar radiation transmitted by a glass tube in a cylindrical parabolic reflector is 90%. Of this 10% loss to the glass, 8% is reflected and 2% is absorbed by the glass.[15] Therefore, of the 1217 Btu per hr calculated in paragraph 4.3, we have 0.02 x 1217 = 24.4 Btu per hr being absorbed by the pyrex glass and 97.4 Btu per hr being reflected. Thus the total heat absorbed by the glass is 334 + 24.4 - 35.4 = 323 Btu per hr. This compares favorably with the amount of heat lost from the shield to the air by convection and radiation (226.5 + 97.4 = 323.9 Btu per hr). Therefore it may be assumed that the temperature of the shield as determined by trial and error is correct at 341°F and that a heat loss to the atmosphere of 323 Btu per hr will be incurred. Detailed calculations are presented in Appendix A.

Thus, of the 1217 Btu per hr from the mirror surface, 323 Btu per hr are lost from the shield through radiation and convection to the air, leaving 894 Btu per hr for the focal tube receiving surface area.

4.4.3 Thermoelectric Element Support. Balsa wood will be used as a support to hold the thermoelectric modules in place. Basically, the configuration of the modules will be as shown in Fig. 3.1 with L = 0.5 inches and D = 0.5 inches. The balsa wood support will be built up of sections as shown in Fig. 4.4 with six modules held in place between each two sections.

The area of the balsa wood that conducts heat will be the total receiver surface area minus the total cross sectional area of the modules which are supported by the balsa. The total receiver surface is 0.627 sq ft per ft of length. Each thermoelectric module has a cross sectional





BALSA WOOD MODULE SUPPORT

FIG 4.4

area of 0.00272 sq ft which, when multiplied by 72, the number of modules required per foot of length, gives a total area of 0.196 sq ft. The required number of modules mentioned will be justified later. Therefore, the surface area of balsa wood through which heat would be conducted is 0.627 - 0.196 = 0.431 sq ft.

The surface temperature on the hot side of the balsa, T_5, will be maintained at 302°F. The surface temperature at T_6 will be maintained at 77°F which is the cold junction temperature for which module output data is given by the manufacturer.

The heat transmitted through the balsa wood is given by the equation[3c]

$$q = -kA \frac{\Delta t}{\Delta L} \text{ Btu per hr} \tag{4.4}$$

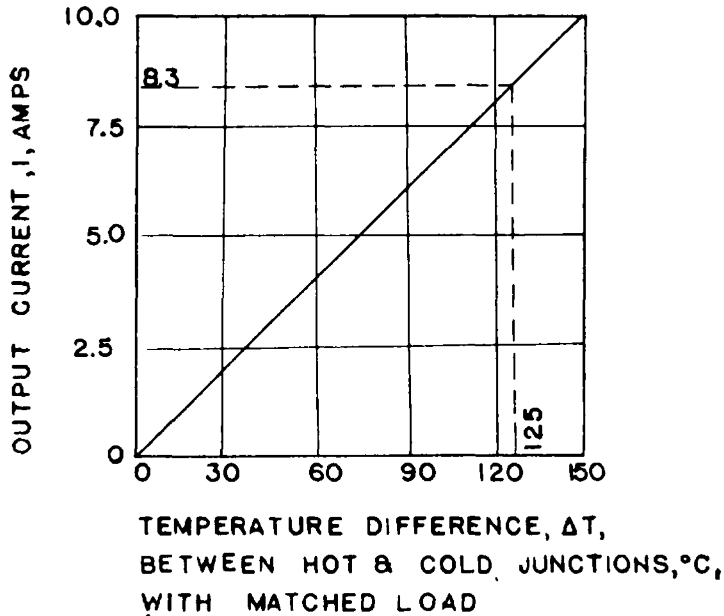
where
- q = rate of heat transmission, Btu per hr
- k = thermal conductivity of the balsa = 0.034 Btu per hr sq ft °F per ft
- A = area = 0.431 sq ft
- t = temperature difference, (302 - 77) = 225°F
- L = material thickness = 0.0417 ft

For the conditions stated, the rate of heat transmission was calculated to be 79 Btu per hr.

The quantity of heat now left for use by the thermoelectric modules is the total available within the receiver, 894 Btu per hr, less the loss by conduction through the balsa wood, 79 Btu per hr. This is 815 Btu per hr. Sample calculations are given in Appendix B.

4.4.4 Thermoelectric Modules. As discussed earlier, the kind of thermoelectric element which will be used is bismuth telluride, Bi_2Te_3. In selection of actual hardware there are several companies that produce similar products. The TA-11 Thermoelectric Junction made by Tecumseh Products Company was selected because more pertinent information was available for this element than for others. This information is presented in Fig. 4.5 which is a reproduced graph of Thermoelectric Generator Output Current vs. Temperature Difference Between Hot and Cold Junctions. A current flow of 8.3 amperes may be produced by a temperature difference of $302^\circ F$ minus $77^\circ F = 225^\circ F$ ($125^\circ C$), as is indicated by the dashed line. This is the operating temperature for which the converter was designed. The figure of merit for this product is 2.5×10^{-3} per $^\circ K$ and the efficiency for a temperature difference of $125^\circ C$ is 5.24%. Sample calculations are given in Appendix C.

Using E(volts) = I(amps) R(ohms) the emf for each module is 0.02075 volts. Since each module uses 11.2 Btu per hr and a total of 815 Btu per hr is available for use.



COLD JUNCTION AT 25°C

RESISTANCE = 0.0025 OHMS

MAX HOT JUNCTION TEMP = 150°C

THERMOELECTRIC GENERATOR OUTPUT

FIG 4.5

72 modules per foot of length of focal tube receiver are needed. Appendix D gives sample calculations for this quantity. These modules are connected in series and therefore, the total emf for 72 modules is 72 x 0.02075 = 1.495 volts. The power output, P = EI, is 8.3 x 1.495 = 12.4 watts, which is equivalent to 42.3 Btu per hr or 0.0166 horsepower.

4.4.5 <u>Receiver Temperature Control</u>. Computations showed that the nitrogen used in the receiver of the focal tube must be kept in motion so that the magnitude of the surface film coefficient may be controlled within a reasonable value. If the gas did not move, the film layer would be too thick and the temperature gradient needed would be greater than that which could be attained during operation of the converter. For this reason, a temperature control system must be installed to maintain a constant module hot junction temperature.

A thermocouple will be installed to indicate the temperature of the hot junction of the thermoelectric elements. The operating temperature is 302°F. An iron-constantan thermocouple at 302°F produces about 8 millivolts. A small amplifier, therefore, will be used and the amplified current will operate a servomotor. This motor in turn controls a slide-wire resistance which regulates the variable-speed fan motor. Thus, if the temperature drops, the film in the focal tube receiver is too thick and the fan must speed up to increase the fluid velocity, which in turn reduces the film and the temperature rises. If the temperature is too high the reverse process takes place. This entire system is thoroughly insulated to prevent the loss of heat due to this circulation system. Appendix E shows some sample calculations.

The calculations indicate a high velocity but this is the starting condition. As the gas warms up due to passing through the focal tube receiver and the energy imparted to it from the fan, it is expected that this velocity will increase. However, as the velocity increases the film decreases and the junction temperature rises. This in turn will slow down the fan speed and the fluid velocity.

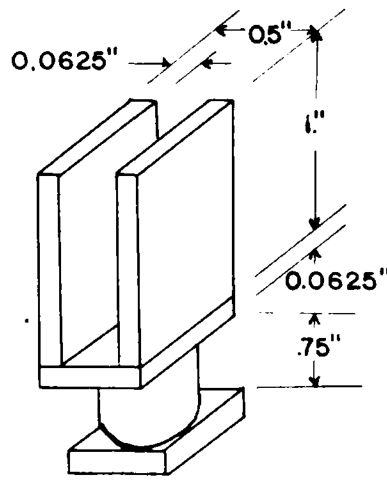
4.5 Cold Junction Temperature Control. The heat used to convert energy from each of the modules is 11.2 Btu per hr per module. This multiplied by an efficiency of 0.0524 produces 0.589 Btu per hr. Thus 11.2 - 0.589 = 10.631 Btu per hr or a total of 72 x 10.631 = 766 Btu per hr must be removed from the cold side of the module surfaces. To this must be added the amount conducted by the balsa of 79 Btu per hr or a total of 766 + 79 = 845 Btu per hr must be dissipated from the cold junction side of the focal tube receiver.

Some of this heat could be lost due to radiation through the pyrex cover. However, the tube enclosing the receiver will be insulated around the sky side by 85% magnesia. This is necessary since the temperature of the gas in the tube should be maintained at $70^{\circ}F$. Without the insulation there would be a high rate of heat conduction from the atmosphere to this gas to contend with in addition to large solar radiation effects. With no insulation and a temperature difference of $2^{\circ}F$ the heat transfer would be approximately 9 Btu per hr.

Calculations showed that fins would be necessary on the cold junction ends of the modules to dissipate the heat at a satisfactory rate. This fact affected the size of the covering tube and the minimum size of this tube that will provide room enough for the fins is a tube of inside diameter equal to 7 in. Even with a tube this large, fins on the bottom row of modules will have to be bent slightly. (See Fig. 4.3.)

Considering the perimeter on the outside of the balsa support, which is the cold junction side of the receiver, the arc length is 0.697 ft. The equivalent diameter, D_e, is 0.838 ft.

The fins to be used will be as indicated in Fig. 4.6 and their effectiveness is 0.8. The velocity of the nitrogen required to carry the heat away from the fins was calculated



HEAT DISSIPATION FINS

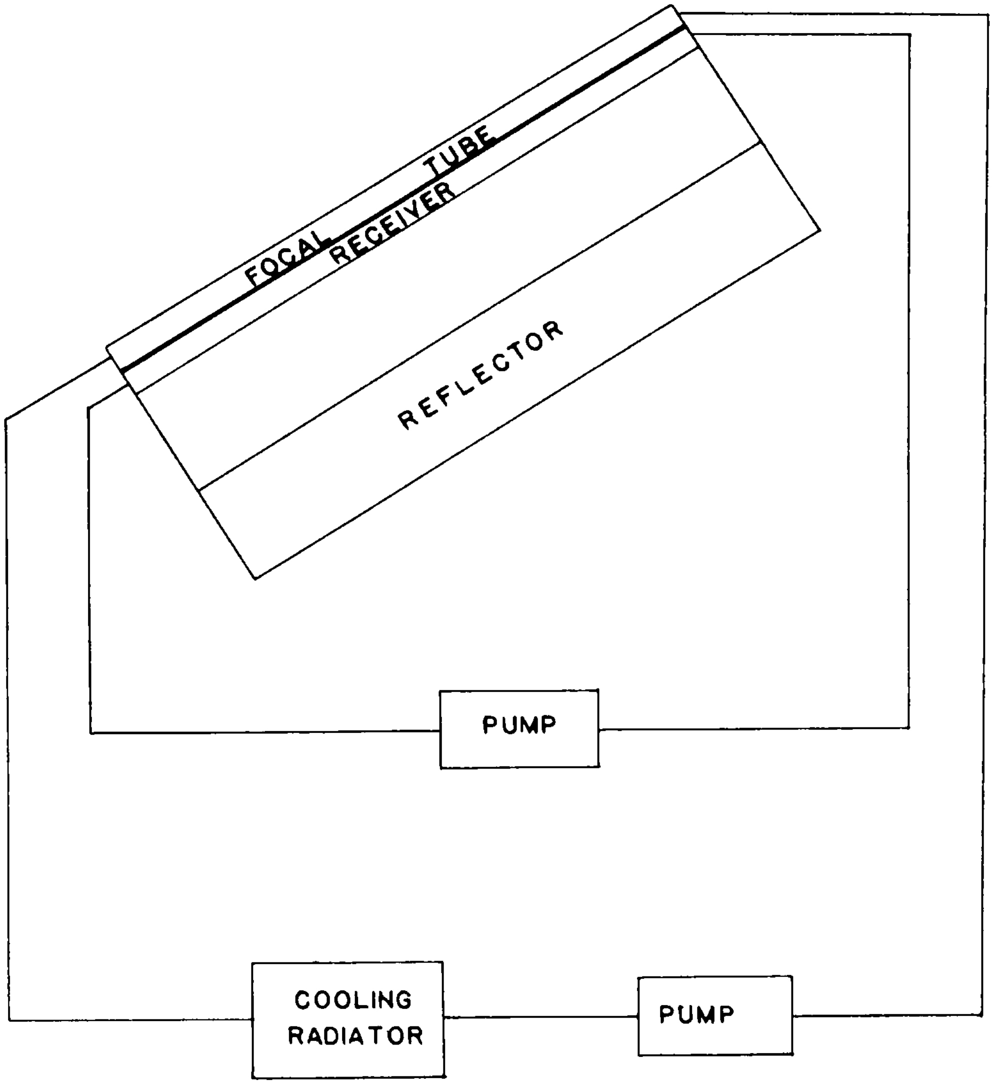
FIG 4.6

to be 387 fps with the intention of maintaining a gas temperature of 70°F. Although this velocity will provide turbulent flow it may be found necessary at some later time to add vanes to the tube to insure turbulent flow between the dissipating fins. Appendix F shows calculations.

This velocity is controlled by a thermocouple installed at the junction level of the thermoelements. This junction is maintained at 77°F and any changes are relayed to adjust the pump moving the gas through the circuit, as described in paragraph 4.4.5.

4.6 <u>The Cooling System</u>. The heated nitrogen gas should exit from the focal tube at a temperature of 72°F and a velocity of 387 fps. The gain of thermal energy stored within this gas is the 845 Btu per hr which was dissipated from the cold junction side of the receiver plus gas frictional energy. The tube used to conduct this gas is a 5 1/4 in. i.d. copper tube. The gas is cooled to 68°F and the heat removed by passing the gas through a finned radiator water cooler, Fig. 4.7. Sample calculations are given in Appendix G.

4.7 <u>The Refrigeration Unit</u>. To provide a means of more accurately measuring the output of the solar converter a small refrigerator will be operated to gather research readings. The unit to be used will be a one cubic foot



COOLING SYSTEM

FIG 4.7

insulated box.

The output of the generating modules is 8.3 amps at 0.02075 volts per module or 12.38 watts for 72 modules. The hot junction temperature will be 77°F or 25°C. This will be maintained by a fan moving air at ambient temperature of 68°F across fins on the outside of the refrigerator. With this arrangement a temperature difference of 12°C can be maintained, and a heat load of 1.3 watts can be pumped. Appendix H. gives sample calculations.

4.8 Total Collector. The total length of the solar collector built at the University of Arizona is 7 ft. However, part of this seven foot length may be shaded. Only on two days of the year, September 21 and March 21, will the sun's rays be normal to the axis of the mirror. On all other days the end plates of the reflector will either cause a shadowed area or reinforce the impinging rays. Bowman's thesis[2] gives a detailed discussion of this problem. To ensure that the maximum use is made of the entire seven feet, compensator mirrors may be used if necessary around the edge of the main reflector.

CHAPTER 5

COSTS, FUTURE USES, AND CONCLUSIONS

5.1 Costs of Materials. The major cost of constructing the proposed device is the purchase of thermoelectric material. The price of the generating elements for example is $99.00 each if bought in quantities between 10 and 49. The cost for one foot of the focal tube receiver then becomes $7,130.00.

This cost seems exorbitant but present production methods are still rather limited. As better methods of manufacture are found and more thermoelectric material is used the price is expected to be reduced. In fact, the prices above were for 1960 and a circular from MELCOR, Trenton, New Jersey, quoted prices in March 1961 at about half of those quoted above. It is expected, that in the not too distant future, thermoelectric elements will be available for a fraction of a dollar each.

The prices of the other construction materials are insignificant compared to the price of the thermoelectric elements. They would consist of:

2 variable speed motors[14]	$ 900.00
2 impellers[11]	$1000.00

1 cooling radiator[11]	\$ 200.00
40 ft of brass tubing[6]	\$ 240.00
2 thermocouples[14]	\$ 36.00
1 solar collector, constructed[2]	\$ 753.00
Total	\$3129.00

5.2 Storage Considerations. No special problems were found concerning storage in batteries since the current produced and used by thermoelements is direct current. The quantity of energy produced is found to be so small, however, that storage is not considered feasible at this time. If at some later date storage seems necessary, a regular wet-cell battery could be used. If storage on space craft is needed a dry-cell type of battery should be used.

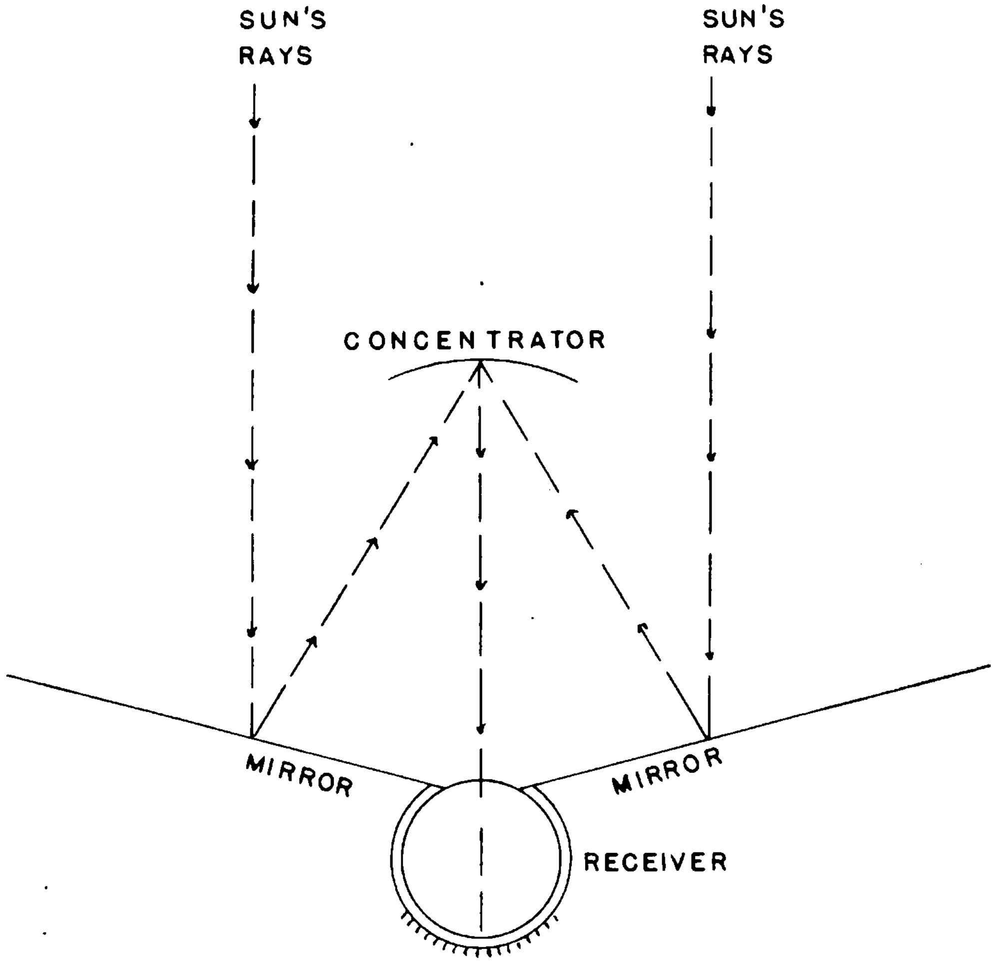
5.3 Future Uses. The solar energy converter proposed here is primarily complicated by the cooling system. In a location where the ambient air temperature were much cooler than in Tucson this could be improved. In a cool area, for instance, the pyrex cover and insulation on the cold junction side of the focal tube receiver could be dispensed with leaving only an aluminum shade. Cooling solely be natural convection and radiation could then be relied upon. This would not be as controlled as the proposed device but should be highly satisfactory.

For use in space the reflector surface might be a double reflector as shown in Fig. 5.1. This would make it possible to leave the fins on the cold junction side of the focal tube receiver uncovered. The excess heat could then be radiated into space.

5.4 Conclusions. The entire seven foot length of the collector will produce only about one tenth of a kilowatt. This is very low considering the initial cost. It is felt, however, that with a considerable reduction in price of thermoelectric material and improvement in quality that this solar energy converter may well prove feasible. This would be true only if, as pointed out in paragraph 5.3, the cooling apparatus could be dispensed with since the cooling motors will use much more power than is generated.

One great advantage that this device has over other solar collectors is that, except for the cooling mechanism and orientation, the energy converters themselves have no moving parts and therefore are free of any maintenance problems, once installed.

In final conclusion, however, it must be admitted that the device at this time is not practical to construct due to cost considerations.



SPACE SOLAR COLLECTOR

FIG 5.1

APPENDIX A

Sample calculations for heat loss through pyrex shield:

The heat lost to the outside air by convection, q_{co}, and radiation, q_{ro}, must equal the heat to the glass from inside the focal tube receiver by convection, q_{ci}, and radiation, q_{ri}.

Eq. 4.1, surface coefficient[3] to ambient air:

$$h_{co} = 0.055 \frac{k}{L} \left(\frac{LV\rho}{\mu}\right)^{0.75}$$

$$h_{co} = 0.055 \frac{0.0182}{0.1667} \left[\frac{(0.1667)(42,250)(0.0601)}{0.052}\right]^{0.75}$$

$$h_{co} = 5.1 \text{ Btu per hr sq ft } {}^{\circ}F$$

Eq. 4.2, rate of heat flow[3] to outside air:

$$q_{co}(\text{convection}) = h_c A\ (t_2 - t_1)$$

$$q_{co} = (5.1)(0.1667)(341-68)$$

$$q_{co} = 226.5 \text{ Btu per hr}$$

$$q_{ro}(\text{radiation}) = 0.08 \times 1217 = 97.4 \text{ (Ref. 14)}$$

$q_{total} = 226.5 + 97.4 = 323.9$ Btu per hr, heat lost to air

Eq. 4.3, radiation inside focal tube receiver:[17c]

$$q_{ri} = \frac{\sigma \epsilon_2 \gamma (2\pi - \gamma)}{4\pi} \left[\frac{\epsilon_1}{\epsilon_2} T_5^4 - T_2^4 \right]$$

$$q_{ri} = \frac{(0.173 \times 10^{-8})(0.94)(1)(5.2832)}{12.5664} \left[\frac{0.97}{0.94}(762)^4 - (801)^4 \right]$$

$$q_{ri} = (0.0685 \times 10^{-8}) \left[(1.043)(3370 \times 10^8) - (4100 \times 10^8) \right]$$

$$q_{ri} = (0.0685)(3570-4100) = (0.0685)(-530) = -35.4$$

$q_{ri} = -35.4$ Btu per hr heat radiated from pyrex shield. To this figure must be added 24.4 Btu per hr absorbed by the glass as the heat enters. Heat loss by convection[3] from inside focal tube:

$$q_{ci} \doteq A h_c \Delta t$$

$h_{ci} = 33$ Btu per hr sq ft °F, Appendix E.

$A = 0.1667$ sq ft

$T_2 = 341°F = 801°R$

$T_3 = 400°F = 860°R$

$q_{ci} = (0.1667)(33)(860-801) = 334$ Btu per hr

This gives a total of 334 + 24.4 - 35.4 = 323 Btu per hr which is the same as the heat lost to the air. The temperature of 341°F is, therefore, correct.

APPENDIX B

<u>Sample calculations for the heat loss due to conduction through the balsa wood:</u>

$C = R\alpha =$ arc length

inside radius $R_i = 2$ in. $= 0.1655$ ft

$\alpha_i = 192^{\circ} = 3.351$ radians

arc length $C_i = 0.558$ ft

outside radius $R_o = 2.5$ in. $= 0.204$ ft

$\alpha_o = 3.351$ radians

arc length $C_o = 0.697$ ft

average of C_o and $C_i = 0.627$ ft. For a length of one foot the area is 0.627 sq ft.

One mudule cross section area is $2\frac{(0.25)^2}{2 \cdot 144} = 0.00272$ sq ft

or a total $= 72 \times 0.00272 = 0.196$ sq ft

Balsa area left:

$0.627 - 0.196 = 0.431$ sq ft

Thermal conductivity for balsa, $k = 0.034$

Thickness of balsa is $x = 0.5$ in. $= 0.0417$ ft

$$q = \frac{kA\Delta t}{x}$$

$$q(\text{balsa}) = \frac{(0.034)(0.431)(225)}{0.0417} = 79 \text{ Btu per hr}$$

Total q = 894 Btu per hr in the focal tube receiver. Subtracting 79 Btu per hr leaves 815 Btu per hr for use of modules.

APPENDIX C

<u>Sample calculations for thermoelectric module efficiency</u>:

Eq. 3.3 is the basic equation for generating efficiency for heat input and electrical output:

$$\eta_{max} = \frac{(T_h - T_c)}{T_h} \left[\frac{\sqrt{1+Z(T_h+T_c)/2}-1}{\sqrt{1+Z(T_h+T_c)/2}+T_c/T_h} \right]$$

Hot junction temperature

$$T_h = 302^\circ F = 150^\circ C + 273 = 423^\circ K$$

Cold junction temperature

$$T_c = 77^\circ F = 25^\circ C + 273 = 298^\circ K$$

$$T = T_h - T_c = 125^\circ C$$

$Z = 2.5 \times 10^{-3} deg^{-1}$ (Tecumseh Products Co.)

$$\eta = \frac{(423-298)}{423} \left[\frac{\sqrt{1+2.5x10^{-3}(423+298)/2}-1}{\sqrt{1+2.5x10^{-3}(423+298)/2}+298/423} \right]$$

$$\eta = .0524$$

APPENDIX D

<u>Sample calculations for number of modules required</u>:

From the chart, Fig. 4.5, assuming a temperature difference of 125°C:

$I = 8.3$ amps

$R = 0.0025$ ohms

$E = IR = 0.02075$ volts

Power $= EI = (8.3)(0.02075) = 0.172$ watts

Watts x 0.05688 = Btu per min

$$q = \frac{(0.172)(0.05688)(60)}{(0.0524} = 11.2 \text{ Btu per hr}$$

By dividing 815 Btu per hr (Par. 4.4.4), the heat available for use through the modules, by 11.2 Btu per hr a total of 72.6 modules is needed for conversion of the available heat. The next lowest whole number of 72 modules will be used.

APPENDIX E

Sample calculations for velocity of fluid (N_2) in focal tube receiver:

The heat from the receiver will be conducted successively through the film layer, then lampblack, then the 1/16 inch copper plate at the hot junction of the module, then to the cold junction of the module. The hot junction temperature will be $T_h = 302^\circ F = 150^\circ C$. This is the maximum temperature recommended by the manufacturer for T_h. The gas temperature, T_3, has been assumed at $400^\circ F$ thus ΔT is $98^\circ F$. Since the conductivity of the copper (224) is so high and the thickness of the lampblack is so thin the temperature change across these is very minor (less than $1^\circ F$) and will not be considered.

One module is considered in these computations. Each modules must receive 11.2 Btu per hr. The copper plate on the hot junction base of the module is a square plate 0.5 in. x 0.5 in. x 2 = 0.5 sq in. which is 0.003470 sq ft.

Using $q = Ah\ t$, the convection coefficient,[3]

$$h_{ci} = \frac{q}{A(T_3 - T_5)}$$

$$11.2 = (0.00347)(400-302)h$$

h_c = 33.1 Btu per hr sq ft ^oF

Now use $h_c = 0.023 \frac{k}{D} \left(\frac{DV\rho}{\mu}\right)^{0.8} \left(\frac{C_p \mu}{k}\right)^{0.4}$ to calculate velocity.[3]

Values for N_2 at an average film temperature 10a of 350^{o}F:

ρ = 0.0453 lb_m per cu ft

c_p = 0.2509 Btu per lb_m ^oF

μ = 0.0442 lb_m per ft hr

k = 0.0202 Btu per hr ft ^oF

D, the diameter = 4 in. = 0.333 ft

Correction factor for pipe length = $\frac{7}{0.333}$ =

21 diameters = 1.14 (Ref. 3f)

Substituting these values in the equation for h_c and solving for velocity:

$$(1.14)(33.1) = (0.023)\left(\frac{0.0202}{0.333}\right)\left[\frac{(0.333)(0.0453)V}{(0.0442)}\right]^{0.8}$$

$$\left[\frac{(0.2509)(0.0442)}{(0.0202)}\right]^{0.4}$$

V = 378 fps

APPENDIX F

Sample calculations for cooling fluid (N_2):

Fin effectiveness:

k = A dimensionless Parameter, n, $= L\sqrt{\frac{hL}{ka}}$

$L = 0.5$ in.

$a = 0.375$ sq in.

$h = 40$ assumed

$k = 224$ (copper)

$n = 0.5 \sqrt{\frac{(40)(0.5)}{(224)(0.375)}} = 0.775$

$\eta = 0.8$ = fin effectiveness, from an empirical graph, pg. 168, Ref. 8.

Velocity of fluid:

Referring to Fig. 4.6 the fin area is 1/16 in. = 0.0625 in.

0.0625 in. x 2 fins = 0.125 in.

0.125 x 0.75 = 0.0937 sq in.

Area of base = 0.75 x 0.5 = 0.375 minus 0.094 which leaves an area of 0.281 sq in.

Surface area = 0.75 x 1.0 x 2x2 = 3 sq in. Therefore the total area = 3.281 sq in. and with effectiveness = 0.8 the effective area is 2.6048 sq in. and the total area = $\frac{2.60}{144} \times 2 \times 72 = 2.6$ sq ft.

Now since the heat conducted by the balsa is small compared to the heat conducted through the modules the total will be considered through the modules. This is a safe-side assumption.

$q = 845$ Btu per hr

$A = 2.6$ sq ft

$\Delta t = 77^oF - 70^oF = 7^oF$

Substituting in $q = Ah\ t$

$h = \frac{845}{(2.6)(7)} = 46.4$

Now for N_2 at 70^oF:

$\rho = 0.0730\ lb_m$ per cu ft

$c_p = 0.2482$ Btu per $lb_m\ ^oF$

$\mu = 0.0426\ lb_m$ per ft hr

$k = 0.0149$ Btu per hr ft ^oF

Correction factor for length of tube = $\frac{7}{0.517}$ = 13.5 dia.

Therefore, a final correction factor of 1.22 is used. (Ref. 3f)

Equivalent diameter:

$D_e = 4\ \frac{A}{P}$

A = area = 21 sq in. = 0.146 sq ft

P = perimeter, use perimeter of heat transfer only

$P = C_i = R_i \quad _i = 0.697$ ft

$D_e = 4 \times 0.146 = 0.838$

$$h_c = 0.023 \frac{k}{D_e} \left(\frac{D_e V \rho}{\mu}\right)^{0.8} \left(\frac{c_p \mu}{k}\right)^{0.4}$$

$$46.4 = 0.023 \frac{0.0149}{0.838} \left[\frac{(0.838)(0.073)V}{0.0426}\right]^{0.8} \left[\frac{(0.2482)(0.0426)}{0.0149}\right]^{0.4} (1.22)$$

$V = 13.9 \times 10^5$ ft per hr = 387 fps

$M = \frac{387}{1135} = 0.341$ Mach number.

APPENDIX G

<u>Sample calculations for the cooling radiator using water in finned tubes</u>:

Temperature of N_2 into radiator = 72°F

Temperature of N_2 leaving radiator = 68°F

Velocity in tube = 23,200 fpm

Area of tube = 0.151 sq ft

Velocity through radiator $\approx$ 450 fpm

$$A_1V_1 = A_2V_2$$

$$(0.151)(23{,}200) = A_2(450)$$

$$A_2 = 7.8 \text{ sq ft}$$

A commercial radiator with an internal flow area of 7.8 sq ft will be used.

By controlling the water input through the coils the temperature of the leaving nitrogen gas can be controlled. This will, in effect, be a manual control on the entire cooling system, since the more heat removed by this means, the slower the gas must be pumped through the cooling system to maintain the desired cold junction temperature for the generating modules.

APPENDIX H

<u>Sample calculations for refrigeration</u>:

The electrical output for one foot of length of the focal tube with 72 modules mounted in the focal tube receiver is:

$I = 8.3$ amps

$E = 0.02075$ volts per module

$P = 0.172$ watts per module, or 12.38 watts output for 72 modules

$R = 0.0025$ ohms

Ambient air temperature = 68°F = 20°C

If the temperature of the hot junction of the module is kept at 25°C (77°F) by moving air at 68°F over large fins on the outside of the regrigerator, and the power input is 2 watts, a heat load of 1.3 watts can be transferred at a temperature difference of 12°C. Thus, according to charts of the Tecumseh Products Company, 6 modules could be used in the refrigerator. This is without considering the resistance of the lead wires and connections between the generating and cooling modules. This circuit would have to be balanced in final construction to account for these resistances.

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