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EVACUATED TUBULAR COLLECTOR UTILIZING A HEAT PIPE

Progress Report, May 1, 1975—August 31, 1975

U. Ortabasi
F. P. Fehlner

1975

Work Performed Under Contract No. EY-76-C-02-2608

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Corning Glass Works

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AN EVACUATED TUBULAR
COLLECTOR UTILIZING
A HEAT PIPE

Progress Report for
the Period May 1, 1975 to August 31, 1975
ERDA Contract E(11-1)-2608

Prepared By
Corning Glass Works
Corning, New York

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An Evacuated Tubular Collector
Utilizing a Heat Pipe

U. Ortabasi
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Corning Glass Works
Research and Development Laboratory
Corning, New York

The initial work necessary to design, build, test, and cost out an evacuated tubular collector utilizing a heat pipe has been completed during the period May 1 - August 31, 1975.

The areas of work covered include: optical and thermal analysis of a cusp collector, heat pipe design, cusp mirror fabrication, selective absorbers, collector assembly, and test facilities.

FOREWORD

This report was prepared under Contract No. E(11-1)-2608 from the Energy Research and Development Administration, Solar Energy Branch with Dr. Stephen L. Sargent as Program Manager. The period May 1, 1975 through August 31, 1975 is covered.

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PROGRESS REPORT

NO. 1

AN EVACUATED TUBULAR
COLLECTOR UTILIZING
A HEAT PIPE

CORNING GLASS WORKS
CORNING, NEW YORK

August 31, 1975

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ABSTRACT

Research and development activities performed by the Corning Glass Works solar group during the period from May 1, 1975 to August 31, 1975 are reported. The analytical studies encompassed optical modeling of a modified cusp reflector, fresnel losses from tubular enclosures as a function of orientation, preliminary work on a digital Monte-Carlo Ray Tracing Computer Code and the determination of U_L losses as a function of vacuum level and temperature of the absorber.

Compatible enclosure materials, wicks and working fluids were selected to assemble heat pipes for use as solar collector absorbers.

Cusp reflectors with good accuracy were fabricated from various commercial bright aluminum sheet and their optical properties determined. Evacuation techniques were developed and special bake-out procedures were worked out to assemble the evacuated tubular collectors. The work on selective coatings narrowed down the field of potential absorber films to "black chrome" which has good stability in vacuum and the necessary α/ϵ value.

Six different heat pipes and one flow-through absorber were fabricated and were partially characterized for thermal performance.

Indoor and outdoor test facilities were completed and calibrated within proposed NBS standards. They are now available

for parametric as well as true-life experiments with solar radiation.

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SECTION I

STATEMENT OF OBJECTIVE

The objective of this project is to conduct an integrated program of research and development to make available an evacuated tubular solar collector having significantly higher system performance characteristics than the existing flat-plate types.

The principal features of the collector include a strong, clear glass tube which vacuum encapsulates an internal cusp reflector having a tubular absorber in its focal area. High performance is achieved by the use of the following advanced features: (1) vacuum insulation, (2) selective absorber coatings, (3) antireflecting coating films, (4) highly efficient removal of the absorbed heat from the solar collector.

Both heat-pipe (Fig. I-1a) and flow-through (Fig. I-1b) concepts as absorber tubes are being examined and ultimately will be compared with the Corning Glass Works evacuated tubular flat plate collector¹ (Fig. I-1c) on a cost-performance basis.

The cusp collector design is based on the stationary (non-tracking) non-focusing, low concentration mode of operation and is intended for use in total solar climate control (SCC) systems which include solar cooling and heating, as well as hot water needs for buildings.

FIGURE I-1

SCHEMATIC OF EVACUATED TUBULAR SOLAR COLLECTOR CONCEPTS
UNDER EVALUATION.

ADVANCED HEAT-PIPE
COLLECTOR

TUBULAR ABSORBER
CUSP REFLECTOR HEAT
REMOVAL BY HEAT-PIPE
ACTION

ADVANCED FLOW -
THROUGH COLLECTOR

TUBULAR ABSORBER
CUSP REFLECTOR HEAT
REMOVAL BY TRANSFER
FLUID

CORNING BASELINE
COLLECTOR

FLAT-PLATE ABSORBER
HEAT REMOVAL BY
TRANSFER FLUID

Basic Features of the Cusp Collector

The stationary tubular cusp collector, as proposed in this work, is displayed in Fig. I-2. The active collector is 48" long and 4" in diameter. The most significant features of this collector are the cusp reflector, vacuum insulation, and ultimately, a cylindrical heat pipe to absorb and convert the sun's energy to heat. The cusp reflector concept, originally introduced by Felix Trombe, has been the recent subject of further investigation by others.^{2,3,4} Calculations recently completed at Corning Glass Works show that a modified cusp reflector can be incorporated into a tubular glass envelope successfully, without losing the "ideal" light collection properties of the cusp. A schematic of such a combination is shown in Fig. I-3.

The geometry of the reflector surface has been designed to force all incoming rays within a large acceptance angle to be "funneled" onto the surface of the absorber tube. Since all the light intercepted is also trapped, the cusp collector is highly sensitive to diffuse radiation. For a concentration ratio close to one, as chosen in this work, a cusp collector in the stationary mode can collect nearly as much total radiation as a flat-plate collector. However, if desired, a cusp collector can be designed to yield concentration ratios larger than unity for increased radiation intensity. This, however, imposes smaller acceptance angles which ultimately may require a tracking collector.

The reflective losses at the reflector surface can be made relatively small by the use of a high quality surface. The high vacuum (10^{-4} - 10^{-5} Torr) inside the tubular envelope insures a

FIGURE I - 2
SCHEMATIC OF THE TUBULAR CUSP COLLECTOR UTILIZING A HEAT PIPE
AND ILLUSTRATION OF THE "PLUG IN" CONCEPT.

FIGURE I-3

CROSS SECTION OF THE TUBULAR CUSP COLLECTOR WITH
HEAT PIPE AS ABSORBER.

long lifetime for such a surface.

The characteristic of the cusp collector, as opposed to the flat plate, is the direct radiative transfer of the intercepted solar energy to the selectively coated absorber, rather than by heat conduction through fins. This radiant energy, absorbed as heat, is in turn transferred through the absorber wall directly either to a heat-transfer fluid in the flow-through version of the collector, or to the wick fluid in the case of a heat-pipe absorber. Hence by the principles of the design, the losses associated with the "fin efficiency" of flat plate absorbers are minimized.

Heat Pipe

The basic heat pipe is a cylindrical, evacuated enclosure which contains a wick structure and a small amount of vaporizable fluid. The device operates on a boiling-condensing cycle. The principle of a heat pipe operating as a solar absorber is shown schematically in Fig. I-4. Solar radiation absorbed at the evaporator end of the heat pipe is transferred to the wick causing the working fluid to evaporate. The pressure gradient created drives the vapor to the cold end where it condenses, thereby transferring the latent heat of the working fluid from the evaporator region to the condenser region. The liquid is then transferred from the cold end back to the hot end by capillary forces in the wick aided sometimes by gravity. By proper design, the temperature drop between the evaporator and the condenser in the heat pipe can be made smaller than the gradients which can develop in heat transfer fluids flowing in tubes, as practiced in conventional collectors. Thus, additional collector efficiency may be achieved by using a heat pipe in a SCC system.

PRINCIPLES OF THE HEAT PIPE AS SOLAR COLLECTOR ABSORBER
FIGURE I - 4

The use of heat pipes in solar collectors will result in a new approach to SCC system design. Instead of pumping a fluid through a conventional solar collector to recover heat, it will only be necessary to pump fluid past the collector containing a heat pipe. In a special design, the pump can be eliminated entirely. This optimum approach is compared in Fig. I-5a with the flow-through approach in Fig. I-5b. The heat pipe as used here has distinct advantages - there is no manifold needed for a heat transfer fluid and no auxiliary power is needed for pumping fluid through a heat transfer loop within the solar collector envelope.

Even when the heat pipe is used with a heat transfer manifold, the decrease in pumping cost could be significant, especially when a high flow rate is desired, as when air is the heat transfer medium.

The heat pipe, in essence, makes the solar collector an independent subsystem of the total SCC system. Instead of being tied directly into a flow-through fluid loop, each heat pipe is isolated. Thus, modular construction and easy replacement become possible.

Vacuum

Heat losses from the absorber plate of a solar collector can be significantly reduced if convective and conductive heat transfer are eliminated. Although several other mechanisms have been tried, evacuation is the most effective way of insulating to minimize heat losses from both the front and back sides of the collector. Figure I-6 shows the difference in the performance of two collectors with the same selective coating, but one being evacuated to less than 10^{-4} Torr, while the other has a conventional double glazing.

a
CONVENTIONAL
FLOW-THROUGH
SYSTEM COUPLED
TO THERMAL
STORAGE.

FIGURE
I - 5

COMPARATIVE ILLUSTRATION OF A FLOW-THROUGH
SOLAR COLLECTOR AND HEAT-PIPE SOLAR
COLLECTOR IN CONJUNCTION WITH THERMAL
STORAGE.

b
HEAT-PIPE SYSTEM
DIRECTLY COUPLED
TO THERMAL
STORAGE

FIGURE I-6

EFFECT OF VACUUM AND GLAZING ON COLLECTOR PERFORMANCE

SOLAR RADIATION: 250 BTU/HR-FT²

AMBIENT AIR TEMP: 50°F

Under the same operational conditions for example, the efficiency of the evacuated collector at a ΔT of 200°F is a factor of 2 times better. This certainly has significant practical implications for solar cooling applications.

Another useful feature of evacuation is the protective environment it offers for the selective absorber coating and the mirror inside the collector tube. When the vacuum stability question is resolved, this kind of protection offers many years of high performance without any surface deterioration of the mirror or the absorber plate.

SECTION II

SUMMARY OF WORK ACCOMPLISHED

CUSP REFLECTOR

A set of parametric equations was derived to describe the contour of a cusp reflector which is confined in a tubular envelope for vacuum encapsulation.

The mathematical analysis included the following aspects:

- . Analytical solutions for upper and lower parts of a cusp reflector confined in a tubular envelope.
- . Calculation of the aperture of a cusp reflector confined in a tubular envelope of given dimensions.
- . Calculation of concentration ratio and its relationship with the acceptance angle.
- . Relationship between a glass envelope of a given radius and the largest cusp reflector which fits inside.

A computer program was written to calculate reflector contour, acceptance angle, and concentration ratio for given absorber and tubular envelope radii.

Using various fabrication techniques, aluminum reflectors with high reflectance were formed in accordance with the analytical cusp curve. The microscopic and macroscopic accuracy of the reflectors was tested by optical and visual means and found to be satisfactory.

TUBULAR GLASS ENVELOPE

The tubular form of the glass envelope, surrounding the reflector and the absorber, introduces new optical phenomena which are different from the ones found in flat plate configurations.

Digital programs were developed to investigate the following phenomena.

- . Reflection losses from a tubular glass envelope and its comparison to losses from flat cover glasses.
- . Shading effects in an array of closely packed tubular collectors.
- . Mutual light scattering between the tubes in a closely packed modular array.
- . Trapping of a beam of light within a tube due to internal reflection.
- . Calculation of total absorbed light by a perfect cusp collector as a function of orientation and latitude throughout the day and throughout the year.

HEAT PIPE ABSORBER

The work on heat pipes included several important features which are critical for the overall performance and safety of the collector. These include

- . A computer program was set up to calculate the required fluid change and the performance limits of the heat pipes which were considered as absorber candidates. The program uses the properties of the fluid, wick and the pipe geometry as input data.
- . A search was undertaken to select the heat-pipe working fluid most suitable for solar collector applications. The screening criteria used in this selection are the high pressure conditions

inside the heat pipe and the chemical decomposition at elevated temperatures.

- . Four different wick concepts were examined for optimal performance. These include homogeneous wick, micro-scratched wick, screw thread wick and axially grooved wick.

- . Heat transfer from the heat pipe condenser to the solar system thermal reservoir has been analyzed. For given condenser dimensions the heat transfer for various collector orientations has been calculated.

- . Pressure containment capability of aluminum and copper heat pipes was investigated and safety factors up to 800°F were calculated as a function of temperature, wall thickness and the amount of fluid charge.

- . Three working models of copper heat pipes with various wicks were fabricated.

The temperature drop ΔT between the evaporator and the condenser surface is now being measured as a function of orientation at various power levels.

FLOW-THROUGH ABSORBER

As an alternative to the heat pipe absorber for removing heat from the cusp collector, the flow-through concept has been investigated. Calculations were made to predict the temperature rise of the fluid as a function of reduced absorber length. A flow-through absorber was designed and fabricated. This pipe has been then used to assemble a prototype cusp collector which was evacuated and tested for its thermal performance.

EVACUATION AND MAINTENANCE OF VACUUM

A temperature controlled bake out furnace to hold a whole collector was designed and fabricated. The vacuum manifold of

the furnace was attached to a mass spectrometer to monitor the evolving gasses during pump down of the collector. Pressure levels of 10^{-5} Torr in true life collectors have been achieved without the use of getters. The use of getters improved the vacuum.

In regard to long term vacuum maintenance, calculations were performed to predict the rate of helium penetration into a sealed tube. The pressure buildup inside the tube was calculated for various glass compositions and tube wall temperatures as a function of time.

INDOOR AND OUTDOOR TEST FACILITIES

To measure optical and thermal properties of collector tubes, indoor and outdoor test facilities were set up. The indoor facility includes a 2' X 5' solar simulation light source and a test stand to measure loss coefficients of finished collectors. Also, another test stand was designed and fabricated to analyze the performance of heat pipe absorbers under various loads and orientations.

An extensive outdoor test facility on the roof top of the Sullivan Park Research Center has been completed. The system measures collector performance under true life conditions and records all relevant data automatically on a teletype. Flow rate, inlet temperature, and collector orientation can be adjusted with high accuracy and repeatability. System shakedown and calibration of the instruments has been completed. Test procedures have been developed to measure the "efficiency curve" of a collector according to proposed NBS standards.

SECTION III

PROGRESS REPORT ON OBJECTIVES

COLLECTOR GEOMETRY AND ANALYTICAL MODELING

CUSP COLLECTOR ANALYSIS

The basic design philosophy behind the evacuated cusp collector is to have a true non-tracking and very low focussing collector capable of operating at temperatures higher than flat plate collectors throughout the year.

Figure I-2 shows the schematic of the cusp collector which will be analyzed in this report. The dimensions of the glass envelope and the absorber pipe are chosen such that the resulting concentration ratio of the cusp reflector will be slightly larger than unity.

It is the purpose of the overall analytical effort to have a complete mathematical model for collector performance which couples the thermodynamics and the optics of the collector to a complete model. Such a model may then be used to predict total cusp collector performance for heat-pipe and flow-through absorbers.

The energy exchange between the absorber mirror, glass cover and the ambient air is very complex, if natural convection and radiative exchanges are considered at the same time. Also the tubular shape of the glass envelope adds additional complication to the overall optics because of the involved phenomena associated

with the cylindrical shape. These are not encountered in the flat plate collectors.

In order to understand the collector in detail, it was decided to conduct separate studies of its thermal and optical properties. The parameters of interest can be seen from the steady-state heat balance equation.

Per unit aperture area, the steady-state heat balance equation for a cusp collector with flow-through and heat-pipe absorber can be written as follows:

Flow-Through:

$$Q_u = A_\alpha F_R \{ H R \gamma \rho_R^{<n>} \tau_{eff} \alpha_{eff} - \frac{1}{C} U_L (T_{fi} - T_a) \} \quad (III-1)$$

Heat Pipe:

$$Q_u = A_\alpha \{ H R \gamma \rho_R^{<n>} \tau_{eff} \alpha_{eff} - \frac{1}{C} U_L (T_r - T_a) \} \quad (III-2)$$

A_α = aperture area of the cusp reflector

F_R = collector heat removal factor

H = total normal incident radiation on a specific test surface

R = geometrical conversion factor from the plane of measurement to the plane of cusp reflector aperture

γ = fraction of the total insolation accepted by the cusp reflector

ρ_R = solar reflectance of the cusp reflector

$<n>$ = average number of reflections

τ_{eff} = effective transmittance of tubular envelope

α_{eff} = effective solar absorptance

C = concentration ratio (A_α/A_r)

A_r = surface area of the receiver

$U_L(T_r - T_a)$ = thermal losses per unit aperture area from an isothermal absorber at a temperature T_r to the ambient air at T_a .

$U_L(T_{f_i} - T_a)$ = thermal losses per unit aperture area from an absorber which is not isothermal

T_{f_i} = fluid inlet temperature for flow-through absorber

Below we will report the results of our analysis of some of the parameters in Eqs. III-1 and III-2.

OPTICAL ANALYSIS OF CUSP REFLECTOR

Introduction

The motivation behind the cusp reflector in conjunction with the tubular heat pipe absorber is threefold.

(a) A cusp reflector essentially "funnels" the light onto an absorber pipe, with a surface area equal to the intercept. The heat is removed from the inside of the pipe wall as useful energy. Consequently there are no conventional back losses in cusp collectors. This is a distinct advantage compared to other low concentration collectors like compound parabolic concentrators where back losses are costly to reduce. Also, for the same intercept area, the cusp collector has about half the absorber surface of the evacuated flat plate through which energy may be lost to the environment by radiation.

(b) The light collection characteristics of the cusp collector system equate to those of a flat plate by having an acceptance angle sufficiently large to collect light from most of the sky. As a result, direct or diffused light from a large fraction of the

hemisphere above the collector will reach the absorber. Consequently, the negative effect of haze, as found in focussing systems, is minimized.

(c) Within the acceptance angle any solar ray intercepted by the aperture of the cusp collector is transported in the form of photons directly to the cylindrical absorber surface rather than being conducted to the heat-transfer fluid through the fins of a flat-plate collector.

Although cusp collector performance is similar to the flat plate, there are distinct differences between the flat plate and a collector with a cusp mirror.

(1) The distribution of angles of incidence onto the absorber is not the same.

(2) The concentration ratio of the cusp collector can be increased above unity by decreasing the acceptance angle.

(3) Depending on the reflectance of the cusp mirror, the reflection losses will reduce the amount of solar flux received by the absorber.

Mathematical Description of the Cusp Mirror

The basic principle of the cusp mirror shown in Fig. I-3 is that any ray of light making an angle α or less with the plane of symmetry, will be reflected to the cylindrical absorber, either directly or through a series of multiple reflections. A restriction on the cusp geometry in tubular collector applications is the requirement that the mirror must fit into a fixed diameter tube (cylindrical glass envelope of the collector). The mathematical description of the mirror surface is given below.

First, the curve for the mirror surface must be divided into two sections, each one described by separate equations. Let α be called the critical angle, and a ray at critical angle be called a critical ray. The dividing point between the two curves occurs at the point B in Fig. III-1 where a critical ray tangent to the absorbing circle intersects the mirror curve. The lower curve (part of the curve between B and the contact point A) is defined such that any normal to this curve is tangent to the absorbing circle. The upper curve (the section of the mirror between B and C) is defined such that a critical ray hitting the mirror surface at any point in this section will be reflected tangent to the cylindrical absorber. In this configuration, any ray striking the surface of the mirror within an angle α of the plane of symmetry will eventually be reflected to the cylindrical absorber by means of single or multiple reflections.

The definition of the mirror is best done in parametric form.

The parameters are shown in Fig. III-1 where:

R = radius of the cylindrical absorber

ζ = radius of the cylindrical envelope

x, y = coordinates of a point on the cusp mirror

x_c, y_c = coordinates of a point on the cylindrical absorber

r = the distance from x_c, y_c to x, y

θ = the angle between x_c, y_c , the origin and the lower y-axis

θ_t = angle associated with the point where the cusp mirror and the cylindrical cover touch each other in the lower portion of the mirror

FIGURE III - I

COMPUTER PLOT OF TUBULAR CUSP COLLECTOR AND
DEFINITION OF SOME IMPORTANT PARAMETERS.

$\theta_{\max}$ = angle associated with the maximum opening of the cusp

θ_B = angle associated with the point B on the cusp where
upper and lower curve of the cusp mirror meet each other

x_m, y_m = coordinates of the maximum opening on the cusp collector

x_t, y_t = coordinates of the point where the cusp mirror and the
cylindrical cover touch each other in the lower portion
of the mirror

A = the point at which the cusp touches the cylindrical
absorber

α = the angle between a critical ray and the normal to
the aperture of the mirror (acceptance angle).

The origin of the coordinate system is taken as the center of
the cylindrical absorber. At $\theta = 0$, i.e. at point A, the mirror
curve will contact the cylindrical absorber. By using simple
geometrical relationships, it can be shown that x and y can be
expressed in the following parametric form:

$$x(r, \theta) = R \sin \theta - r(\theta) \cos \theta \quad (\text{III-3})$$

$$y(r, \theta) = -R \cos \theta - r(\theta) \sin (\theta) \quad (\text{III-4})$$

The derivatives of x and y with respect to θ are then given
by

$$\begin{aligned} \frac{dx}{d\theta} &= \frac{\partial x}{\partial \theta} + \frac{\partial x}{\partial r} \frac{\partial r}{\partial \theta} \\ &= R \cos \theta + r \sin \theta - \frac{\partial r}{\partial \theta} \cos \theta \end{aligned} \quad (\text{III-5})$$

$$\begin{aligned}\frac{dy}{d\theta} &= \frac{\partial y}{\partial \theta} + \frac{\partial y}{\partial r} \frac{\partial r}{\partial \theta} \\ &= R \sin \theta - r \cos \theta - \frac{\partial r}{\partial \theta} \sin \theta\end{aligned}\quad (\text{III-6})$$

Solution for the Lower Part of the Mirror

In terms of vectors, the orientation of the tangent at a given point (x,y) on the mirror curve is given by

$$\bar{t} = \bar{i} \frac{dx}{d\theta} + \bar{j} \frac{dy}{d\theta} \quad (\text{III-7})$$

It can also be shown that the direction of a vector from the point (x,y) on the mirror curve to a tangential point (x_c, y_c) on the cylindrical absorber is given by

$$\bar{r} = \bar{i} \cos \theta + \bar{j} \sin \theta \quad (\text{III-8})$$

We require now that in the lower part of the mirror curve, $\bar{r}$ is always normal to $\bar{t}$, i.e. the dot product must vanish:

$$(\bar{r} \cdot \bar{t}) = \frac{dx}{d\theta} \cos \theta + \frac{dy}{d\theta} \sin \theta = 0 \quad (\text{III-9})$$

By multiplying Eqs. (III-5) and (III-6) by $\cos \theta$ and $\sin \theta$ respectively and adding them we obtain

$$\frac{dx}{d\theta} \cos \theta + \frac{dy}{d\theta} \sin \theta = R - \frac{\partial r}{\partial \theta} \quad (\text{III-10})$$

From Eqs. (III-9) and (III-10), we obtain then

$$R - \frac{\partial r}{\partial \theta} = 0 \quad (\text{III-11})$$

or by integration

$$r = R\theta \quad (\text{III-12})$$

The equations describing the lower part of the mirror curve (i.e. from A to B) then have the following explicit form:

$$x = R \sin \theta - R\theta \cos \theta \quad (\text{III-13})$$

$$y = -R \cos \theta - R\theta \sin \theta \quad (\text{III-14})$$

where θ covers a range from 0 to $\theta_B = \pi/2 + \alpha$. (For θ_B see Eq. (III-23 below.)

Solution for the Upper Part of the Mirror

The orientation of a critical ray at the upper part of the mirror is given by

$$\bar{c} = \bar{i} \sin \alpha - \bar{j} \cos \alpha \quad (\text{III-15})$$

The critical ray will be reflected tangentially to the cylindrical absorber, the reflected vector being $\bar{r}$. Since $\bar{c}$ and $\bar{r}$ are unit vectors, their dot product with $\bar{t}$, i.e., the tangent to the mirror curve, must be equal.

$$(\bar{t} \cdot \bar{c}) = (\bar{t} \cdot \bar{r}) \quad (\text{III-16})$$

From Eqs. III-7 and III-8, we obtain by calculating the dot product the following expression

$$(\bar{t} \cdot \bar{c}) = \frac{dx}{d\theta} \sin \alpha - \frac{dy}{d\theta} \cos \alpha \quad (\text{III-17})$$

According to Eq. III-10, we have, however

$$(\bar{r} \cdot \bar{t}) = R - \frac{\partial r}{\partial \theta} \quad (\text{III-18})$$

Consequently,

$$\frac{dx}{d\theta} \sin \alpha - \frac{dy}{d\theta} \cos \alpha = R - \frac{\partial r}{\partial \theta} \quad (\text{III-19})$$

or integrated

$$x \sin \alpha - y \cos \alpha + b = R\theta - r \quad (\text{III-20})$$

where b is the integration constant.

Upon substituting x and y from Eqs. III-3 and III-4 respectively, we get:

$$R \cos (\theta - \alpha) + r \sin (\theta - \alpha) + b = R\theta - r \quad (\text{III-21})$$

The constant b can be obtained from the boundary condition at the point B, where the critical ray is reflected into itself.

The relation between α and θ is given by

$$\psi = \theta - (\pi/2 + \alpha) \quad (\text{III-22})$$

where ψ is the angle between the critical ray and the normal to the tangent at the point of reflection on the mirror. At the point B we have $\psi = 0$ so that

$$\theta_B = \pi/2 + \alpha \quad (\text{III-23})$$

Thus, upon substituting Eq. III-23 into Eq. III-21 we find

$$b = -R(\pi/2 + \alpha) \quad (\text{III-24})$$

Solving Eq. III-21 using b , one obtains,

$$r = R[\pi/2 + \alpha + \theta - \cos(\theta - \alpha)]/[1 + \sin(\theta - \alpha)] \quad (\text{III-25})$$

The explicit parametric equations describing the upper part of the mirror curve are then given by

$$x = R[\sin \theta - \frac{[\pi/2 + \alpha + \theta - \cos(\theta - \alpha)] \cos \theta}{1 + \sin(\theta - \alpha)}] \quad (\text{III-26})$$

$$y = -R[\cos \theta + \frac{[\pi/2 + \alpha + \theta - \cos(\theta - \alpha)] \sin \theta}{1 + \sin(\theta - \alpha)}] \quad (\text{III-27})$$

where θ covers a range from θ_B to $\theta_{\max} = 3\pi/2 - \alpha$. (For $\theta_{\max}$, see Eq. III-30 below.)

Calculation of Maximum Opening $2x_m$ and $\theta_{\max}$

The maximum opening $2x_m$ of the cusp mirror can be obtained from the equation

$$\frac{dx}{d\theta} = R \cos \theta + r \sin \theta - \frac{\partial r}{\partial \theta} \cos \theta = 0 \quad (\text{III-28})$$

with

$$\frac{\partial r}{\partial \theta} = \frac{R[1 + \sin(\theta - \alpha)]^2 - R[\pi/2 + \alpha + \theta - \cos(\theta - \alpha)] \cos(\theta - \alpha)}{[1 + \sin(\theta - \alpha)]^2} \quad (\text{III-29})$$

The solution is:

$$\theta_{\max} = 3\pi/2 - \alpha \quad (\text{III-30})$$

Then from Eqs. III-25 and III-26 we obtain, respectively

$$r_{\max} = R \frac{(\pi + \sin \alpha \cos \alpha)}{\sin^2 \alpha} \quad (\text{III-31})$$

and

$$2x_{\max} = \frac{2\pi R}{\sin \alpha} \quad (\text{III-32})$$

Calculation of the Concentration Ratio

The concentration ratio is defined as the ratio of the intercept area for the incident flux to the area of the absorber surface

$$C = \frac{2x_{\max} \ell}{2\pi R \ell} = \frac{1}{\sin \alpha} \quad (\text{III-33})$$

where ℓ is the collector length.

Relationship Between a Glass Envelope of Radius ζ and the Largest Cusp Collector Which Fits

As shown in Fig. III-1, the glass envelope will touch the cusp mirror at two points. At the point x_t, y_t the tangent to the mirror curve is also tangent to the cylindrical cover envelope. Then the normal from the center of the envelope $(0, y_c)$ to the point x_t, y_t must also be a normal to the curve. In this case, the equation of the normal satisfies the following relation:

$$x_t \left(\frac{dx}{d\theta} \right)_{\theta_t} + (y_t - y_c) \left(\frac{dy}{d\theta} \right)_{\theta_t} = 0 \quad (\text{III-34})$$

If Eqs. III-3, III-4, III-5, and III-6 are substituted into Eq. III-34 we obtain:

$$y_c \left(\frac{dy}{d\theta} \right)_{\theta_t} = r(\theta_t) \left(\frac{\partial r}{\partial \theta} \right)_{\theta_t} \quad (\text{III-35})$$

Remembering that in the lower part of the mirror curve we have $r(\theta_t) = R\theta_t$, we find:

$$y_c = - \frac{R}{\cos \theta_t} \quad (\text{III-36})$$

The second point where the mirror curve touches the glass envelope is (x_m, y_m) i.e. the point of maximum opening. Therefore (x_m, y_m) must satisfy the following relationship:

$$x_m^2 + (y_m - y_c)^2 = \zeta^2 \quad (\text{III-37})$$

or

$$y_c = y_m - \sqrt{\zeta^2 - x_m^2} \quad (\text{III-38})$$

or

$$y_c = y_m - R \sqrt{\zeta^2 / R^2 - \pi^2 / \sin \alpha} \quad (\text{III-39})$$

(For y_m , see Eq. III-40.)

Also, the center of the cover envelope must be the same distance from the point of maximum opening and the point of tangency (x_t, y_t) :

$$x_t^2 + (y_t - y_c)^2 = x_m^2 + (y_m - y_c)^2 \quad (\text{III-40})$$

or

$$x_t^2 + y_t^2 - 2y_t y_c = x_m^2 + y_m^2 - 2y_m y_c \quad (\text{III-41})$$

Using Eqs. III-3 and III-4, we obtain:

$$x_t^2 + y_t^2 = R^2 + r^2(\theta_t) \quad (\text{III-42})$$

$$x_m^2 + y_m^2 = R^2 + r^2(\theta_m) \quad (\text{III-43})$$

Then from Eqs. III-41, III-42 and III-43 we calculate:

$$y_c = \frac{r^2(\theta_m) - r^2(\theta_t)}{2(y_m - y_t)} \quad (\text{III-44})$$

where

$$y_m = -R \cos \theta_{\max} - R \frac{[\pi/2 + \alpha + \theta_{\max} - \cos(\theta_{\max} - \alpha)] \sin \theta_{\max}}{1 + \sin(\theta_{\max} - \alpha)} \quad (\text{III-45})$$

$$y_m = R \left[\sin \alpha - \frac{2\pi + \sin 2\alpha (-\cos \alpha)}{2 \sin^2 \alpha} \right] \quad (\text{III-46})$$

$$y_t = -R[\cos \theta_t + \theta_t \sin \theta_t] \quad (\text{III-47})$$

$$r(\theta_m) = R \frac{[\pi + \sin \alpha \cos \alpha]}{\sin^2 \alpha} \quad (\text{III-48})$$

$$r(\theta_t) = R\theta_t \quad (\text{III-49})$$

Equations III-36, III-39, and III-44 can now be solved simultaneously for three unknown parameters. For example if acceptance angle α and cover radius ζ are given, the radius R of the cylindrical absorber plus θ_t and y_c , the displacement of the center of the cover with respect to the center of the absorber ($x = 0, y = 0$) can be calculated. Also, if ζ and R are known, α , θ_t and y_c can be calculated. Similarly if, the concentration ratio C of Eq. III-3 and R of the absorber circle are known, ζ , y_c and θ_t can be calculated.

Figure III-2a and III-2b show the ray tracing obtained with a laser beam at 0° and 30° of incidence with respect to the plane of aperture. The aluminum cusp reflector templet was machined with high accuracy to follow the analytical solutions given above. It can be seen from these figures that the flux distribution around the absorber pipe is not isotropic. The bright area underneath the absorber pipe indicates a higher flux in this region. It is also important to note that the light passing through certain regions of the aperture undergoes multiple reflections before it hits the absorber. These two phenomena are essential in determining the performance of the cusp collector and these quantitative analyses will be one of the tasks associated with the digital Monte Carlo Ray Tracing Code.

OPTICAL ANALYSIS OF TUBULAR GLASS ENVELOPE

The Fresnel reflections from a cylindrical glass envelope for a given position of the sun vary appreciably from those from a flat glass cover. Because of the curvature of the cylindrical surface, the direct component of solar radiation hits the different

Figure III-2
Ray Tracing Patterns Obtained Using
A Laser Beam and A Computer Controlled
Machined Aluminum Templet

a

b

sections of the glass wall at different angles of incidence. At any given time of the day the angle of incidence at a test point on the glass wall may vary between 0 and 90 degrees depending on the collector-sun relationship. Since the Fresnel losses are angularly dependent, the light transmitted through the envelope will then be the average over the range of incidence angles.

To investigate the effect of a cylindrical glass envelope on the collector performance, analytical formulae were derived to analyze the associated phenomena, independent of the cusp reflector optics. Figure III-3 shows these phenomena schematically. Included are:

- . Fresnel reflections from cylindrical envelopes (Fig. III-3a)
- . Self shading effects in a modular array (Fig. III-3b)
- . Mutual scattering between adjacent tubes in a modular array (Fig. III-3a)
- . Light trapping by internal reflection (Fig. III-3c)

Scattering of light from a back plane, a notable contribution to the performance of evacuated flat plates, need not be considered in this case because of the presence of the cusp mirror.

The magnitude of gains and losses due to these phenomena is strongly dependent on the position of the sun, the orientation of the collector tubes, and the separation of tubes in a modular array. Calculations were made for three characteristic orientations as defined in Fig. III-4. In the case of the north-south polar mount, the collector axis is aligned north-south but is tipped up out of the horizontal plane by an angle ϕ equal to the local latitude. The aperture of the cusp faces south. In the east-west horizontal mount,

FIGURE III - 3

OPTICAL PHENOMENA ASSOCIATED WITH AN ARRAY OF CUSP COLLECTORS.

a) REFLECTIVE LOSSES

b) SHADING EFFECT BY ADJACENT TUBES

c) MUTUAL LIGHT SCATTERING BETWEEN ADJACENT TUBES AND LIGHT TRAPPING BY INTERNAL REFLECTION.

FIGURE III - 4

DEFINITION OF SOME CHARACTERISTIC COLLECTOR ORIENTATIONS.

the collector axis is oriented in the east-west direction and the normal to the cusp reflector aperture points towards the zenith. In the north-south vertical mount, the axis of the collector is oriented in the direction of the zenith and the normal to the cusp reflector aperture points south.

Fresnel Reflections

The results of a computer program (FRLTAP) written to compare the reflective losses from a tubular envelope and a flat plate cover for equal reflector aperture, are shown in Fig. III-5.

The collector orientation, declination angle of the sun, geographic location and the refractive index of the glass are given in the same figure. The calculation considers all direct radiation which would be intercepted by the aperture of a cusp reflector during the day. Any cut-off due to the acceptance angle of the cusp has not been considered.

The reflective losses from the tube achieve a maximum of 13.3% at solar noon. This is approximately 5.1% higher than the flat plate cover glass. However, the loss curves cross each other at about 9:45 a.m. and 2:15 p.m. solar time so that the tubes transmit more light than the flat plate in the morning and evening. An integration between 6:00 a.m. and 6:00 p.m. with the assumption of constant air mass, i.e. constant solar flux, shows that the flat plate reflects 1.3 times more direct solar radiation during such a day.

Self Shading and Orientation

The shading effect of collector tubes in a modular array can

have significant effects on the overall module performance depending on the orientation and the separation of the tubes. A computer program (EGRNEW) was developed to calculate the amount of solar radiation intercepted by the aperture of the cusp reflector throughout the day as a function of solar time. To minimize the complexity of the problem, the cusp reflector was assumed to be truncated and the aperture was set equal to the maximum cross section of the tube. Also, any light beam which was intercepted by the aperture was considered irrespective of the acceptance angle of the cusp reflector. Figures III-6 and III-7 show the energy intercepted by a cusp reflector in a tubular envelope relative to the energy intercepted by an uncovered cusp at solar noon on a given day. The 21st of June and December are depicted to demonstrate the effect of orientation and the time of the year on the daily performance of the collector. The changes of solar mass during the daily and seasonal apparent movements of the Sun were considered. The calculations were based on the effect of shading at a tube separation of 0.375".

Figure III-8 shows daily total intercepted energy versus the day of the year including shading effects and Fresnel losses. The asymmetry between the first and second half of the year is due to periodic annual transmissivity changes in the atmosphere which are included in the calculation.⁵ In Fig. III-9, yearly total energy intercepted versus the latitude of the geographic location is displayed. The integration limits chosen were 6:00 a.m. to 6:00 p.m.

Mutual Scattering and Light Trapping

Mutal light scattering and light trapping may contribute to the performance of the collector array depending on the tube

FIGURE III-6

RELATIVE ENERGY INTERCEPTED ON JUNE 21 VERSUS SOLAR TIME
INCLUDING SHADING EFFECT.

FIGURE III-7

RELATIVE ENERGY INTERCEPTED ON DECEMBER 21 VERSUS SOLAR TIME INCLUDING SHADING EFFECT.

FIGURE III - 8

DAILY INTERCEPTED ENERGY VERSUS DAY OF THE YEAR INCLUDING SHADING EFFECT AND FRESNEL LOSSES.

FIGURE III - 9

YEARLY TOTAL ENERGY INTERCEPTED VERSUS THE LATITUDE OF THE GEOGRAPHIC LOCATION.

separation and orientation. The right hand ordinate of Fig. III-10 displays the percent contribution to the energy intercepted by mutual scattering and light trapping. As can be seen from this figure, the scattering-trapping contribution does not exceed 3 to 4% of the intercepted light even at solar noon. Smaller tube separations will increase the flux seen by the absorber.

Conclusions

Although a single tube reflects about 5% more of direct solar radiation at solar noon than a flat plate, total energy intercepted by the truncated cusp from sunset to sunrise is about the same for both.

Depending on the collector orientation and separation, the effect of shading on the performance of the collector array can be appreciable. The orientation of the collector must be chosen carefully by considering the geographic location and peak solar climate control system demand during the year.

North-south polar mount is the most insensitive orientation for a tubular collector with regard to changes in latitude of geographic location and apparent movements of the Sun throughout the year.

Mutual scattering between adjacent tubes and internal reflections within the tubes can improve the performance of the collector array by a few percent. This contribution will depend on the orientation and separation of the tubes.

THERMAL LOSSES DUE TO CONVECTION AND CONDUCTION

Due to the complex geometry of the cusp reflector and the eccentric location of the absorber pipe, a rigorous treatment of the

FIGURE III - 10

MUTUAL LIGHT SCATTERING BETWEEN TUBES AND INTERNAL REFLECTION.

convective heat transfer is difficult. However, accurate laboratory experiments can give reliable loss data over a wide range of pressure and absorber temperature.

A transient method was chosen to determine the overall heat loss coefficient U_L of a cusp collector. The principles of this method can be seen from the governing time differential heat balance equations for a single cover collector with an isothermal absorber and fluid. Under the assumption that the cusp mirror does not participate in heat exchange, one can write⁶

$$(mC)_r \frac{dT_r}{dt} = S + U_1(T_g - T_r) \quad (\text{III-50})$$

where the subscripts r and g represent receiver (absorber) and glass envelope respectively, U_1 is the heat loss coefficient from the receiver to the glass envelope, m is mass, C is the heat capacity, and S is the energy flux into the receiver. The heat balance equation for the glass envelope can be written as

$$(mC)_g \frac{dT_g}{dt} = U_1(T_r - T_g) + U_2(T_a - T_g) \quad (\text{III-51})$$

where U_2 is the loss coefficient from the glass envelope to the ambient air at the temperature T_a . Equations III-50 and III-51 can be solved simultaneously. However, it is simpler if we make the assumption that $(T_g - T_a)/(T_r - T_a)$ remains constant at its steady state value. In this case one can write

$$U_2(T_g - T_a) = U_L(T_r - T_a) \quad (\text{III-52})$$

where U_L is the overall heat loss coefficient and is given by

$$U_L = \frac{U_1 U_2}{U_1 + U_2} \quad (\text{III-53})$$

For a constant ambient temperature T_a in the laboratory, we find by differentiation of Eq. III-52

$$\frac{dT_g}{dt} = \frac{U_L}{U_2} \frac{dT_r}{dt} \quad (\text{III-54})$$

Using Eqs. III-50, III-51 and III-54, we obtain the following time differential equation for the receiver temperature:

$$\left\{ (mC)_r + \frac{U_L}{U_2} (mC)_g \right\} \frac{dT_r}{dt} = S - U_L (T_r - T_a) \quad (\text{III-55})$$

The bracketed term in Eq. III-55 represents the effective heat capacity of the collector.

The essence of the transient method is to elevate the temperature of the receiver by resistance heating up to a specified temperature, then turn off the power (i.e. $S = 0$) and record the glass envelope and receiver temperatures continuously during the system cool down. In this case we have

$$\left\{ (mC)_r + \frac{U_L}{U_2} (mC)_g \right\} \frac{dT_r}{dt} = -U_L (T_r - T_a) \quad (\text{III-56})$$

Solving for U_L we obtain

$$U_L = - \frac{(mC)_r \frac{dT_r}{dt} U_2}{(mC)_g \frac{dT_r}{dt} + U_2(T_r - T_a)} \quad (\text{III-57})$$

The quantities $(mC)_r$ and $(mC)_g$ are generally known and U_2 can be calculated with good accuracy. The rate of temperature change dT_r/dt and the excess temperature $(T_r - T_a)$ can be obtained experimentally, by observing the cooling of the absorber after turning off the power to the absorber (i.e. $S = 0$). Thus Eq. III-57 can be used to determine U_L as a function of receiver temperature and pressure inside the tubular envelope.

The experimental setup used for measuring the transient behavior of the cusp collector model is shown in Fig. III-11. Figure III-12 shows the cooling curves obtained at various pressures by using a 1 foot long model of the cusp collector design. These data were then least squares fitted to an exponential function and derivatives dT_r/dt were computed at various receiver temperatures. From Eq. III-57 and known T_a , the overall loss coefficient U_L was calculated as a function of excess receiver temperature $\Delta T_r = T_r - T_a$. The results are shown in Fig. III-13.

Very useful information can be obtained when U_L is plotted versus pressure at given absorber temperatures. This is shown in Fig. III-14. The dotted portion of the curves of U_L versus pressure in Fig. III-14 indicate some uncertainty in the data points obtained in the 10^{-2} Torr range. Several attempts to measure U_L at 10^{-2} Torr yielded data which were not repeatable and had appreciably higher values than would be expected from U_L as a monotonic function of pressure. This anomaly in heat transfer

FIGURE III - II

TEST SETUP TO MEASURE THE HEAT LOSS COEFFICIENT U_L OF A CUSP COLLECTOR BY TRANSIENT METHOD.

FIGURE III-12

COOLING OF AN ISOLATED ABSORBER IN A CUSP COLLECTOR AS A
FUNCTION OF TIME.

$T_{AMB} = 77^{\circ}F$

100 ON Y-AXIS $\approx 356^{\circ}F$

FIGURE III-13

OVERALL HEAT LOSS COEFFICIENT U_L OF A
CUSP COLLECTOR AS A FUNCTION OF EXCESS
ABSORBER TEMPERATURE.

FIGURE III - 14

OVERALL LOSS COEFFICIENT U_L OF A CUSP COLLECTOR AS
A FUNCTION OF RESIDUAL COLLECTOR PRESSURE AT VARIOUS
ABSORBER TEMPERATURES.

$T_{AMB} = 77^\circ F$

seems to increase as the absorber temperature increases. A proven explanation for this behavior is not yet available. Heat losses in this pressure range are under continued investigation.

The data in Fig. III-14 include the radiative losses from the receiver. However, it is possible to extract direct information on convective and conductive losses from the data by subtracting the U_L values obtained at pressures lower than 1×10^{-4} Torr. In the present experiment, using a thermally isolated absorber, the contribution of conductive losses to the overall heat transfer can safely be assumed to be negligible below this pressure range. Under this assumption, Fig. III-15 displays the heat transfer losses due to conduction and convection as a function of pressure at several receiver temperatures.

The application of this data to actual performance calculations must be done carefully. The absorber pipe in the simulation test module had no connection to the headers and was supported by a special glass structure for minimum thermal loss. Therefore, application to a working model should include the heat leaks due to the header mechanical support mechanism for the absorber.

Conclusions

Laboratory experiments yield accurate and detailed heat loss data, which can be used in actual performance calculations. However, realistic corrections must be made for header and support losses in working collectors.

The convective and conductive losses in a cusp collector show a strong dependence on the pressure inside the tube and the plate temperature.

To make full use of the vacuum, the pressure should be 10^{-4} Torr or lower.

FIGURE III - 15

There appears to be an anomaly in the U_L values around 10^{-2} Torr which requires further investigation.

PRELIMINARY WORK ON MONTE-CARLO RAY TRACING PROGRAM

A Monte-Carlo ray tracing routine for the heat-pipe absorber-cusp reflector combination is currently under development. The purpose of the program is to determine the fraction of light which is absorbed by the heat pipe and to gain information on the variation of light intensity at different locations on the absorber.

For any fixed angle of incidence, a ray is chosen at random (position and wavelength) and followed analytically as it strikes the mirror and reflects, until it finally strikes the heat pipe or is lost. After each reflection, the fraction of energy remaining is computed.

The heat pipe is divided into equal segments. The fraction of each ray reaching each segment is summed. The fraction of the total energy reaching any segment of the heat pipe can then be computed, if a statistically significant number of rays have been chosen. (The current code requires 2000 rays.)

Initially, the mirror shape is assumed to be perfect and the rays are two-dimensional. It is intended however to extend the program to include the mirror imperfections by allowing the normal to fluctuate randomly about its mean so that some scattering will be accounted for. In addition, the final form of the routine will be extended to three-dimensional rays.

The flow diagram of the Monte-Carlo ray tracing routine for the cusp collector is given in Fig. III-16. A detailed description of the software will be given in the next progress report. Below we summarize the information expected to be obtained from this

Figure III-16
Flow Diagram of the Monte Carlo
Ray Tracing Program for the Cusp Reflector

simulation computer code:

- . Flux distribution around the heat pipe at various angles of incidence. (Information necessary for wick design.)
- . Average number of reflections $\langle n \rangle$ for various angles of incidence. (Information necessary to calculate $\rho_R^{\langle n \rangle}$, the effective reflectance of the cusp mirror for given angles of incidence.)
- . Percent contribution from various sections of the cusp mirror to the total energy absorbed. (Information necessary to optimize the truncation in order to reduce the mirror surface area.)
- . The effect of uncertainties in the mirror fabrication on the collector performance. (Information necessary to define tolerances in mirror fabrication.)
- . The effect of uncertainties in the relative positioning of the absorber and the mirror on the collector performance. (Information necessary for collector assembly.)
- . The effect of collector orientation and geographic latitude on the overall performance of the collector. (Information necessary for system installation.)
- . The optimal tube separation in a modular array for minimizing shading effects. (Information necessary to optimize collector array performance.)

HEAT PIPES

Heat pipes have replaced many conventional heat transfer devices because of their capability to transfer heat at high rates and over relatively long distances with a very small temperature drop.

In solar thermal applications using little or no focussing

of sunlight, prevailing fluxes are in the order of a few tenths of a watt per cm^2 . Therefore in general, the heat-carrying capacity of regular heat pipes should not impose any limitations. A conventional screen wick can tolerate, for example, internal evaporator heat fluxes up to 5 to 6 watts per cm^2 . The solar aperture areas of the cusp reflectors fabricated for this project are about 1300 cm^2 , so that the total intercepted energy per collector tube should never exceed 150 watts or 0.115 watts per cm^2 .

DESIGN REQUIREMENTS

- . The proposed evacuated collector will normally operate in a temperature range of $250 - 300^\circ\text{F}$ and the heat will be transferred via the condenser into the heat storage fluid at manifold inlet temperatures of $200^\circ - 220^\circ\text{F}$.

- . Depending on the geographic location and the purpose of the application, the heat pipe will have to operate at various angles with respect to the horizontal.

Besides these normal operational criteria extreme conditions will have to be considered.

- . Thermal runaway: A selectively coated heat pipe contained in a fully evacuated tubular collector can achieve stagnation temperatures in the order of 800°F . Consequently, heat-pipe design will have to consider extreme pressures which may develop as a result of such temperatures.

- . Freezing: Due to the diode action and the low thermal mass, night cooling may result in subzero temperatures at the heat pipe. A speedy recovery of the heat pipe action from a cold start is desirable. Provisions in the design must be made to recover from the frozen state quickly.

PHYSICAL DIMENSIONS

The exterior dimensions and the shape of the heat pipe for the proposed collector are shown in Fig. III-17. It consists of a 48" long straight section (absorber), a jog (0.958" upward and 2" long) and another 5" section (condenser) with a flange to serve as a base for the glass-to-metal seal. The jog joining the absorber and the condenser is a design requirement to facilitate simple assembly of the heat pipe, header and the tubular glass envelope.

SELECTION OF HEAT PIPE DESIGNS AND MATERIALS

Wick Structure and Shell Design

Recent developments in heat pipe technology have shown that the wick and shell of the heat pipe can be made out of the same material with great simplicity in design and low manufacturing cost.

Several such designs have been chosen for study during this phase of the program. All are based on developments which came early in the evolution of practical heat pipes: A layer of screen pressed against the wall, circumferential screw thread grooves, axial grooves down the length of the heat pipe and "micro scratches" on the wall surface, functionally equivalent to very fine threads. These wick configurations are shown schematically in Fig. III-18. The photographs (Figs. III-19a to III-19e) depict the details of screen, the extruded grooves, screw threads, and a series of "micro scratches" on the interior wall surface, respectively. All four wick configurations have been shown to have relatively high permeability and good heat transfer capability when operated horizontally or in the reflux mode. Depending on the thickness of the wicking structure,

FIGURE III-17
PHYSICAL DIMENSIONS OF THE HEAT PIPE

FIGURE III - 18

SCHEMATICS OF THE WICK CONFIGURATIONS
INVESTIGATED FOR ADVANCED COLLECTOR
ABSORBER.

CONCENTRIC
SCREEN WICK

RECTANGULAR
AXIAL GROOVES

MICRO-SCRATCH
HEAT PIPE WALL

SCREW THREAD
HEAT PIPE WALL

FIGURE III - 19

A. Screen Wick

B. Extruded Grooves - Cross Section

FIGURE III - 19

CONTINUED

C. Screw Thread - Top View

D. Screw Thread - Lateral Section

FIGURE III - 19

CONTINUED

E. Microscratches

a simple flow shield or artery may be necessary to shield the fluid flow from entrainment. This artery serves only as a shield and does not really function as an integral part of the wick, as arteries do in most heat pipes reported in the literature.

For the reflux mode chosen for this program, the wick inside the heat pipe is not required to carry fluid long distances by means of capillary action. In other words, the system will work in gravity-aided mode. A wick design which is sufficient to wet the sides and top by pulling fluid from the rivulet running downhill along the bottom of the heat pipe is adequate. A summary of wick design features, their fabrication methods and their evaluation is given in Table III-1. Shells and fluids are also included.

Wick and Shell Material

Copper and alloys of aluminum were chosen in this phase of the program as materials to fabricate integral wick and shell. These choices were based on the following reasons:

- . Good workability.
- . Availability of high grade stock.
- . Extensive experience as heat pipe material; high conductance.
- . Proven compatibility with well known working fluids.
- . Long experience as solar collector absorber material in conjunction with selective coatings.
- . Known technology to facilitate leak tight glass-to-metal seals.

An obvious disadvantage of both aluminum as well as copper is the need for thick tube walls to resist increasing vapor pressures which may develop in runaway conditions. The significant parameters with regard to their ability to contain pressure and the trade-offs

TABLE III-1

Summary of Selected Heat Pipe DesignFeatures and Material Properties

<u>Components/ Materials</u>	<u>Comments and Characteristic Dimensions</u>	<u>Heat Transfer and Other Properties</u>	
HEAT PIPE SHELL	Aluminum	Weldable Aluminum Alloys 5052 H-16 or 6063-T5 Tube OD 0.987" Tube ID 0.870"	Thermal Conductivity: 122 BTU/hr ft ² °F Yield Strength: 40 X 10 ³ lbs/in ² Annealed: 8 X 10 ³ lbs/in ² Heat Capacity: 23 X 10 ⁻² BTU/lb°F Density: 9.8 X 10 ⁻² lb/in ³ Annealing Temp: 775°F
	Copper	OFHC Copper ODA 10 ² Tube OD 1" Tube ID 0.870"	Thermal Conductivity: 222 BTU/hr ft ² °F Yield Strength: 40 X 10 ³ lbs/in ² Annealed: 10 X 10 ³ lbs/in ² Heat Capacity: 9.2 X 10 ⁻² BTU/lb °F Density: 32.3 X 10 ⁻² lb/in ³ Annealing Temp: 700-1200°F
<hr/>			
WICKS	Copper Screen	Spiral Wrapped 1 1/2 layers of 100 X 100 mesh screen	Permeability: P-M Capillary Pumping: G Thermal Conductance: P Cost Complexity: G
	Circumfer- ential Grooves	Scrow thread 14 grooves/inch 0.005 inch deep 60° equilateral triangle shape	Permeability: G Capillary Pumping: F Thermal Conductance: M-G Cost Complexity: G
	Axial Grooves	Extruded rectangular 36 grooves around circumference groove width: 0.035" groove depth: 0.035"	Permeability: P-M Capillary Pumping: G Thermal Conductance: G Cost Complexity: M
	Micro- Scratch	Micro grooves cut with a special tool densely packed groove width: 0.003" groove depth: 0.001"	Permeability: P Capillary Pumping: G Thermal Conductance: G Cost Complexity: G

TABLE III-1CONTINUED

<u>Components/ Materials</u>	<u>Comments and Characteristic Dimensions</u>	<u>Heat Transfer and Other Properties</u>
Water	Compatible with copper	Low thermal conductance High liquid transfer factor High pressure Chemically stable
FLUIDS FC-75	Compatible with aluminum	Low thermal conductance Low liquid transfer factor Low pressure Chemically stable (up to 700°F)

KEY

G - Good

M - Medium

F - Fair

P - Poor

will be evaluated in the section on pressure containment. However, neither copper nor aluminum is a limiting factor in the early phases of the program where thermal performance of prototype collectors will be assessed at operating temperatures (250 - 300°F) under controlled laboratory conditions.

In order to be compatible with water, a relatively pure copper alloy (C10100 or C10200, for example) must be used, so that no detailed selection process was conducted here. Relevant mechanical and thermal properties of pure copper are given in the section on pressure containment.

A more extensive investigation was conducted for aluminum. Tubing can be most easily extruded with longitudinal grooves from 6000-series alloys, and 6063 was the choice of the extrusion supplier. This alloy has reasonable room temperature mechanical properties and is weldable, although a filler material must be used.

Tubing and end caps were welded together in combinations of a 3000-series alloy (3003), a 5000-series alloy (5052) and a 6000-series alloy (6063), and the most satisfactory welds were obtained using 5052 tubing for the envelope and 5052 sheet for the end caps. This alloy was chosen for use in aluminum heat pipes where the wick is not an integral part of the tube.

Wick Fluids

In the selection of the wick fluid (or working fluid), three criteria must be met: suitable fluid and thermal properties, compatibility with the potential container-wick material, and chemical stability at elevated temperatures.

The fluid properties normally considered when comparing or selecting heat pipe fluids are customarily combined into two "Figures of Merit": The "O-g Figure of Merit", or liquid transport factor, $\sigma\lambda/\nu$, where λ is the latent heat of vaporization, σ is the surface tension, and ν the kinematic viscosity; and the "l-g Figure of Merit", or wicking height factor, σ/δ , where δ is the density. The first is largely irrelevant in selecting a working fluid for a system to be operated in the reflux mode. The second is important in comparing the ability of fluids to wick upward from a rivulet flowing along the bottom of an inclined heat pipe.

The most severe design condition for the solar heat pipe application is the thermal run-away condition, which occurs when the collector is exposed to full solar input and all heat-transfer fluid is drained from the circulating system. It is estimated that under these conditions the heat pipe may reach temperatures of at least 800°F. It is a requirement that after this condition has been discovered and corrected, the collector stays intact and the heat pipe performs normally again. Two problems are imposed by this requirement:

- . The internal pressure in the heat pipe must not become so high that the pipe fails.
- . No decomposition of the working fluid (which could generate noncondensable gas) can be allowed to take place.

All real fluids are compromises. Ruling out liquid metals such as mercury because of the danger of handling such materials, water is the most stable working fluid known at these

temperatures. Unfortunately water has an extremely high vapor pressure at required specific volumes when the temperature rises to 800°F and above. (See Section on Pressure Containment.) Other choices, various organic substances, have a tendency to decompose at temperatures above about 530°F or even lower in some cases. A fluid which has been successfully used in heat pipes, and which shows stability to at least 700°F is an inert fluorocarbon, FC-75 (octofluorobutyl ether) sold by the 3M Company. This fluid has a low vapor pressure at 800°F and is presently undergoing tests at high temperatures.

Other fluids which have been successfully used in heat pipes were evaluated and rejected. A list of these fluids, and the reasons for their rejection is presented in Table III-2. Fluid properties of water and FC-75 currently considered as possible candidates are given in Appendix A.

A summary of the selected design features and the material properties for the manufacturing of the prototype heat pipes in this phase of the program are depicted in Table III-1.

Anticipated combinations of materials and the chosen wick designs for fabricating the prototype heat pipes for this phase of the program are listed in Table III-3.

PERFORMANCE LIMITS

The performance of a heat pipe follows the basic principles of fluid mechanics. Consequently, heat transfer performance limits associated with fluid mechanics will also be encountered in heat pipe performance. These limits occur for two reasons:

- . Liquid flow limits in the wick.
- . Vapor flow limits in the shell.

TABLE III-2Fluids Other Than Water ConsideredFor Use as Solar Absorber Heat Pipe Fluid

<u>Fluid</u>	<u>Reason for Rejection</u>
Dow A (Eutectic mixture of diphenyl and phenylether, Dow Chemical)	Unstable at high temperatures
"Freons" (Series of halogenated hydrocarbons)	Those tested at high temperatures tend to be unstable
CP-9, CP-63 (Proprietary chemicals, Monsanto)	Unstable
PFP-1 (Proprietary heat transfer fluid, Sun Oil Company)	Unstable above 500°F
Xylene Methanol Acetone	Untried above 400°F or, in the case of xylene, 500°F. Will be considered in future program.
Ammonia	Extremely high pressure.
FC-75 (Octofluorobutyl ether, 3M Company)	Stable at high temperature; compatible with aluminum. Under investigation.

TABLE III-3
Design and Material Combinations
for Prototype Heat Pipes

<u>Prototype Number</u>	<u>Container</u>	<u>Combination Wick</u>	<u>Fluid</u>
1	Copper	Screen	Water
2	Copper	Spiral Grooves	Water
3	Copper	Micro Scratch	Water
4	Aluminum	Screen	FC-75 *
5	Aluminum	Axial Grooves	FC-75 *
6	Aluminum	Micro Scratch	FC-75 *

* Tentative, under investigation

Liquid flow problems may occur due to insufficient liquid flow return to the evaporator wick. This is usually caused by insufficient wick pumping (wicking limit) or by the formation of bubbles in the wick (boiling limit). Vapor flow problems may be caused by the sonic flow limit, entrainment limit, and/or pressure gradient limit.

These performance limits have been discussed previously in the literature by Busse,⁷ Kumme,⁸ and Feldman.⁹ The calculations for the heat pipes under investigation were carried out using a Fortran II code developed in this phase of the solar collector program. The mathematical formulation and the associated computer code are given in Appendix B. The summary of the results are given below.

Results of the Computer Analysis

Two of the wick structures under test could readily be studied by means of the computer program:

- . The homogeneous screen wick.
- . The axially extruded wick.

The homogeneous wick was analyzed assuming both water and FC-75 as the working fluid. The axially extruded wick was analyzed only with FC-75, as there are no plans to fabricate an axially-grooved/copper/water heat pipe.

In all cases, the "wick pumping" limit calculations indicated that, as expected, a single layer of screen or coarse wall grooves cannot transport significant fluid by capillary forces alone. Since the condenser is nearly an inch above the evaporator due to the "dog leg", even in the horizontal mode gravity effects must be

included in a realistic analysis. This is somewhat difficult to do. However, load carrying capabilities in excess of 250 watts have been demonstrated in laboratory tests. This is at least 100 watts more than the heat pipes will ever experience operating as a solar absorber in the present advanced collector.

The sonic limit in screen wick and axially grooved heat pipes with FC-75 should be equal because the vapor flow area is identical. The limit was calculated to be 2.5 kilowatts, also a load far in excess of any this pipe will experience in use. The sonic limit for the water heat pipe was calculated to be 13.6 kilowatts. Although not calculated, the sonic limit for the water pipes with micro scratch or threaded wicks will be slightly larger than this as the vapor area is slightly larger.

Because of its high surface tension, the calculated entrainment limit of the screen wick/water heat pipe was 20.8 kilowatts, but the corresponding values for the screen wick/FC-75 and the axial groove/FC-75 heat pipes were 450 watts and 185 watts, respectively. The latter corresponds to only a 60% safety factor, indicating that test results must be analyzed carefully for signs of this phenomenon. However, under test, the pipe of this design carried 250 watts successfully. This discrepancy is probably due to the uncertainty in assigning a value to the surface tension of FC-75 at 300°F. No unusual results indicated that entrainment was taking place inside the pipe observed. Calculated or experimentally determined performance limits for heat pipes under consideration for solar absorbers are summarized in Table III-4. The pressure gradient limit is also included.

TABLE III-4

Calculated Performance Limits

<u>Wick/Fluid Configuration</u>	<u>Wick Pumping Limit [kW]</u>	<u>Entrainment Limit [kW]</u>	<u>Sonic Limit [kW]</u>	<u>Pressure Gradient Limit [kW]</u>
Screen/Water	>0.25	20.8	13.6	6.3*; 45**
Spiral Grooves/Water	>0.25 [†]	20.8 [†]	13.6 [†]	6.3*; 45** [†]
Micro Scratch/Water	>0.25 [†]	-	-	-
Screen/FC-75	>0.25	0.45	2.5	1.7*; 12**
Axial Grooves/FC-75	>0.25	0.185	2.5	1.7*; 12**
Micro-Scratch/FC-75	>0.25 [†]	-	-	-

*Horizontal

**Vertical

†Estimated

PRESSURE CONTAINMENT

Since these heat pipes may be exposed to a temporary high temperature excursion, the buildup of internal pressure in the pipes is of concern and must be known accurately to determine the stresses in the heat pipe wall. Therefore, the internal pressure in heat pipes charged with water and FC-75 has been calculated as a function of temperature. When all of the fluid in the pipe is in the vapor state (which occurs at the temperature where the total internal volume of the pipe divided by the fluid inventory equals the saturation specific volume), the pressure no longer rises with temperature along the saturation line, but rather travels along the appropriate constant specific volume curve.

One exception to these curves is the case in which the specific volume calculated for a given heat pipe is less than the critical volume. If an attempt were to be made to raise such a pipe to the critical temperature, at some lower temperature the heat pipe will completely fill with fluid and burst.

Curves for both water and FC-75 are presented in Figs. III-20 and III-21.

In order to consider the pressure containment problem meaningfully, the mechanical properties of the chosen envelope materials must be known. The tensile strengths of pure copper and various weldable aluminum alloys are presented in Fig. III-22. Using this information, a burst pressure can be calculated:

$$P_{\text{burst}} = 2s \frac{R_o - R_i}{R_o + R_i} \quad (\text{III-58})$$

FIGURE III - 20

VAPOR PRESSURE IN A WATER HEAT PIPE AS A FUNCTION OF SPECIFIC VOLUME AND TEMPERATURE

FIGURE III - 21

VAPOR PRESSURE IN FC-75 HEAT PIPE AS A FUNCTION OF SPECIFIC VOLUME AND TEMPERATURE.

where s is the tensile strength, and R_o and R_i are the outside and inside radii, respectively. Choosing an appropriate safety factor, an allowable pressure is determined, and a maximum permissible charge is read from the pressure-temperature charts. For a given wick design, the charge required to operate the pipe can then be calculated and compared with the maximum safe charge, and the pressure at any point can be found from the chart.

One problem associated with this procedure is that it becomes evident after a few calculations that the major stumbling block is not high pressure, but poor resistance to stress of wall materials. In addition to having low tensile strength at temperatures above 600 K (620°F), both aluminum and copper are subject to creep.

Figure III-23 shows a burst analysis for a copper/water heat pipe similar to those delivered in this program. The saturation pressure is plotted against temperature, as is a maximum "safe" pressure (half the burst pressure). The two lines cross at a temperature corresponding to the highest temperature to which that heat pipe may be exposed, 507 K (453°F). A similar analysis for an FC-75/aluminum heat pipe is shown in Fig. III-24. This pipe is supercritical before it bursts; again, care must be used to ensure the specific volume calculated for the pipe is greater than the critical volume, or the pipe will burst.

Tests were run with copper/screen/water, copper/spiral grooves/water and aluminum/axially grooves/FC-75 heat pipes and minimum charges for proper performance were determined. The calculated maximum allowable temperatures with 100% safety factor are listed in Table III-5. It is evident from this data that neither copper

FIGURE III - 23

DETERMINATION OF MAXIMUM SAFE PRESSURE
IN COPPER/COPPER/WATER HEAT PIPE

FIGURE III - 24

MAXIMUM SAFE PRESSURE, A1/FC-75 HEAT PIPE

TABLE III-5Pressure Containment Data

<u>Heat Pipe Configuration</u>	<u>Fluid Charge</u>	<u>Max Temperature with 100% Safety Factor</u>
Copper/Screen/Water	45 cm ³	453°F
Copper/Spiral Grooves/Water	125 cm ³	453°F
Aluminum/Axial Grooves/FC-75	140 cm ³	536°F

nor aluminum heat pipes of the dimensions contemplated would be able to withstand the pressures in the case of runaway conditions where temperatures might exceed 800°F.

Other ways of avoiding this problem are to use a wall material with higher strength at temperature, or to use more exotic means of temporarily removing some of the charge or increasing the volume until the high temperature condition passes. The investigation of this problem is continuing.

TESTING OF HEAT PIPES

The use of heat pipes as solar absorbers requires a careful investigation of the heat transfer steps between the evaporator surface and the heat-transfer fluid removing heat from the condenser end. The "fin efficiency" of a heat pipe collector will be determined by the intrinsic thermal conductance (U) of the heat pipe. The temperature drop between the absorber surface and condenser surface can be written as

$$\Delta T = Q \cdot U^{-1} \quad (\text{III-59})$$

where Q = thermal load

$$\Delta T = T_{\text{abs}} - T_{\text{cond}}$$

$$U^{-1} = \frac{1}{A_{\text{abs}} h_{\text{evap}}} + \frac{1}{A_{\text{cond}} h_{\text{cond}}}$$

h_{evap} = film coefficient for the evaporator

h_{cond} = film coefficient for the condenser

T_{abs} = absorber surface temperature

T_{cond} = condenser surface temperature

A_{abs} = absorber surface area

A_{cond} = condenser surface area

The method of determining ΔT which has given the best results in the past has been the highly empirical one of determining h_{evap} and h_{cond} experimentally for a given wick/fluid combination. Using Eq. III-59 and typical film coefficients, expected temperature drops for various types of heat pipes have been calculated. The results for $Q = 250$ watts and $Q = 116$ watts (average midday insolation load) are tabulated in Table III-6. It is noteworthy that the microscratch /FC-75 configuration yields the lowest ΔT as compared to screen and axial grooves in aluminum heat pipes. This trend is also in agreement with experimental data reported below.

Test Procedure and Results

The heat pipes were tested by applying 250 watts by means of electrical resistance heaters at the lower quadrant of the heat pipe. Cooling was provided by blowing ambient air over the condenser at a rate sufficient to maintain the operating temperature at 420K (300°F), the projected operating temperature of the heat pipe in the system.

Power was applied, and temperatures were allowed to stabilize. After the data was taken, the pipe was allowed to cool, the tilt angle changed, and the test repeated. Each pipe was tested horizontally, vertically, and at a 45° angle, the last two test conditions with the evaporator down.

In order to determine the fluid charge required, an estimate was made, fluid put into the pipe, and then fluid was incrementally removed until the pipe dried out. Ten cc more fluid was then added, and that charge was determined to be the final charge.

TABLE III-6

Film Coefficients and Calculated Temperature Drops
for Prototype Solar Absorber Heat Pipes

<u>Heat Pipe Configuration</u>	<u>h_{evap} [BTU/hrFt²°F]</u>	<u>h_{cond} [BTU/hrFt²°F]</u>	<u>ΔT(at 250 Watt) [°F]</u>	<u>ΔT(at 116 Watt) [°F]</u>
Cu/Screen/H ₂ O	1300	2000	2.1	1.8
Al/Axial Grooves/FC-75	115	200	21.4	18.0
Al/Micro Scratch/FC-75	500	700	5.9	5.0
Al/Screen/FC-75	400	600	7.0	5.9

In addition, wick performance studies were carried out on a straight piece of tubing with removable fittings at both ends. This device made it possible to test wick configurations before actually installing them in deliverable heat pipes.

Figure III-25 shows thermocouple locations for the test runs on all prototype heat pipes and straight test vehicles at a standard test load of 250 watts at 0, 45, and 90 degrees. Results are tabulated in Tables III-7 and III-8 where ΔT is the difference between the average evaporator temperature (T_1 through T_4) and the average condenser temperature (T_5 and T_6). All models except the screen wick worked well.

HEAT TRANSFER FROM THE CONDENSER TO THE SYSTEM THERMAL RESERVOIR

The heat removal process from the solar collector utilizing a heat pipe absorber occurs through the condenser area, which is directly immersed into the heat transfer fluid. Figure III-26 shows schematically a typical configuration for a single tube. The manifold, which carries the heat transfer fluid, is a pipe with an inside diameter slightly larger than the exposed condenser length.

Below we calculate the magnitude of free convection and forced heat removal from such a straight cylindrical condenser immersed in a cross flow. The following parameters were utilized:

- . Condenser length: 0.5'
- . Condenser diameter: 0.0817'
- . Manifold diameter: 0.5'
- . Heat transfer fluid: water
- . Flow rate: 1 lb/min

THERMOCOUPLE LOCATIONS FOR HEAT PIPE TESTS

FIGURE III - 25

TABLE III-7
Wick Test Vehicles

Angle	Evaporator				Condenser		$\Delta T(K)$
	T ₁ (°C)	T ₂ (°C)	T ₃ (°C)	T ₄ (°C)	T ₅ (°C)	T ₆ (°C)	
TEST VEHICLE - Threaded Wall - Cu/H ₂ O							
0°	114	113	114	114	112	110	2.8
45°	105	115	116	117	117	113	1.0*
90°	103	111	114	113	113	110	1.2*
TEST VEHICLE - Screen (Over Threaded Wall) - Cu/H ₂ O							
0°	106	107	105	107	104	104	2.3
45°	105	108	111	109	109	106	1.8*
90°	100	117	117	117	117	113	2.0*
90°	112	117	118	118	117	114	2.2*(A)
TEST VEHICLE - Extruded Groove - (Al/(CH ₃) ₂ CO)							
0°	110	112	111	108	108	108	2.3
45°	112	110	110	110	110	110	0.5
90°	109	108	108	108	108	108	0.3
TEST VEHICLE - Extruded Groove - (Al/FC-75)							
0°	122	123	123	117	116	115	5.8
45°	114	114	113	113	111	112	2.0
90°	110	108	105	106	105	104	2.8

* Thermocouple #1 Omitted

(A) Reading taken after 60 minutes rather than 30.

TABLE III-8
Delivered Heat Pipes

Angle	Evaporator				Condenser		
	T ₁ (°C)	T ₂ (°C)	T ₃ (°C)	T ₄ (°C)	T ₅ (°C)	T ₆ (°C)	ΔT (K)
PIPE #1: Screen Wick: Cu/ H ₂ O							
0°	103	103	103	103	99	95	6.0
45°	105	109	113	105	105	103	4.0
90°	99	105	110	101	99	97	5.8
PIPE #2: Threaded Wall: Cu/H ₂ O							
0°	117	117	117	117	116	115	1.5
45°	112	119	119	118	118	118	0.7*
90°	108	118	118	116	116	116	1.3*
PIPE #3: Microscratch: Cu/H ₂ O							
0°	110	122	122	123	123	122	---*
45°	110	110	109	110	109	108	1.3
90°	125	125	124	125	125	124	0.3

* - Thermocouple #1 omitted

FIGURE III - 26

SCHEMATIC ILLUSTRATIONS OF HEAT REMOVAL FROM ADVANCED COLLECTOR.

- . Heat pipe operating temperature: 300°F
- . Manifold inlet temperature: 220°F
- . Intermediate fluid temperature: 260°F

Free Convection from a Condenser in Horizontal and Vertical Position

Taking water properties at the intermediate temperature 260°F, we have for the horizontal case:

- . Prandtl Number: $Pr = 1.39$

$$. \text{ Grashof Number } G_{r_D} = \left(\frac{g\beta\rho^2}{\mu^2} \right) L^3 \Delta T = 2.51 \times 10^9 L^3 \Delta T \quad (\text{III-60})$$

For a condenser of 25 mm diameter, with a ΔT of 80°F, we get

$$G_{r_D} = 1.11 \times 10^8$$

To calculate the average heat transfer coefficient from single pipes, the following experimentally verified formula for the Nusselt number can be utilized.¹⁰

$$\begin{aligned} \bar{N}_{u_D} &= 0.53 (G_{r_D} \cdot Pr)^{1/4} \\ &= 59.1 \end{aligned} \quad (\text{III-61})$$

The associated heat transfer coefficient is then calculated from:

$$\bar{h}_c = \frac{k}{D} \cdot \bar{N}_{u_D} \quad (\text{III-62})$$

where k is the thermal conductivity of water at 260°F and D is the diameter of the condenser in feet. We get:

$$\bar{h}_c = \frac{0.396}{0.0833} \times 59.1 = 281 \text{ BTU/hr ft}^2 \text{ } ^\circ\text{F}$$

With the available surface condenser area of $A_{\text{cond}} = 0.109 \text{ ft}^2$ and a ΔT of 80°F , the heat transferable at the condenser wall will be

$$Q = \bar{h}_c A_{\text{cond}} \Delta T \quad (\text{III-63})$$

$$= 2452 \text{ BTU/hr or } 717 \text{ watts}$$

Thus the design is more than adequate to transfer the amount of power of ca. 400 BTU/hr absorbed by 1.34 sq. ft. of solar collector at a radiation level of 300 BTU/hr ft^2 . The same configuration can be analyzed for a vertically oriented heat pipe. In this case:

$$G_{r_D} = 1.45 \times 10^{10}$$

and

$$G_{r_D} \cdot P_r = 2.03 \times 10^{10}$$

as the characteristic dimension is the length, rather than the diameter. From Kreith¹¹

$$\bar{N}_{u_D} = 0.0210 (G_r P_r)^{2/5} \quad (\text{III-64})$$

$$\bar{N}_{u_D} = 279$$

Then

$$\bar{h}_c = \frac{0.396 \times 12 \times 279}{5} = 265 \text{ BTU/hr ft}^2 \text{ } ^\circ\text{F}$$

an answer not much different in magnitude from that obtained in the horizontal case, and so the same conclusions apply.

Forced Convection from a Horizontal Condenser in Cross Flow

A typical case in a solar application would involve a water flow of 1 lb/min over the condenser. Let the diameter of the fluid carrying manifold in which the condenser is imbedded be 5 inches. The Reynolds number for this system can be calculated from

$$Re_D = \frac{VD}{\nu} \quad (\text{III-65})$$

where V = free stream velocity (ft/min)

ζ = free stream density (lb/ft³)

ν = kinematic viscosity (ft²/min)

Thus

$$Re_D = 67.15$$

According to Whitaker,¹² the average heat transfer coefficient in cross flow over a single tube can be determined from the relation

$$h_c = \frac{k}{D} (0.4 Re_D^{0.5} + 0.06 Re_D^{0.67}) Pr^{0.4} (\mu_c/\mu_\infty) \quad (\text{III-66})$$

for $1.0 > Re_D > 10^5$ and $0.7 > Pr > 300$.

All physical properties in this expression should be evaluated at the free stream temperature (260°F) except μ_c , which is the viscosity at the condenser surface temperature T_c .

For $k = 0.396$ BTU/hr ft at 260°F and

$\mu_c = 0.126 \times 10^{-3}$ lb/ft sec at 300°F, we obtain

$$\bar{h}_c = 22.2 \text{ BTU/hr ft}^2$$

or

$$Q \approx 194 \text{ BTU/hr or } 57 \text{ watts}$$

Thus, from the point of view of heat transfer, the effect of forced heat removal at standard fluid flow rates is not significant.

Conclusions

Data presented in Table III-8 demonstrates that the temperature drop ΔT caused by the heat pipe is of the same order of magnitude as temperature gradients found in conventional collectors between the surface of the absorber and the heat transfer fluid. In the case of the microscratch wick/water configuration, no detectable ΔT at 0° angle was measured. Since the test conditions (250 watts at 300°F) exceed the requirements of solar applications, it is safe to assume that there are no difficulties in building heat pipe collectors to meet the operating requirements for solar cooling and heating. The problems remaining are those of failure mode in operation and cost.

MIRROR AND COATINGS

CUSP MIRROR

Fabrication of the cusp mirror is a critical task in the construction of the advanced collector. Absolute reflectance, dimensional tolerances, mechanical rigidity and stable mounting must all be considered. Therefore, the selection of available materials and fabrication methods has been done with great care.

Materials

Freshly deposited silver on glass has a higher reflectance in the visible and IR regions of the spectrum than any other single metal on glass. See Fig. III-27. In evacuated tubular collectors, the silver sulfide tarnish normally produced by the atmosphere and associated with silver coatings¹³ is prevented by the vacuum environment. As a result, the substrate optical surface and dimensional control become the significant factors in producing a highly efficient mirror system. Fabricating the proper surface for depositing metallic silver that will perform satisfactorily at 300°C in a vacuum presents numerous technical challenges.

Firstly, unless an optically correct curved surface is obtained, light will be scattered and misdirected away from the absorber. Total reflectance must be high and of the total reflectance, 95% or more should be specular. Secondly, for long vacuum life, the use of all organic materials and adhesives must be avoided.

Several methods for producing a glass mirror were considered: but rejected because of time limitations.

FIGURE III-27
REFLECTANCE OF EVAPORATED METALS AND ALUMINUM LIGHTING SHEET

(1) The 100 mm diameter glass envelope could be shaped on one side to conform to the cusp curve. This could be done when the tube was formed or in a subsequent finishing operation. Considerable time and effort would be involved in perfecting this method due to stresses involved in forming and in evacuation, the work involved in developing the forming method, and in resolving the end closure problems.

(2) Glass could be sagged to the cusp shape. A sharp cusp point could not be readily achieved by this method and when sagging against a mold, surface irregularities occur that detrimentally affect mirror specularity. Mounting a glass mirror within the 100 mm envelope also requires fixturing that will positively align the glass to the absorber pipe and prevent slippage, breakage, and abrasion between the envelope and mirror.

(3) Glass micro sheet less than 5 mils thick can be silvered and shaped in two halves to the cusp curve. In this case, proper orientation and support must be provided by brackets or a cradle. Normally, micro sheet is sold in squares 6" X 6" up to 13" X 14". Special arrangements for long strips could be made with the factory at higher costs.

One of the above methods will most likely be developed, if the cusp collector proves superior. However, the cost and time involved in perfecting the above approaches, when weighed against the performance achieved and the schedule for the current program, forced consideration of a more immediate, although less perfect, solution: metal mirrors.

High quality aluminum lighting sheet with total reflectivity of 80 - 85% in the visible region is commercially available. With its ease of forming to the required cusp shape, aluminum lighting sheet was chosen as the initial cusp reflector material. Figure III-27 shows the reflectance of evaporated metals compared with three commercially available lighting sheet materials.^{14,15}

Alcoa, Kingston Industries, Reynolds, Conalco, and Kaiser Aluminum Materials were investigated. Several suppliers recommended starting with mill finished, one-side-bright alloys and subjecting them to a brightening dip of phosphoric-nitric acid for improved reflectance, but numerous experiments did not produce materials equal to or better than the anodized lighting sheet commercially available.

Most aluminum lighting sheet is sold with an anodized surface to protect against natural oxidation and weathering and subsequent loss of the highly reflective surface. The presence of this protective anodic film presents an interesting situation when determining reflectance for the different wavelengths of the solar spectrum.^{15,16}

A Beckman DK-2 spectrophotometer was used to determine total reflectance curves compared to evaporated aluminum on glass for four of the best available materials. Due to the interference effect of the anodic film, periodically oscillating reflectance as a function of wavelength was obtained. Removal of the anodized surface in a phosphoric-chromic acid bath improved the reflectivity significantly and resulted in no oscillations throughout the visible range of the spectrum. Total spectral reflectance of four materials, weighted by the Mass 2 solar spectrum, are summarized in Table III-9.

TABLE III-9
Total Reflectance* Data for Various
Commercial Bright Aluminum Sheets

λ [nm]	Kingston C4/20	Alcoa 5457 ACA	Conalco 5657-B	Alzak H-18	Reference** Mirror
340	89.74%	90.02%	-	90.82%	91.6%
360	90.38	90.38	-	90.66	91.5
380	90.67	90.48	89.82%	90.29	91.7
400	90.47	90.19	89.81	90.44	91.6
450	90.84	90.17	89.89	90.55	91.5
500	90.64	90.45	90.07	90.54	91.4
550	90.14	89.85	89.76	90.14	91.0
600	89.74	89.17	89.17	89.46	90.5
650	89.04	88.95	88.76	88.95	89.8
700	88.34	87.97	87.69	88.16	89.0
750	86.69	86.50	86.32	86.87	87.8
800	84.74	85.69	85.26	84.40	86.3
850	85.75	86.73	86.03	85.59	87.7
900	89.41	89.50	89.05	89.23	90.6
950	90.76	90.67	90.30	90.76	91.5
1000	93.42	93.33	92.85	93.14	93.8
1500	96.90	97.20	96.61	97.00	97.2
2000	97.30	97.00	96.90	97.00	97.3
Solar***	90.82%	90.77%	90.44%	90.74%	91.6%

*Samples without anodic film overcoat

**Evaporated aluminum on glass - SiO₂ overcoated

***Air Mass 2 total reflectance

The optical design of the cusp mirror is such that a highly specular mirror surface is required for directing the photons onto the absorber. A highly diffuse component is undesirable in that considerable energy is lost by being scattered away from the absorber and back into the atmosphere. Specular reflection tests are part of an intensive program currently underway for total characterization of candidates for cusp mirror materials.

Most rays strike the mirror at an angle of incidence other than normal as shown by Fig. III-28.

In order to obtain reflectance data and determine angular dependence for the candidate materials, an experimental setup consisting of a goniometer, modulated tungsten-krypton collimated light source, sample holder, and photocell detector was built. The comparison of the light source with the solar spectrum (Air Mass 2) is shown in Fig. III-29. Samples were placed in the holder and percent reflectance versus angle of incidence measured and recorded. The results for stripped Alzak H-18 are shown in Fig. III-30.

Materials thickness was also considered, as this parameter is important from the cost and handling aspects. However, cusp mirror weight has only a slight effect on the overall weight of the finished collector. Since the mechanical requirements are only that the cusp mirror be self supporting in holding its shape, materials as thin as 0.012" thick was successfully shaped. However, for the initial collectors, the more common thickness of 0.020" was used to minimize fabrication and handling problems.

FIGURE III - 28

CROSS SECTION OF ADVANCED COLLECTOR SHOWING
PHYSICAL DIMENSIONS

FIGURE III - 29

SIMULATION OF SOLAR SPECTRUM FOR ANGULAR REFLECTANCE MEASUREMENTS

FIGURE III - 30

ANGULAR DEPENDENCE OF REFLECTANCE FOR STRIPPED ALZAK H-18

Forming

Two methods of forming the compound curve of the 4 ft long cusp mirror were selected in order to meet the performance and timing requirements for the program. Both were pursued simultaneously using computer printouts of the theoretical curve for correct optical coupling to the 95.2 mm I.D. of the glass envelope and the 25 mm O.D. of the absorber. X and Y coordinates of the reflecting surface were used in preparing tooling, gage templates, and form blocks to assure compliance as closely as possible with the theoretical design. Although the Monte Carlo ray tracing studies of required mirror accuracy are incomplete at this time, an initial target of ± 0.010 " variation from curve design has been assumed.

A large press capable of pressing 4 ft long cusp mirrors was located and made available for experimental runs. Due to the nature of the process, no expensive molds are required. Metal sheets up to 0.080" thick are readily formed to the shape of the form block by pressing them between the form block and a large rubber bed. This slow speed operation does not lend itself to low cost, high volume production, but is ideally suited for accurately shaping smooth curves in experimental work. Different materials with different hardness values exhibit different degrees of spring back after pressing. In attempts to minimize this, several form blocks with variation in curvature were designed and fabricated. Annealing schedules of 5 and 10 minutes at 400°C were also used to soften the reflector sheets to a point where satisfactory mirrors could be produced. Work on this process

was stopped when it became evident that a satisfactory product could be obtained by roll forming.

Twelve companies were contacted regarding roll forming the cusp mirror and a contract was negotiated for material and experimental tooling for 100 piece sample quantities to our specifications. Roll forming is a highly automated, high speed process where coiled material, slit to the proper width, is fed continuously through a series of rolls where each roll set progressively shapes the blank to the desired contour. The cusp mirror shape is ideally suited for this production method. Initial samples for approval indicated the need for some modification to the tooling and this is now underway to improve the free standing shape of the mirror. Kinglux C 4/20 NA material has been used so far in the process.

Because of the critical alignment problem between the cusp mirror and absorber, a baffle shown in Fig. III-31 was designed for each end of the mirror. This baffle supports the mirror and orients it positively to the absorber. It is fastened to the mirror by slots and tabs designed so that they do not contact the glass surface. Positive optical control is maintained while conductive losses are kept to a minimum. Shadow losses are also minimized since the baffles are cut away at the top and are made of highly reflective aluminum. Shade losses at one end are partially recovered by reflective gains at the other.

CUSP MIRROR BAFFLE

FIGURE III - 31

APPLICATION OF SELECTIVE ABSORBERS

Most solar collector devices which have been installed up to the present time utilize a flat black absorber surface to maximize energy absorption. These surfaces are also good radiators of thermal energy. Hence losses are very high, increasing as the fourth power of the absorber temperature. As a consequence, the high temperature efficiency of these thermal collectors is very low, and the maximum temperature of operation is limited.

If the advantage of high efficiency vacuum insulation in a tubular evacuated collector is to be maintained, a selective absorber surface must be incorporated into the device. A survey of various reported coating methods was undertaken to determine which one might be applicable to our specific requirements.

Survey of Selective Absorber Materials

The traditional optical method for preparing a selective absorber, or dark mirror, is to deposit a multilayer thin film coating. Film thickness is such that destructive interference occurs for radiant energy in the solar spectrum, while the surface appears highly reflective to the longer wavelength IR energy. Often the cover layer can be formed of a material having a relatively low index of refraction, reducing surface reflection and further increasing solar absorption capability. Extreme precision is required in controlling thickness of the various layers; this is accomplished using vacuum evaporation equipment with an associated film thickness measurement capability. The process is well known, and small optical type surfaces are readily produced. Very large, specialized equipment would be required to produce surfaces large enough for solar thermal collectors, and the estimated cost is high.

A number of attempts have been made to develop lower cost selective absorbers. Mattox and Sowell review developments in this area.¹⁷ In 1955, H. Tabor recognized the requirements for a single layer selective coating and suggested means for accomplishing this.¹⁸ Recent publications by Peterson and Ramsey,¹⁹ and by McDonald²⁰ outline further technology for single layer selective coatings.

Single layer coatings depend upon several phenomena for their performance. Semiconductors have an energy gap which makes them natural absorbers of short wavelength radiation and transmitters of longer wavelength energy. This transition does not usually occur near the desired wavelength. In addition, these materials have high indices of refraction, resulting in high reflectivity at their interface with air. However, certain oxides and sulfides were found to have useful properties in this work, and development continued.

Very thin copper oxide films can be readily formed by chemical reaction, and these were shown to have encouraging selective properties. Experimental work indicates that the oxide film must be thin enough to appear transparent to long wavelength radiation, yet thick enough to present a dark absorbing surface to shorter wavelength solar radiation. The transition zone can be varied with oxide thickness, although the semiconducting property of copper oxide may play a part in establishing this wavelength. IR reflectivity of the coating is found to depend very heavily on the metal substrate surface. It must appear smooth and reflective to wavelengths greater than two microns.

Various additional oxide and/or sulfide coatings can be applied by electroplating. In all cases, the coating composition and thickness determine the absorptance of solar energy, while the coating thickness and substrate surface affect IR reflectance. When these parameters are optimized, it is possible to produce selective coatings with characteristics approaching those obtained with an evaporated multilayer dark mirror. The processes involved do tend to be more adaptable to large scale production and manufacturing costs should be considerably less than for evaporated films.

Specific Requirements for Selective Coatings

Solar absorptance (α) is a measure of the ability of a surface to absorb solar energy. The efficiency of a solar collector is very much dependent upon this value; a target value of $\alpha \geq 0.90$ has been established for this program. Another major factor in maintaining good collector efficiency is the tendency of the absorbing surface to re-radiate thermal energy. To limit this radiation, the IR emissivity (ϵ) must be kept as low as possible. The target value for this program is $\epsilon \leq 0.10$. A selectivity ratio (α/ϵ) is often given to characterize absorber surfaces. In actual practice, this ratio is so drastically affected by small changes in ϵ that it does not always give a clear indication of absorber utility.

During use, the selective coating material is subjected to high temperatures in high vacuum; it is most important that the material used be thermally stable and not degrade under use conditions. Outgassing during operation would rapidly destroy the enclosure vacuum with a major loss of collector efficiency.

Choice of Absorber Coatings

A continuing survey of materials and processes for producing selective absorber coatings is being made. Corning Glass Works (CGW) has obtained samples from many commercial vendors; other candidate coatings have been prepared in this laboratory. In an attempt to focus quickly on a useful coating, we concentrated on those candidates which appeared capable of yielding an α/ϵ ratio of at least 9 and which could be produced in the long run for minimum cost. The candidates showing the most promise were Black MagicTM (a copper oxide film), and "Black Chrome" (a chromium oxide surface). Many coatings submitted by outside vendors were found to be very poor. The impression is that the production of selective absorber coatings is not well understood.

Copper Oxide Process

Black MagicTM is a trade name of the Mitchel-Bradford Co. An apparently very similar coating is called Ebonol CTM, manufactured by Enthone, Inc. These strongly alkaline solutions are applied to a copper surface causing rapid oxidation which is predominately CuO, although Cu₂O is also present in small amounts. The surface produced is rough on a micro scale, and it is often used in thicker films as an adhesion layer for further coats. Rubbing the surface with a soft cloth or paper produces a very obvious burnishing effect as well as some degradation of solar absorptance.

The chemical reaction occurs so rapidly with full strength solution, that immersion times become very short and difficult to control. Dilution to one-half the manufacturer's recommended strength proved most practical.

A typical coating process for laboratory use would include the following steps:

(1) Solution preparation

(a) M-16 alkali cleaner

1/2 lb/1 gal H_2O @ 180°F

(b) Bright dip solution

5 - 10% H_2SO_4 in H_2O @ room temp.

(c) Black MagicTM solution

1 lb/1 gal H_2O @195 - 200°F

(2) Process steps

(a) 2 min. in alkali cleaner

(b) Water rinse

(c) 1 min. in acid bright dip

(d) Thorough rinse in running water

(e) 1 - 2 min. in Black MagicTM. Use no mechanical agitation in this bath.

(f) Water rinse and blot dry.

The M-16 cleaner and Black MagicTM baths should be kept covered as much as possible to prevent water evaporation. As the level drops, water should be added to retain the initial volume.

Eventually, the bath starts to become depleted, and reaction times become longer. Additional Black MagicTM can be carefully added to the solution. A boiling point test for concentration is given by the manufacturer, but this seems to lack sensitivity. A methodical log of process conditions probably provides the best method of monitoring bath condition. IR emissivity of the resulting film is a satisfactory indicator of film thickness within the normal operating range.

"Black Chrome" Process

The use of the commercially available black chrome decorative plating finish as a selective absorber was suggested by MacDonald of NASA Lewis.²¹ For this application, plating thickness is less than for decorative finishes.

Most of the early plating work was done by plating black chrome over an electroplated nickel layer. This proves necessary when steel substrates are used to prevent chromic acid attack. Samples prepared by the Harshaw Chemical Co. and tested in our laboratory indicated that better results could be obtained on copper without the intermediate nickel layer. This had the added advantage of eliminating the longest plating operation, thus significantly reducing process cost.

Harshaw provides a very complete instruction brochure (CRX-0270) on the preparation and use of the ChromonyxTM black chromium plating process. A premixed bath can be obtained from the company, and use of this premixed bath is assumed in the following process.

A 500 ml beaker, containing about 400 ml of plating solution was used. The anode was of low carbon steel, and 2 - 3 times the cathode area. In these experiments, a 2" X 4" section of expanded metal was shaped to the beaker wall and used as the plating anode.

The test samples were either 1" square, or 1" X 2", giving cathode areas of 2 - 4 in². Plating current was set at 300 amperes/ft², or 4 amperes for the 1" X 1" substrate. Recommended plating current is 200 - 300 amperes/ft².

Substrates are cleaned and bright dipped as in the Black MagicTM process. The bright dipping is best done in the plating rack to avoid the necessity of handling the substrates after final cleaning.

Immerse in the plating bath for 45 - 90 seconds with current adjusted to the proper level. In laboratory experiments, a 60 second plating time yielded excellent results, with the selectivity ratio reaching 20. Variations in plating bath conditions, and in plating current density require that process calibration runs be made with substrates of the desired size and shape. Some modification of process conditions will usually be needed to obtain a uniform coating having the desired properties. As soon as plating is completed, the substrates are removed, rinsed thoroughly in water, and blotted dry as before.

The plating bath is operated at room temperature, and it should not exceed 85°F. A large amount of heat is generated by the high plating current and it is possible that artificial cooling would be required for continuous operation of the bath.

Effect of Substrate Material and Surface Conditions on Selective Properties

The IR emissivity of the oxide type selective absorber can be no lower than that of the substrate surface on which it is formed. Thus it is important that this characteristic of the substrate be established prior to the coating process. Tabulated values of the reflectance of freshly evaporated metal films are available in The American Institute of Physics Handbook. Copper is seen to be a somewhat selective surface itself, with reflectance rising rapidly in the wavelength range of 0.4 to 0.6 microns. Its reflectance at 5 microns is 98.3%. Aluminum is a good reflector over the entire visible and IR range except for a dip in the near IR portion of the spectrum. Its IR reflectance is about 0.6% less than that of copper at 5 microns.

Reflectance of bulk metal surfaces will be somewhat less than that of freshly evaporated films on glass. Reflectance will also depend to a considerable extent on the surface finish and cleanliness of the sample material.

The best copper surfaces measured have approached an IR reflectance value of 0.98. Polished aluminum can reach about 0.97, or just slightly less than copper. Experience has indicated that copper sheets received with thicknesses of 0.020" or less usually have good surface finishes. When carefully cleaned, satisfactory IR reflectance has been obtained. Thicker copper often has a more coarse surface, or a larger number of defects. This requires mechanical buffing to achieve the desired properties. The same observation is probably true of aluminum sheets as well, but the greater range of hardness and additives make generalization more difficult.

Surfaces can be improved by using buffing compounds on a cloth wheel. This is necessary if machining or forming operations have been performed on the material.

Visual appearance can be deceptive. IR reflectance measurements must be made to properly characterize the substrate surface. For example, attempts were made to improve copper reflectance by electro-polishing the surface. This process did not materially improve a poor surface, but visible reflectance of a good surface was enhanced. However, measurements made over the entire IR radiation range indicate that very little improvement has been made in the overall IR reflectance. It was concluded that mechanical buffing is the most simple and direct means of achieving necessary reflectance properties.

ABSORBER COATING PROPERTIES AND MEASUREMENT TECHNIQUES

Solar Absorptance

Solar absorptance (α) is measured as solar reflectance (ρ). It is assumed that the solar absorber has zero transmittance; therefore the equation

$$\alpha = 1 - \rho \quad (\text{III-71})$$

can be used in determining solar absorptance. In every case, solar absorptance must be determined taking into account the spectral distribution of solar energy at the Earth's surface. Two approaches to this measurement have been used in Corning Glass Works.

(1) The Technical Services Optical Properties Laboratory of CGW uses a Beckman DK-2 Spectrophotometer for this work. A radiant energy source is scanned from 0.3 microns wavelength to 2.5 microns. The reflected radiation from the sample at each wavelength is collected in a sphere and measured photoelectrically. Thus the instrument responds to total reflected radiation while the light energy is supplied normal to the test surface. Results of the measurement are presented as a plotted spectral reflectance curve for the test surface. For each test, a barium sulfate on glass standard reflector is also recorded.

Reference to a solar spectrum distribution chart for Air Mass 2 characterizes the energy available at the Earth's surface. This is normalized, and four reference wavelengths are chosen to numerically approximate the energy distribution. An integration around these points establishes weighting factors for the curve.

<u>Wavelength</u>	<u>Factor</u>
0.5 μ	13
0.7 μ	10
1.2 μ	5
1.5 μ	2

Using this information, the measured reflectance at each of the four wavelengths is multiplied by the appropriate weighting factor - the sum of these values divided by 30 is taken as the solar reflectance.

(2) The Gier-Dunkle Instrument Co. produces the MS-251 solar reflectometer. In the early phase of our program, samples were sent to TRW, Inc. for measurement with this instrument. Later, an MS-251 was obtained here and used for direct sample measurement.

Operation of the instrument is much simplified since solar reflectance is read directly on a digital meter. An approximate solar spectrum is established by use of an unfiltered Xenon arc lamp as the energy source. Unfortunately, this source is a better solar match in space than at the Earth's surface, and the resultant spectral differences have some effect on the reflectance value. In spite of this, the MS-251 is valuable in obtaining a fast indication of solar absorptance.

IR Emittance

Emittance of a metal is related to its IR reflectance which is measured at wavelengths generally exceeding 2 microns. Since this is entirely outside of the visible spectrum, visual examination of a sample is of little use in determining its IR reflectance. The ability of such surfaces to emit thermal radiation is the property of interest. Assuming an opaque material, the surface

emittance (ϵ) can be taken as:

$$\epsilon = 1 - \rho \quad (\text{III-72})$$

Again two approaches are available for measuring ϵ .

(1) The Technical Services Optical Properties Laboratory of CGW uses a Perkin-Elmer 221 IR Spectrophotometer for measuring ϵ . Its radiant energy source is scanned from 2 microns wavelength to 20 microns covering the range of IR radiation from black bodies in the 100 - 200°C (212 - 392°F) temperature range. Unfortunately, the detector in this instrument responds only to specular reflection; energy reflected in a diffuse pattern is largely undetected. Where very thin selective films were applied to smooth substrate surfaces, the resultant error did not appear large. However, as the range of test surfaces increased, many of the measured results became suspect. Again, data from the spectrophotometric curve was converted to a single emittance value by using a weighting system based upon a normalized radiation curve for a 135°C (275°F) black body.

(2) The Gier-Dunkle DB-100 offers a direct method of measuring IR reflectance. Essentially the sample surface is subjected alternately to diffuse radiation from two chambers heated to different temperatures. Reflected energy from the surface is measured by a vacuum thermocouple; the difference temperature detected from the two sources is calibrated and read out as IR reflectance. Instrument accuracy is given as $\pm 3\%$ of full scale when measuring selective surfaces. Reproducibility is $\pm 0.2\%$ of full scale.

Prior to receipt of the DB-100 instrument in Corning, samples were sent to TRW for measurement on a similar instrument. They

reported excellent agreement between values obtained on the DB-100 and values determined using emittance values from reference sources. Samples returned from TRW measurements were remeasured in Corning after receipt of our DB-100; agreement was very good, well within the accuracy specification of the instrument.

COMPARISON OF RESULTS BY DIFFERENT MEASUREMENT TECHNIQUES

Solar Absorptance

Table III-10 contains values of solar absorptance measured by the available methods. Single sample comparisons are hazardous because there is no assurance that both measurements were made at exactly the same spot. Taken over a larger number of samples though, the average difference indicates that the Gier-Dunkle value is 3.3 percentage units lower than the weighted Beckman spectrophotometer values. This difference was also noted at TRW, and they estimate a value of -3.5% as compared with their standard instrumentation. The probable cause of this difference is the failure of the Xenon arc spectral distribution to match solar distribution at Air Mass 2. The Gier-Dunkle equipment has proven capable of detecting differences between sample absorptances when proper measurement care is taken. Absolute accuracy has not been established, nor does the manufacturer specify this value. While the 3.3% correction appears reasonable for values of absorptance of 0.75, there is not enough data to extend this relationship to lower values. Samples were measured at TRW for absorptance several months before receipt of our MS-251. The somewhat larger than desired discrepancy between the measured values includes 8 months of sample handling, test, and storage. It is not possible to accurately specify the difference in readings obtained from the two instruments.

TABLE III-10

Comparison of Solar Absorptance Measurements

<u>Sample Number</u>	<u>Coating</u>	<u>Corning Gier-Dunkle MS-251</u>	<u>TRW Gier-Dunkle MS-251</u>	<u>Corning Wgt'd Beckman DK-2</u>	<u>NASA Wgt'd Cary 14</u>	<u>TRW-Wgt'd Spectral Reflectance</u>
OCLI	Dark Mirror		0.86	0.94		0.90
SE-123	Black Magic TM	0.81	0.79	0.86		
SE-133-15	Black Magic TM	0.79	0.75	0.79		
SE-133-16	Black Magic TM	0.79	0.75	0.80		
SE-133-18	Black Magic TM	0.81	0.78	0.81		
SE-133-19	Black Magic TM	0.83	0.79	0.86		
SE-133-20	Black Magic TM		0.76			0.79
SE-165-1	Black Magic TM	0.85		0.88		
SE-165-2	Black Magic TM	0.89		0.91		
SE-140-1	"Black Chrome"		0.88	0.94		
SE-140-2	"Black Chrome"		0.89	0.96		
SE-140-3	"Black Chrome"		0.89	0.96		
SE-163-A1	"Black Chrome"	0.87		0.92		
SE-163-B1	"Black Chrome"	0.91		0.95		
SE-163-C1	"Black Chrome"	0.94		0.97		
SE-164	"Black Chrome"	0.92		0.96		
SE-142	"Black Chrome"		0.74		0.73	
SE-206-1	"Black Chrome"	0.93		0.97		
SE-206-2	"Black Chrome"	0.92		0.94		
SE-206-3	"Black Chrome"	0.93		0.96		

The single comparison between the TRW Gier-Dunkle value and a NASA value is encouraging but not conclusive. NASA's value was obtained by computer weighting of a curve derived from the Cary-14 Spectrophotometer.

IR Emissivity

Table III-11 reports data comparing emissivity values determined by various methods. Experience has shown that measurements by the PE-221 are unreliable because of its specular response. In some cases, thin selective coatings on smooth, flat surfaces do yield comparable results; rough surfaces appear high in emissivity with this instrument. Concave or convex surfaces always produce erroneous results.

Comparative readings between the two DB-100 Instruments have been good, well within the claimed accuracy. TRW compared their results with available literature information, and the results were again good. On this basis, we have adopted the Gier-Dunkle DB-100 values as the most reliable indicator of IR emissivity. Instrument repeatability is indicated by measurements taken two weeks apart on five samples; agreement is excellent.

Sample 142 was also measured at NASA Lewis, using a Willey 318 Spectrophotometer capable of measuring total reflectance. Emissivity was calculated using a weighted analysis of the curve. While the reported value looks low, this sample was an early, very light black chrome surface, and the uniformity was not good.

TABLE III-11

Comparison of IR Emittance Measurements

<u>Sample Number</u>	<u>Coating</u>	<u>Corning Gier-Dunkle DB-100</u>	<u>Corning* Gier-Dunkle DB-100</u>	<u>TRW Gier-Dunkle DB-100</u>	<u>Corning Wgt'd PE-221</u>	<u>NASA Wgt'd Willey 318</u>
OCLI	Dark Mirror	0.03		0.02	0.04	
SE-123	Black Magic TM	0.07		0.05	0.19	
SE-133-15	Black Magic TM	0.03		0.02	0.09	
SE-133-16	Black Magic TM	0.05		0.03	0.23	
SE-133-18	Black Magic TM	0.06		0.04	0.20	
SE-133-19	Black Magic TM	0.09		0.07	0.25	
SE-144	Black Magic TM	0.08			0.17	
SE-148-1	Black Magic TM	0.07	0.07			
SE-148-2	Black Magic TM	0.10	0.10			
SE-148-3	Black Magic TM	0.07	0.07			
SE-148-4	Black Magic TM	0.06	0.07			
SE-148-5	Black Magic TM	0.08	0.08			
SE-140-1	"Black Chrome"	0.27		0.26		
SE-140-2	"Black Chrome"	0.33		0.34		
SE-140-3	"Black Chrome"	0.40		0.41		
SE-140-4	"Black Chrome"	0.21		0.16		
SE-140-5	"Black Chrome"	0.23		0.26		
SE-140-5	"Black Chrome"	0.23		0.26		
SE-142	"Black Chrome"	0.10		0.08		0.05

*Measurements in this column made 15 days after original measurements.

The checked emissivity values were measured eight months after the TRW measurements; as noted previously, the increase in emissivity probably results from sample handling.

Summary of Optical/Thermal Characteristics

Table III-12 lists typical values of solar absorptance and IR emittance for various types of selective surfaces. Most of these materials can be adjusted over a range of values by variations in coating thickness, substrate surface, and substrate composition.

The two coatings of most immediate interest, Black MagicTM, and "Black Chrome" share the characteristic that as coating thickness is increased to improve absorptance, IR emissivity also increases. This is probably true of at least all single layer selective coatings. It means that a practical compromise must be attained between the two values, the target depending upon collector design and operating conditions. The reported values come from attempts to maximize solar absorptance while keeping the IR emissivity at 10% or less.

For the single layer films; "Black Chrome", when properly applied, is capable of giving the best results. Difficulty has been encountered, however, in applying a uniform coating of black chrome on a large flat surface. In many cases, the resulting non-uniformity has led to disappointing results. Good progress has been made in controlling this process but more work remains to be done. Very preliminary tests in plating 25 mm dia. cylindrical samples have indicated that the uniformity problem will be far less severe with this type of substrate.

TABLE III-12

Typical Absorptance and Emissivity Values

<u>Sample Number</u>	<u>Source</u>	<u>Substrate</u>	<u>Coating</u>	<u>Absorptance α</u>	<u>Emissivity ϵ</u>	<u>α/ϵ</u>
-	OCLI	0.003" Al	Multilayer Dark Mirror	0.89	0.02	44.5
SE-133-15	CGW	0.005" Cu OFHC Electropolished	Black Magic TM (20% normal thickness)	0.78	0.02	39.0
SE-152	CGW	0.017" Cu Good surface	Black Magic TM Std process	0.85	0.05	17.0
SE-165-7	CGW	0.040" Cu Roll bond	Black Magic TM Std process	0.87	0.09	9.7
SE-161-3	Harshaw	0.016" Cu Mill finish	"Black Chrome"	0.97	0.04	24.3
SE-206-2	CGW	0.020" Cu OFHC Mill finish	"Black Chrome"	0.95	0.04	23.8
SE-163-B1	Olympic	0.016" H ₂ Fired Cu Mill finish	"Black Chrome"	0.94	0.13	7.2
SE-174-6	Harshaw	0.016" Cu Mill finish	"Black Chrome" (From Olympic bath)	0.96	0.05	19.2

STABILITY OF OPTICAL PROPERTIES DURING THERMAL EXPOSURE

Deterioration of selective coatings in an evacuated collector can have a twofold adverse effect on the overall thermal performance. First, the selectivity (α/ϵ) may drop due to the degradation of the absorptive film; second, gases may evolve due to the associated chemical reactions and convective losses to the environment may increase.

Data available at the present in regard to changes in vacuum as a function of temperature and duration of the test is sparse.

Of necessity, a program was started to investigate problems associated with the instability of chemical and physical properties of selective coatings in vacuum.

An evacuated environment may have a reducing effect on the absorber coating. Oxides have the tendency to be reduced slowly when heated in vacuum. It is important whether these changes occur only above some limiting temperature or occur more slowly in the range of operating temperatures.

Several short duration experiments were conducted with small samples to obtain preliminary information for further evaluation. The selectivity and appearance of samples were monitored before and after several hours of exposure to high temperatures in vacuum. Table III-13 summarizes the results of these tests.

Another essential phenomena coupled to the stability of the selective absorbers in vacuum is the rate of outgassing at elevated temperatures. Data listed in Table III-14 show the total gas evolved at bake out temperatures from 200°C to 600°C (392 to 112°F).

Black-MagicTM appears to be less effective than "Black Chrome" in general. Short term, high temperature testing is essential to

TABLE III-13

Thermal Stability of Absorber Materials

<u>Sample Number</u>	<u>Source</u>	<u>Substrate</u>	<u>Coating</u>	<u>Environment</u>	<u>Results</u>
- $\alpha = 0.89$ $\epsilon = 0.02$	OCLI	0.003" Al	Multilayer Dark mirror	2 hrs* 335°C 4 X 10 ⁻⁵ Torr	No visible change
SE-147-1 $\alpha = 0.91$ $\epsilon = 0.08$	CGW	0.017" Cu	Black Magic TM Std process	1 1/2 hrs* 275°C 4 X 10 ⁻⁵ Torr	Slight color change
SE-139-1A $\alpha = 0.94$ $\epsilon = 0.24$	S-O-H	0.115" Cu Roll bond	"Black Chrome" over Ni plate	3 hrs* 290°C 4 X 10 ⁻⁵ Torr	No visible change
SE-155-5 $\alpha = 0.95$ $\epsilon = 0.07$	Harshaw	0.016" Cu	"Black Chrome" over Ni flash coat	4 hrs* 390°C 4 X 10 ⁻⁵ Torr	Possible slight decrease in ϵ ($<0.5\%$)
- $\alpha = 0.81$ $\epsilon = 0.06$	CGW	0.005" Cu OFHC	Black Magic TM	3 hrs 300°C Std bake out 1 X 10 ⁻⁴ Torr	α decreased to 0.78 ϵ decreased to 0.03
SE-164 $\alpha = 0.91$ $\epsilon = 0.06$	Harshaw	0.016" Cu H ₂ fired	"Black Chrome"	3 hrs 400°C Std bakeout 1 X 10 ⁻⁴ Torr	α decreased to 0.89 ϵ decreased to 0.05

*Heated by Xenon lamp radiation through a glass wall.

TABLE III-14

Absorber Material Outgassing to 600°C

<u>Sample Number</u>	<u>Source</u>	<u>Substrate</u>	<u>Coating</u>	<u>Total Gas Excluding H₂O Liter-Microns/cm³</u>	<u>Temp Where Visible Change First Noted</u>
SE-151-1	Olympic	0.040" Cu Roll bond	"Black Chrome" -Early sample -Large thickness gradient	119	600°C
SE-200-11	CGW	0.005 Cu OFHC	"Black Nickel" (Lab bath)	2585	200°C
SE-147-1	CGW	0.017" Cu	Black Magic TM Std process	81	300 - 400°C
SE-147-3	Honeywell	0.012" Steel	"Black nickel"	32	200°C
SE-164	Harshaw	0.016" Cu H ₂ fired Mill finish	"Black Chrome"	163	500°C

delineating fabrication processes, because it is necessary to thoroughly bake out the entire collector assembly during the evacuation process. The absorber especially must be heated as far above the projected stagnation temperatures as possible to prevent future outgassing and loss of vacuum. Although useful data is obtained from these short term exposure experiments, they are not conclusive for longer times at lower temperatures at which the collectors will operate. Experiments to explore life time of collectors are under way.

COLLECTOR ASSEMBLY

The completion of the evacuated cusp collector, whether flow-through or heat-pipe, requires two distinct steps in the fabrication process

- (1) Structural assembly
- (2) Evacuation

Below we report the progress made in these areas.

HEAT PIPE ABSORBER

Both copper and Code 7740 glass are presently used in the construction of the advanced collector. Since the expansion of copper is so much greater than the borosilicate glass envelope, Housekeeper seals were selected for joining the metal heat pipes to the evacuated tubular envelopes.

When properly shaped to a feather edge, as shown in Fig. III-33, OFHC copper can be made to wet glass and form an elastic component that follows the expansion and contraction of glass without exceeding its stress limits.²³ A vacuum tight joint, common in the vacuum and power tube industry, can be formed in this manner.

COPPER-GLASS HOUSEKEEPER SEAL

FIGURE III-32

This glass-to-metal seal is combined with the "plug in" concept of the advanced heat pipe collector as shown in Figs. I-2, III-17, III-26, III-33, and III-34. Commercially available CGW couplings are used to join the beaded glass flange at the condenser end with the storage tank directly or via a manifold carrying the heat transfer fluid. By slipping the collectors into or out of the coupling and adjusting only one nut on the coupling, the heat pipe collector can be connected or removed at will.

The conventional glass working techniques used in sealing the Housekeeper seals require that care be taken to obtain adequate annealing of stresses in the glass without excessive oxidation of the copper. It is also important that sufficient thickness be maintained in the exposed thin sections of the Housekeeper seals to prevent collapse due to the mechanical forces resulting from the vacuum and installed heat pipe.

Areas of stress concentration introduced when fabricating the header can be readily observed by inspection with a polariscope. A fully annealed product is desirable. In no cases should stresses greater than 250 psi tension be permitted in finished pieces.

FLOW-THROUGH ABSORBER

Figure III-35 shows the design of the advanced flow-through collector to be used to compare with the heat-pipe version. As shown, the heat-transfer fluid enters one end and exits the other. This minimizes problems of air locks and simplifies the internal manifolding of the absorber pipe.

Note that a glass plug is inserted inside the absorber pipe. This effectively reduces the cross sectional area from 0.70 in².

FIGURE III - 33
HEAT PIPE DIMENSIONS AND HEADER ASSEMBLY

FIGURE III - 34

ADVANCED HEAT PIPE COLLECTOR AND "PLUG IN" COLLECTOR CONCEPT

FIGURE III - 35
ADVANCED FLOW-THROUGH COLLECTOR

to 0.12 in^2 . The resulting six-fold increase in fluid velocity adjacent to the absorber surface has a significant effect on increasing heat transfer. Absolute values depend upon the Reynolds number and resulting heat transfer coefficient.

Metal plugs are used to connect the components, i.e. the glass-to-metal seals, expansion bellows, and absorber pipe. High temperature silver solder is used throughout. Care must be taken in assembly and annealing so as not to excessively oxidize the copper parts. At this writing, three flow-through collectors have been assembled. Vacuum to 10^{-4} Torr has been obtained on one of the three but vacuum leaks developed on the other two during sealing of the glass envelope. Cooling techniques during sealing have now been designed to reduce the chance of this recurring in the future.

EVACUATION

Proper functioning of the evacuated tubular collector demands that a vacuum of 10^{-4} Torr or better be maintained over a period of years. Achieving and maintaining such a level of vacuum for long periods of time requires extreme cleanliness and good evacuation techniques. This part of the report covers the work done recently to establish a procedure for attaining a high vacuum in a large tubular solar collector. Only collectors having flat absorber plates (Fig. I-1C) have been processed so far.

Equipment

The equipment used for experimentally pumping solar collectors is shown schematically in Fig. III-36. The basic pumping system is a 6" fractionating diffusion pump with DC 705 fluid, coupled to a high conductance liquid nitrogen cold trap. Associated piping

FIGURE III - 36

PUMP SYSTEM FOR SOLAR COLLECTOR EVACUATION

and valving facilitate roughing and high vacuum pumping through either of two paths or both in parallel for increased conduction. The lower path contains a quadropole mass spectrometer used to observe the relative amounts of different gases being pumped from the collector. Maximum sensitivity is obtained when all collector gases are pumped through the lower vacuum line.

The total system is connected to the collector, located in the furnace, using 3/4" OD thick-wall PyrexTM tubing. Furnace inside dimensions are 15" deep, 9" wide and 4.5' long with 4.5" diameter holes in each end to facilitate loading and unloading of the collectors. Furnace temperature is controlled using a Honeywell two level controller. Set temperature can be maintained $\pm 5^{\circ}\text{C}$ and actual temperature is monitored using separate thermocouples and a millivolt potentiometer.

Collector Evacuation

Two collector pump down procedures were evaluated. Because of ease of handling, the collector was placed in the furnace while at room temperature, connected to the vacuum system and pumping started. At the same time the furnace was turned on and brought up to temperature.

Approximately 30 minutes were required for the furnace to reach 300°C (572°F) for a collector containing an absorber coated with Black MagicTM. Roughing time was over 10 minutes to reach 20 microns and the collector pressure remained in the 10^{-3} Torr range until the furnace reached the set temperature. Apparently substantial quantities of gas were being given off continuously as the temperature increased thus keeping the pump loaded.

The second procedure involved loading the collector in the

furnace while the furnace was at the desired temperature, connecting the pump system and immediately starting evacuation. The time required to make the vacuum connections allowed the furnace to recover and the collector tube pressure reached the 10^{-4} Torr range easily in less than 10 minutes from start of pumping. This is by far the best procedure for production because the collector is exposed to a lower vacuum at bake out temperature for a larger part of the evacuation cycle. In both cases, standard tipoff procedures were used.

A three hour cycle was found adequate for a tip-off room temperature pressure in the low 10^{-4} Torr range using the second procedure. The usual cycle time for the first procedure was arbitrarily set at four hours which resulted in the same level of vacuum after tip-off as was reached in the shorter, hot-pumped procedure.

All collector tubes evacuated using this vacuum system had ion gauges mounted in the tubes. A pressure differential of two orders of magnitude between the collector tube and the diffusion pump was always observed. This is not surprising, considering the out-gassing load presented by the collector at the bake-out temperature.

Calculated pumping speed at the collector tube was one liter per second with both pumping lines open. Typical pressure measurements just prior to tip-off were 6×10^{-5} Torr in the collector and 4×10^{-7} Torr above the pump. A considerable longer cycle (possibly 12 to 24 hours) would reduce the pressure difference by lowering the collector pressure but this is not economically feasible.

A 300°C (572°F) bake-out temperature for copper oxide Black MagicTM absorber films was established based on optical selectivity measurements. Film measurements made on the Gier-Dunkel Solar Reflectometer before and after bake-out indicate an absorptance loss of 3 to 5 percent after exposure to 300°C in vacuum for 3 hours. This indicates that 300°C is an absolute maximum temperature with a 25 to 50°C reduction in temperature possibly being an advantageous trade-off against pumping time.

A decision for a 400°C (752°F) bake-out for collectors containing "Black Chrome" absorbers was made based also on selectivity data. Absorptance measurements show a reduction of only 2 to 3 percent in absorptance after a 400°C bake-out of "Black Chrome" chromium oxide films. "Black Chrome" films have a 5 to 10 percent higher absorptance than Black MagicTM films. Thus the smaller reduction during bake out is not as detrimental to overall collector operation.

Both the 300°C Black MagicTM and the 400°C "Black Chrome" collectors equilibrate at a pressure between 1×10^{-4} and 5×10^{-4} Torr at room temperature after tip-off.

Vacuum Maintenance

A relatively inexpensive method of achieving and maintaining a high vacuum in a static system is through the use of a getter material. The flashing of a barium getter in the collector tube easily reduces the room temperature pressure from the 10^{-4} Torr range to the 10^{-6} Torr range.

A barium antenna getter assembly obtained from King Laboratories Inc. containing a barium charge of 253 mg was used for initial experiments. With a proper getter flash, 80% or nearly 200 mg of barium should be available for getter action. Work on proper

getter flashing techniques is continuing

Conclusions

Some degradation of Black MagicTM absorber films occurs during the 300°C bake-out.

The 400°C bake-out of the "Black Chrome" absorber film for three hours is acceptable.

A faster pump down and a low pressure is maintained over a longer portion of the evacuation cycle when the collector is near the bake-out temperature when pumping is started.

Getter flashing is necessary to reach a vacuum level in the 10^{-6} Torr range using the processes as outlined. Proper getter flashing will increase the time that satisfactory vacuum levels can be maintained.

TEST FACILITIES

To evaluate the performance of a single evacuated solar collector tube, and ultimately modular arrays in the latter part of the program, indoor and outdoor test facilities were set up.

INDOOR TEST FACILITY

A major problem in conducting outdoor solar tests is the limited availability of sunlight due to adverse and frequently changing weather conditions. Therefore, the need for a solar simulator that closely approximates solar radiation and can be used indoors, year round is apparent. The major portion of this task includes the design, construction, and operation of such a simulator.

Design and Construction

Sophisticated commercial solar simulators are generally high in cost. Within the framework of our program, the main purpose of the simulator is to determine the relative effect of design changes rather than absolute ratings for collector performance.

Consequently a NASA-Lewis concept²¹ was adopted for design and construction purposes. This low-cost Air Mass 2 solar simulator is dimensioned to irradiate a 4' X 1' area with good uniformity. A total of 40 General Electric QuartzlineTM ECH lamps, rated 300 watts at 120 volts, were used in the simulator. These lamps contain a tungsten-halogen bulb in an open, ellipsoidal reflector having a dichroic coating. Nonejector lamp holders were mounted on a 3/8" thick aluminum sheet with four rows of ten holders per row. The lamps were individually rated for intensity and then inserted into the holders in a pattern to secure a uniform flux. Optical grade acrylic Fresnel lenses were used to collimate the light from the lamps. The 0.09" thick lenses have a circular groove density of 100 lines per inch and a focal length of 10". The diameter of the outermost groove is 6". The lenses were cut into hexagonal shape; 3 15/32" on each of 6 edges and 6" across the flats. They were mounted on a piece of plywood placed 10 1/4" in front of the lamp plate and fastened with screws at each junction where three lenses meet. Threaded rods were placed between the lenses and the lamp plate to give the lenses more support and prevent them from buckling due to heating. A sheet metal shroud was installed on the sides and across the top of the lamp plate to direct the air flow from two 20" fans which cooled the lamps and lenses.

The overall dimensions of the simulator are 88" X 60" X 38 1/2". Figure III-37 shows a photograph of the simulator. Each lamp-lens combination is aligned on an identical axis. It should be noted that the central portion of the light beam is the most uniform section of the array. There is an edge-effect at the perimeter of the array which results in a dip in irradiance at the edges. The intensity of the lamps can be adjusted by varying the lamp voltage using the Variac on the associated power supply.

Set-Up and Testing Procedure

A four foot evacuated flat plate collector manufactured by CGW (Fig. I-1C) was tested using the simulator. The collector was evacuated to 10^{-6} Torr. The absorber plate was 0.016 thick copper, coated with "Black Chrome" and welded to 5/16" OD copper tubing. Absorptance of the absorber was approximately 90%, emittance 10%.

The collector was mounted on plywood and located 9'8" from the simulator. The simulator was turned on and allowed to warm up for ten minutes. Because of the dimensions of the test plane, only three rows of lamps were used in this test. The basic equation governing solar thermal collector efficiency is:

$$\eta = \frac{\dot{m} C_p \delta T}{A_c H} \quad (\text{III-73})$$

where η = collector efficiency

$\dot{m}$ = mass flow rate, lb/hr

C_p = specific heat, BTU/lb °F

δT = change in temperature ($T_{\text{out}} - T_{\text{in}}$), °F

A_c = area of collector absorber plate, ft²

H = solar simulator flux, BTU/hr ft²

Fig. III-37

A schematic of the optical and thermal arrangement for solar simulation testing is given in Fig. III-38. The flux was monitored in the plane of the tube axis. The change in temperature was monitored with platinum resistance thermometers that measured inlet and outlet temperature of the fluid passing through the absorber plate. The mass flow rate was determined by weighing on a balance the fluid collected in a 60 sec interval. As a preliminary test run, the effect of the angle of incidence on the collector efficiency was measured. The graph in Fig. III-39 represents the results obtained with the system shown in Figs. III-37 and III-38. Because of the cylindrical symmetry of the envelope, and the flat absorber plate, the efficiency (η) versus angle of incidence (θ) curve should exhibit a cosine behavior. As can be seen from Fig. III-39, the measured efficiency approximately follows the cosine of the angle of incidence. The improvement towards higher θ is probably due to the increased scattering of light from the background structure.

Several improvements need to be made on the solar simulator. The air flow from the fans needs adjusting so that the lenses do not heat up as much as they presently do. As the lenses heat and expand, they deform or buckle. This in return affects the distribution of light at the test plane. Work also needs to be done to improve the irradiance uniformity across the width of the light beam. The uniformity of the light beam varies with separation distance between the simulator and the test plane.

OUTDOOR TEST FACILITY

Set-Up and Testing Procedure

The objective of the outdoor test facility is to evaluate the performance of the evacuated collector under true-life conditions.

FIGURE III - 38

OPTICAL AND THERMAL ARRANGEMENT FOR SOLAR SIMULATION TESTING

FIGURE III-39

EFFECT OF INCIDENCE ANGLE ON THE PERFORMANCE OF THE EVACUATED FLATPLATE COLLECTOR TESTED WITH A SOLAR SIMULATOR.

The schematic of the test facility completed in this phase of the program is given in Fig. III-40.

The heat transfer fluid (a 50% by weight ethylene glycol-water solution) at a controlled temperature is pumped through the flow meter and then to the solar collector under test. Thermocouples in the inlet and outlet manifold of the collector module are used to monitor the inlet and outlet temperatures, T_{in} and T_{out} of the fluid, respectively. The ambient temperature T_a , is also measured. The value of solar flux is obtained with a commercial pyronometer, Eppley PSP or Spectrolab SR-75. Wind velocity is monitored with a WeathermeasureTM, Model W123 anemometer.

The outdoor meters and sensors are coupled to an indoor scanner-printer system by means of an electronic interface. This system consists mainly of signal conditioners and integrators. Some information is also recorded independently on strip charts for visual check and monitoring purposes.

The variable data required for a solar thermal performance test are:

- . Solar flux, H (BTU/ft² hr)
- . Mass flow rate $\dot{m}$ (lb/hr)
- . Inlet fluid temperature , T_{in} (°F)
- . Outlet fluid temperature, T_{out} (°F)
- . Ambient temperature, T_a , (°F)
- . Wind speed (ft/hr).

These data are taken continuously and the averaged numbers are automatically printed after preselected intervals. The efficiency of the collector is then calculated as in Eq. III-73.

Fig. III-40
Schematic of Outdoor Test Facility

Although some successful testing of the evacuated flat plate collectors has been done, additional work is still required to update the facility. The data integrator needs to be tested and new temperature sensing elements (platinum resistor thermometers) and their associated electronics need to be installed.

SECTION IVPLANS FOR THE REMAINING PART OF THE PROPOSAL

Figures IV-1 and IV-2 display the flow diagram and the work plan schedule for the heat-pipe solar collector program. respectively. According to these plans, the following tasks will be completed for the second progress report.

TASK 1: Collector Geometry and Analytical Modeling

(c and d) The cusp collector thermal-optical model will be completed by combining the results obtained for optical and thermal properties of the collector. The model will predict the thermal performance of single flow-through and heat pipe collectors as well as modular arrays.

TASK 2: Fabrication

(b) Based on the data obtained from selectivity and stability tests, the most suitable absorber coating will be chosen. Prototype heat pipes and flow-through absorbers will be coated with the chosen film and subjected to thermal performance tests.

(c) Remaining problems in the accuracy of cusp mirror fabrication will be analyzed in the light of the Monte Carlo digital ray tracing program. By means of a trade-off analysis, the most efficient commercial bright aluminum sheet will be chosen for the fabrication of the collector tubes.

(d) Copper glass-to-metal seals present a problem with regard to their strength. As a possible solution, a stronger Kovar-to-Code 7052 glass seal will be examined.

Figure IV-1
CORNING GLASS WORKS ADVANCED COLLECTOR PROGRAM

TASK

[illegible]

(e) Evacuation of the totally assembled cusp collector imposes some problems. A step-by-step bake-out and evacuation technique will be used to evacuate the collector efficiently. The application of getters will be investigated for long term vacuum maintenance.

(f) Six prototypes of heat-pipe and flow-through absorbers, developed in the first four-month period, will be assembled with the optimized cusp mirror and evacuated to form complete collectors. Also the design and construction of a manifold and support structure will be completed as a further step towards fabricating modular arrays.

TASK 3: Heat Pipe

(d) Following the thermal testing of the six prototypes completed in the first part of the program, failure, safety and life tests will be conducted. Depending on the results, the final choice will be made of the heat pipe design (material, wick, fluid) to be incorporated into arrays.

TASK 4: Flow-Through Absorber

(d) Using the already chosen design, a total of 10 flow-through collectors will be fabricated. After individual testing, these will be assembled to form a 4' X 4' modular array for outdoor testing.

TASK 5: Indoor Testing

(c) Using the test procedures established in the first four months of the program, components and single tube collectors will be parametrically tested with regard to their optical and thermal properties. This will include static and transient heat loss coefficients and performance measurements as a function of orientation, level of vacuum, and selective absorber films, and antireflecting coatings.

TASK 6: Outdoor Testing

Test procedures developed to measure the "efficiency curve" according to the proposed NBS standards will be employed to determine the performance of assembled single tube heat-pipe, flow-through and evacuated flat-plate collectors. The results will be compared with the predictions of the thermal-optical model.

TASK 7: Costing

Preliminary cost estimates will continue to be gathered.

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SECTION V

APPENDICES

APPENDIX A

FLUID PROPERTIES OF WATER

AND

SOME FLUORINERT LIQUIDS

Fluid Properties of Water (SI Units)

Figure V-1

Figure V-3

Density of the
Fluorinert Liquids

Figure V-4

Viscosity of the
Fluorinert Liquids

TABLE V-1
Properties of Water and FC-75

	<u>Freezing Point (K)</u>	<u>Boiling Point (K)</u>	<u>Critical Point (K)</u>	<u>Critical Press (N/M²)</u>	<u>Critical Volume M³/Kg</u>
Water	273.2	373.2	647.3	221.1 x 10 ⁵	3.134 x 10 ⁻³
FC-75	180.6*	375.6	500.6	160.0 x 10 ⁴	1.7 x 10 ⁻³

*Pour point - freezing point not available

APPENDIX B-1PERFORMANCE LIMITSCAPILLARY PUMPING LIMIT

This limitation on heat pipe performance applies mainly to heat pipes being run at or near horizontal (or with the evaporator above the condenser) with a fill sufficient to saturate the wick without a large excess puddle.

The available capillary pumping power to drive the heat pipe is a property of the working fluid and the wick. In general, the expression is of the form: surface tension divided by some characteristic dimension times a constant. A certain pumping heat loss must be overcome (the difference in pressure due to the elevation of the evaporator over the condenser), and the remainder can be used to overcome the viscous pressure drop experienced by the liquid flowing down the pipe through the wick.

In general, the heat pipe will dry out when the following condition is not met:

$$\Delta P_{CAP} > \Delta P_{HEAD} + \Delta P_{FLOW}$$

For details on how these terms might be calculated, see Appendix B-2 below: "Computer Program for Heat Pipe Performance Limits."

SONIC LIMIT

The sonic limit is the rate of heat transfer at which the Mach number of the vapor flow at the end of the evaporator equals one. Thus,

$$Q_{\text{SONIC}} = \lambda A_v \rho_v V_{\text{SONIC}}$$

To evaluate this expression in terms of conditions at the beginning of the evaporation; i.e., at stagnation; the equation developed by Levy¹ may be used:

$$Q_{\text{SONIC}} = \lambda A_v \rho_v \sqrt{\frac{\gamma RT}{2(\gamma + 1)}}$$

where γ is the ratio of specific heats.

ENTRAINMENT LIMIT

The entrainment limit is that point at which the vapor inertial forces become significant with respect to liquid surface tension forces, and droplets of liquid are torn from the wick.

The ratio of these forces, called the Weber number, is expressed as:

$$W_e = \frac{\rho_v v^2 z}{\sigma}$$

where v is the average vapor velocity and z is a characteristic dimension associated with the liquid surface. Taking the limiting condition as a Weber number of one,

$$Q_{\text{ENTRAINMENT}} = A_v h_{fg} \sqrt{\frac{\rho_v \sigma}{z}}$$

PRESSURE GRADIENT LIMIT

For heat pipes relying on gravity-aided liquid flow, a new limit has been proposed by Kemme, et al:²

$$Q_{\text{MAX}} = A_v \lambda \sqrt{\frac{\rho_v}{A} \cdot \frac{\rho g h_e}{2}}$$

where A_v is the vapor flow area, A is a constant (equal to $\pi^2/8$ for most ambient temperature heat pipes), and h_e is the evaporator length. This limit was derived analytically after experimental results suggesting it were obtained.

APPENDIX B-2COMPUTER PROGRAM FOR HEAT PIPEPERFORMANCE LIMITS

The program is a FORTRAN II-language program set up to calculate the required charge and performance limits of simple heat pipes on the G265 computer. The basic outline of the program is shown in Fig. V-6. All inputs and calculations are performed in SI units.

The first request is for basic heat pipe data: NFLUID, NWICK, TEMP, LEFF, L, AVAP, and IDIAM. (See Appendix A-3 for a list of input variables). If NFLUID is 1, the user is then asked for the fluid properties. If NFLUID is other than 1, tables of properties for the appropriate fluid are read into storage, and properties corresponding to the temperature are calculated by linear interpolation.

Next, depending on the type of wick, after receiving appropriate inputs the program assigns a characteristic length for calculation of the entrainment limit, and calculates the capillary heat rise, the required fluid charge, and the fluid loss coefficient.

The characteristic dimension is the effective pore radius for a homogeneous wick (screen, felt metal, foam metal, or sintered powder), and the width of the groove opening for an axially-grooved wick.

The required fill is calculated by multiplying the available liquid volume by the liquid density at that temperature, and

FIGURE V - 6
FLOW DIAGRAM FOR HEAT PIPE PERFORMANCE LIMITS

adding to it the product of the vapor volume and vapor density.

The capillary head rise (the capillary pressure generated by the wick structure) and the fluid loss coefficient ($\Delta P/Q$ - a measure of the viscous pressure drop experienced by the liquid flowing down the pipe through the wick) are dependent on the type of wick geometry chosen. Liquid pressure drops are calculated by some modification of the Hagen-Poiseuille law.

For a homogeneous wick, two experimentally-determined parameters are required: r_c , an effective pore radius, and C_r , a dimensionless constant including both permeability and a tortuosity factor. Then

$$CL = \frac{\Delta P}{Q} = \frac{\mu L_{eff}}{h_{fg} \rho_f A_L r_c^2 C_r}, \text{ and}$$

$$\Delta P_{cap} = \frac{2\sigma}{r_c}.$$

For rectangular axial grooves, with a width D and depth W , the pressure drop in flow down a rectangular tube of dimension $2D \times W$ is calculated using the friction factor given by Kays,³ and using the procedure defined by Florschuetz,⁴ which leads to the equation

$$CL = (f \cdot R_e) \frac{2\mu L_{eff}}{h_{fg} \rho_f A_L D_h^2 n_g}$$

and

$$\Delta P_{cap} = \frac{2\sigma \cos \theta}{w},$$

where D_h is the hydraulic diameter of the groove, n_g is the number of grooves around the periphery, and θ is the wetting angle of the fluid in contact with the groove wall.

For trapezoidal grooves or irregular grooves, the pressure drop is calculated using the Hagen-Poiseuille law for flow in circular tubes, and using the hydraulic diameter instead of the tube diameter:

$$CL = 32 \frac{\mu L_{eff}}{h_{fg} \rho_f A D_h^2 n_{gr}}$$

In both cases, the capillary pressure rise is given by the expression

$$p_{cap} = \frac{2\sigma \cos(\theta + \alpha)}{w}$$

where α is the deviation of the wall angle from vertical.

A vapor pressure drop is then calculated from

$$CV = \frac{\Delta P_{vap}}{Q} = 32 \frac{\mu_v L_{eff}}{h_{fg} \rho_v A_v D^2}$$

When the head loss, the required static pumping head (which is the vertical distance from the top of the evaporator to the bottom of the condenser) is calculated using the expression

$$\Delta P_{head} = \rho_f g(\text{tilt} + D),$$

the wick pumping limit can then be calculated.

The procedure used is taken from that of Kosson⁵ yielding:

$$Q_{\max} = \frac{-\left(\frac{2a_4}{a_3}\right)}{1 + \sqrt{1 - \frac{4a_1^2 a_4}{a_3^2}}}$$

the equation used in the program.

Equations for the ionic and entrainment limits are taken from Appendix B-1 to complete the equations used in the heat-pipe program.

APPENDIX B-3

Input Variables for B-2

ALPHA (radians)	-	angle by which the groove wall departs from vertical.
AREAL (m^2)	-	for homogeneous wick, wick cross-sectional area for groove wick, area of one groove.
AVAP (m^2)	-	flow area of the vapor space.
CR	-	empirical dimensionless constant incorporating permeability.
D (m)	-	depth of groove.
GAMMA	-	ratio of specific heats (C_p/C_v)
GRNO	-	number of grooves around the circumference.
HFG (J/kg)	-	latent heat of vaporization.
IDIAM (m)	-	inside diameter of the heat pipe.
L (m)	-	overall length of the heat pipe.
LEFF (m)	-	distance between midpoints of evaporator and condenser.
MW	-	molecular weight.
NFLUID	-	fluid code number required by program.
NFLUID=1	-	fluid properties to be input at terminal.
NFLUID=2	-	water properties to be read from table.
NFLUID=3	-	methanol properties to be read from table.
NFLUID=4	-	ammonia properties to be read from table.
NWICK	-	wick code number required by program.
NWICK=1	-	homogeneous wick.
NWICK=2	-	rectangular axial grooves.
NWICK=3	-	trapezoidal axial grooves.
NWICK=4	-	axial grooves of irregular cross-section.
PERIM (m)	-	wetted perimeter of a groove.
POROS	-	volume porosity of homogeneous wick material.
PVAP (N/m^2)	-	vapor pressure.
RC (m)	-	effective pore radius of homogeneous wick.
RHOF (kg/m^3)	-	fluid density.
RHOV (kg/m^3)	-	vapor density.
STEN (N/M)	-	surface tension.
TEMP (K)	-	temperature.
THETA (radians)	-	wetting angle at fluid/wall interface.
VIS ($N-s/m^2$)	-	liquid dynamic viscosity.
VISV ($N-s/m^2$)	-	vapor dynamic viscosity.
W (m)	-	width of groove at top.
WR (m)	-	width of trapezoidal groove at base.

APPENDIX B-4

References for B-2

1. Levy, E. K., "Theoretical Investigation of Heat Pipes Operating at Low Vapor Pressures," Trans. ASME: JET, Vol. 90, Series B, No. 4, November, 1968, pages 547 - 552.
2. Kemme, J. E., "Heat Pipe Design Considerations," Report LA-4221, Los Alamos Scientific Laboratory, Los Alamos, New Mexico, August 1969.
3. Kays, W. M., Convective Heat and Mass Transfer, McGraw-Hill, 1966.
4. Florschuetz, L. W., "Brief Notes on Calculation of Liquid Pressure Drop in Heat Pipes," Los Alamos Scientific Laboratory, N-5 Tech. Memo No. 170, August, 1970.
5. Kosson, R., "Lecture Notes - Composite Heat Pipe Wick Systems," Grumman Aerospace Corporation, Bethpage, New York, 1973.

SUN 9:09 TUES. 07/01/75

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100 .SFILE TTYSUM, WATER, METH, NH3
110 DIMENSION XRHOF(30), XRHOF(30), XTEMP(30), XHFG(30), XSTEN(30), XVIS(30)
120 +XVISV(30), XPVAP(30)
130 REAL LEFF, MW, L, IDIAM
140 NREC=0
150 REWIND 1
160 KOUNT=0
170 ITILT=0
180 PRINT, "NFLUID, NWICK, TEMP, LEFF, L, AVAP, IDIAM=? "
190 INPUT, NFLUID, NWICK, TEMP, LEFF, L, AVAP, IDIAM
200 999 GO TO (1001, 1002, 1003, 1004), NFLUID
210 1001 PRINT, "FLUID PROPERTIES?"
220 INPUT, RHOF, RHOV, HFG, STEN, VIS, VISV, GAMMA, PVAP, MW, THETA
230 GO TO 1999
240 1002 PRINT, "FLUID CHOSEN IS WATER"
250 IF (KOUNT) 1900, 1900, 1906
260 1003 PRINT, "FLUID CHOSEN IS METHANOL"
270 IF (KOUNT) 1900, 1900, 1906
280 1004 PRINT, "FLUID CHOSEN IS AMMONIA"
290 IF (KOUNT) 1900, 1900, 1906
300 1900 READ(NFLUID) N
310 READ (NFLUID) (XTEMP(I), I=1, N), (XPVAP(I), I=1, N), (XRHOF(I), I=1, N),
320 + (XRHOV(I), I=1, N), (XSTEN(I), I=1, N), (XHFG(I), I=1, N), (XVIS(I), I=1, N)
330 +, (XVISV(I), I=1, N), GAMMA, MW
340 1906 KOUNT=1
350 IF (TEMP-XTEMP(1)) 1990, 1901, 1901
360 1901 IF (TEMP-XTEMP(N)) 1902, 1902, 1990
370 1902 I=2
380 1903 IF (TEMP-XTEMP(I)) 1905, 1905, 1904
390 1904 I=I+1
400 GO TO 1903
410 1905 J=I-1
420 RATIO=(XTEMP(I)-TEMP)/(XTEMP(I)-XTEMP(J))
430 RHOF=XRHOF(I)-(XRHOF(I)-XRHOF(J))*RATIO
440 RHOV=XRHOV(I)-(XRHOV(I)-XRHOV(J))*RATIO
450 HFG=XHFG(I)-(XHFG(I)-XHFG(J))*RATIO
460 STEN=XSTEN(I)-(XSTEN(I)-XSTEN(J))*RATIO
470 VIS=XVIS(I)-(XVIS(I)-XVIS(J))*RATIO
480 VISV=XVISV(I)-(XVISV(I)-XVISV(J))*RATIO
490 PVAP=XPVAP(I)-(XPVAP(I)-XPVAP(J))*RATIO
500 GO TO 1999
510 1990 PRINT, "TEMPERATURE SELECTED IS OUT OF THE RANGE OF THE TABLE."
520 PRINT, " IF YOU WISH TO:"
530 PRINT, " INPUT FLUID PROPERTIES BY HAND, TYPE -1"
540 PRINT, " TRY A DIFFERENT TEMPERATURE, TYPE 0"
550 PRINT, " STOP WHILE YOU TRY TO REGROUP, TYPE 1"

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APPENDIX B-5

Printout of B-2

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560 INPUT, ICHEK
570 IF (ICHEK)1001,997,998
580 1999 GO TO (100,200,300,400), MWICK
590 100 PRINT,"WICK IS HOMOGENEOUS SCREEN OR FELT"
600 IF (ITEMP) 101,101,102
610 101 PRINT,"RC,CR,LIQUID FLOW AREA,POROSITY=?"
620 INPUT, RC, CR, AREAL, POROS
630 102 CL=VIS*LEFF/(HFG*RHOE*AREAL*RC*RC*CR)
640 Z=RC
650 FILL=(AREAL*POROS*RHOE+AVAP*RHOV)*L
660 PCAP=2.*STEN/RC
670 GO TO 1000
680 200 PRINT,"RECTANGULAR AXIAL GROOVES"
690 IF (ITEMP) 201,201,20
700 201 PRINT,"D,W,GRNO=?"
710 INPUT,D, W, GRNO  $\text{AREAL} = D * W$ 
720 DHYD=4.*D*W/(2.*D+W)
730 ASTAR=2.*D/W
740 ISTAR=(1./ASTAR)*10.
750 GO TO (1,2,3,4,5,6,7,8,9,10), ISTAR
760 1 FXRE=24.+26.*(-1./ASTAR)
770 GO TO 20
780 2 FXRE=21.4+22.*(.1-1./ASTAR)
790 GO TO 20
800 3 FXRE=19.2+16.*(.2-1./ASTAR)
810 GO TO 20
820 4 FXRE=17.6+11.*(.3-1./ASTAR)
830 GO TO 20
840 5 FXRE=16.5+9.*(.4-1./ASTAR)
850 GO TO 20
860 6 FXRE=15.6+6.*(.5-1./ASTAR)
870 GO TO 20
880 7 FXRE=15.+3.*(.6-1./ASTAR)
890 GO TO 20
900 8 FXRE=14.7+2.*(.7-1./ASTAR)
910 GO TO 20
920 9 FXRE=14.5+2.*(.8-1./ASTAR)
930 GO TO 20
940 10 FXRE=15.2-1./ASTAR
950 20 CL=FXRE*2.*VIS*LEFF/(HFG*RHOE*AREAL*DHYD*DHYD*GRNO)
960 FILL=(D*W*GRNO*RHOE+AVAP*RHOV)*L
970 Z=W
980 PCAP=2.*STEN*COS(THETA)/W
990 GO TO 1000
1000 300 PRINT,"AXIAL GROOVE WICK, TRAPEZOIDAL GROOVES"
1010 IF (ITEMP) 301,301,302
1020 301 PRINT,"W,WR,D,GRNO=?"
1030 INPUT, W, WR, D, GRNO
1040 AREAL=(W+WR)*D/2.
1050 PERIM=WR+2.*SORT(((W-WR)/2.)*((W-WR)/2.)+D*D)
1060 DHYD=4.*AREAL/PERIM
1070 ALPHA=ATAN((W-WR)/(2.*D))
1080 302 CL=32.*VIS*LEFF/(HFG*RHOE*AREAL*DHYD*DHYD*GRNO)
1090 FILL=(AREAL*GRNO*RHOE+AVAP*RHOV)*L

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1100 Z=W
1110 PCAP=2.*STEN*COS(THETA+ALPHA)/W
1120 GO TO 1000
1130 400 PRINT,"AXIAL GROOVES - IRREGULAR CROSS-SECTION"
1140 IF (ITEMP) 401,401,402
1150 401 PRINT,"WETTED PERIMETER, FLOW AREA PER GROOVE, GRNO, W, ALPHA=?"
1160 INPUT, PERIM, AREAL, GRNO, W, ALPHA
1170 DHYD=4.*AREAL/PERIM
1180 402 CI=32.*VIS*LEFF/(HEG*RHOE*AREAL*DHYD*DHYD*GRNO)
1190 FILL=(AREAL*GRNO*RHOE+AVAP*RHOV)*L
1200 Z=W
1210 PCAP=2.*STEN*COS(THETA+ALPHA)/W
1220 GO TO 1000
1230 1000 QSONC=AVAP*HFG*RHOV*SQR(GAMMA*8.31434*TEMP/(MW*2.*(GAMMA+1.)))
1240 QENT=AVAP*HEG*SQR(RHOV*STEN/Z)
1250 CV=32.*VISV*LEFF/(RHOV*HFG*AVAP*IDIAM*IDIAM)
1260 1500 PRINT,"ADVERSE TILT = ?"
1270 INPUT,TILT
1280 PHEAD=RHOE*9.81*(TILT+IDIAM)
1290 NREC=NREC+1
1300 A1=CI/PVAP
1310 A2=(PCAP-PHEAD)/PVAP
1320 A3=2.*A1*(1.-A2)+2.*CV/PVAP
1330 A4=(1.-A2)*(1.-A2)-1.
1340 OWICK=-2.*A4/(A3*(1.+SQR(1.-4.*A1*A1*A4/A3)))
1350 WRITE (1) TILT,OWICK,QENT,QSONC,FILL,TEMP
1360 IF (ITILT) 1497,1497,1498
1370 1497 PRINT,"REQUIRED CHARGE = ",FILL,"KG"
1380 PRINT,"QSONIC = ",QSONC," WATTS"
1390 PRINT,"QENTRAINMENT = ",QENT," WATTS"
1400 1498 PRINT,"OWICK = ",OWICK," WATTS"
1410 PRINT/"ANOTHER TILT? IF YES, TYPE 1. IF NO TYPE 0."
1420 INPUT,ITILT
1430 IF (ITILT) 1499,1499,1500
1440 1499 CONTINUE
1450 PRINT"NEW TEMPERATURE? IF YES, TYPE 1. IF NO, TYPE 0."
1460 INPUT,ITEMP
1470 IF (ITEMP) 998,998,997
1480 997 PRINT,"TEMP=?"
1490 INPUT,TEMP
1500 GO TO 999
1510 998 PRINT,"DO YOU WANT A TABLE SUMMARIZING YOUR RESULTS?"
1520 PRINT,"TYPE 1 FOR YES, 0 FOR NO"
1530 INPUT, ICHEK2
1540 IF (ICHEK2) 996,996,995
1550 995 REWIND 1
1560 PRINT,"TILT","OWICK","QENT","QSONIC","FILL","TEMP"
1570 PRINT," (M) "," (W) "," (W) "," (W) "," (KG) "," (K) "
1580 DO 994 I=1,NREC
1590 READ (1) TILT,OWICK,QENT,QSONC,FILL,TEMP
1600 PRINT," "
1610 PRINT,TILT,OWICK,QENT,QSONC,FILL,TEMP
1620 994 CONTINUE
1630 996 CONTINUE
1640 STOP
1650 END

```

WATER 9:19 TUES. 07/01/75

100 12

110 275., 300., 325., 350., 375., 400., 425., 450., 475., 500., 525., 550., 700.

120 3500., 13000., 42000., $1.E+5$, $2.4E+5$, $4.17E+5$, $8.5E+5$, $1.5E+6$, $2.6E+6$, $4.E+6$

130 $6.E+6$, $1.E+3$, 995., 985., 970., 960., 940., 915., 890., 860., 830., 800., 750.

140 $5.5E-3$, $2.5E-2$, .09., .25., .60., 1.3, 2.5, 4.6, 8.0, 13., 22., 34.

150 .075., .071., .057., .0628., .058., .0535., .0485., .043., .033., .0325., .027., .022

160 $2.5E+6$, $2.44E+6$, $2.37E+6$, $2.31E+6$, $2.24E+6$, $2.18E+6$, $2.1E+6$, $2.01E+6$, $1.93E+6$

170 $1.82E+6$, $1.7E+6$, $1.55E+6$, $1.6E-3$, $8.3E-4$, $5.2E-4$, $3.7E-4$, $2.8E-4$, $2.1E-4$

180 $1.8E-4$, $1.5E-4$, $1.3E-4$, $1.2E-4$, $1.E-4$, $9.E-5$, $9.6E-6$, $1.02E-5$, $1.12E-5$

190 $1.2E-5$

200 $1.32E-5$, $1.44E-5$, $1.58E-5$, $1.74E-5$, $1.9E-5$, $2.12E-5$, $2.38E-5$, $2.72E-5$

210 1.33, 18.