Report No. COO-2608-3

AN EVACUATED TUBULAR
COLLECTOR UTILIZING
A HEAT PIPE

ERDA Contract EY-76-C-02-2608

Final Report for
the Period
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ABSTRACT

This final report concludes the evaluation of the three evacuated tubular solar thermal collector designs described in previous ERDA Reports No. COO-2608-1 and COO-2608-2

- a. Heat Pipe/Cusp
- b. Counter Flow/Cusp
- c. Counter Flow/Flat Plate

During the period 1 August 1977 through 30 September 1977 comparative field testing of 4' x 4' modules was completed for all three designs. Thermal analysis of data shows that the evacuated heat pipe/cusp collector is a viable concept, more suitable for medium to high temperatures and high solar flux intensities.

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1. INTRODUCTION

It is well established that an evacuated collector with an internal pressure equal to or less than 10^{-4} Torr, with a highly selective absorber coating i.e. $\alpha/\epsilon \geq 10$ can achieve efficiencies of 55-60% at a ΔT of 200°F and a flux of $250 \text{ BTU/ft}^2 \text{ hr}$.

In the development of these advanced collectors for residential or commercial use, factors other than the thermal performance start gaining more importance. Excluding the all important manufacturing cost, other technical features and the method of interfacing them to the rest of the solar system may greatly vary from one evacuated collector to another. The shape of the absorber, the method of vacuum confinement, the heat transfer mode in the collector and the technique of manifolding influence the applicability of a certain type of collector.

Under this contract, a novel heat pipe/cusp collector was examined in comparison with two other forms of evacuated tubular collectors. Figure 1 shows the cross section of these collectors and depicts the important design and heat transfer of each type. The distinct difference between the geometrically identical cusp collectors is the conversion and transfer of solar energy within the collector. The principles of the heat pipe action in the cylindrical absorber have been treated in the previous report under this contract.

Comparative testing of the three concepts as single tube collectors has been summarized in Report No. COO-2608-2. The

SCHEMATIC OF EVACUATED TUBULAR SOLAR COLLECTOR CONCEPTS UNDER EVALUATION.

ADVANCED HEAT-PIPE
COLLECTOR

TUBULAR ABSORBER, CUSP
REFLECTOR, HEAT REMOVAL
BY HEAT-PIPE ACTION

ADVANCED COUNTER
FLOW COLLECTOR

TUBULAR ABSORBER, CUSP
REFLECTOR, HEAT REMOVAL
BY TRANSFER LIQUID

CORNING BASELINE
COLLECTOR

FLAT-PLATE ABSORBER,
HEAT REMOVAL BY
TRANSFER LIQUID

Figure 1

major task of this final phase was the simultaneous field testing of the 4' x 4' modules consisting of 10 tubes each under the same ambient conditions. For this purpose, modules manufactured at Corning Glass Works' solar facility were shipped to Desert Sunshine Exposure Tests, Inc. to undergo parametric testing. Presentation of the results obtained and their analytical evaluation in this report concludes the final tasks of the contract.

2. GENERAL DESCRIPTION OF THE COLLECTOR MODULES

The schematic of the basic 4' x 4' module design used for all the three collector types is shown in Fig. 2. Section B in this figure represents the manifold box of the 10 tube array. The box contains the manifold assembly carrying the heat transfer fluid to and from the collector tubes. Ample insulation is supplied around the manifold to keep the "manifold losses" down to a minimum. This has been a critical issue during the experiments because of the small absorber-to-manifold ratio affecting the efficiency rather drastically. Section A fulfills the structural function of holding the collector tubes together and improving the structural rigidity and strength. The tubes are separated by $3/8$ of an inch from each other. Although it is not essential, the manifold is usually mounted and fastened on half inch plywood by screws in sections A and B.

Since the heat removal mechanisms are totally different in the case of the heat pipe and conventional flow-through collectors, they will be described separately below.

2.1 Heat Pipe/Cusp Collector Module

Figure 3 shows the isometric sketch and the flow cross-section of the Heat Pipe/Cusp Collector Manifold. The solar radiation converted into latent heat of the absorber is removed from the interior of the collectors by the heat pipe action. Each heat pipe has a one inch diameter by five inch

H = OVERALL HEIGHT

	L (inches)	W (inches)	H (inches)
HEAT PIPE MODULE	68 1/4	56 1/2	10 1/2
COUNTER FLOW MODULE	68 1/4	56 1/2	10 1/2
FLAT PLATE MODULE	69 1/2	45	7

Figure 2

Figure 3

4.

long condenser cylinder to reject the sensible heat into the heat transfer fluid in cross-flow. Figures 4 and 5 show further details of the manifold mounting and the "Plug-In" sockets. Both the manifold and the socket were made out of stainless steel. As seen from Fig. 5 the collector is coupled to the socket by means of a clamp designed for Corning Chemical Systems. Overall dimensions of the exterior box are shown in Fig. 6. Fiberglass material packed around the manifold fills the total interior of the box and furnishes the insulation against ambient. Photographs in Fig. 7a and 7b show the manifold and the collectors before assembly for testing. The overall projectional area of the module is 26.65 ft^2 . The associated aperture area used for efficiency calculations is 11.54 ft^2 . The pressure drop ΔP between the inlet and the outlet of the manifold is 1 to 2 psi over a modular mass flow rate range of about 165 to 350 lb/hr or 14 to 30 lb/hr per square foot of aperture.

2.2 Flat Plate and Counterflow/Cusp Collector Modules

In principle, both designs have the same type of manifold, although the flow patterns and absorber geometry inside the collector are very different. As shown in Fig. 8a the tubes are connected in series, i.e. the outlet of each single tube is connected to the inlet of the next tube. Insulation is provided by polyurethane foam which fills the interior of the "manifold box" (See Section B in Fig. 2). Photographs in Figs. 8a and 8b show a typical module before and after the foaming steps during fabrication.

Figure 4

LID

Figure 5

Figure 6

MANIFOLD FOR THE HEAT PIPE MODULE

Figure 7a

MANIFOLD FOR THE HEAT PIPE MODULE

Figure 7b

MANIFOLD PLUMBING FOR THE COUNTERFLOW COLLECTOR

Figure 8a

INSULATION PROCEDURE FOR MANIFOLD
OF THE COUNTERFLOW COLLECTOR

Figure 8b

The overall projectional areas of the flat plate and the counterflow/cusp modules are 26.64 ft^2 and 21.64 ft^2 , respectively. The flat plate module has an effective absorber area of 10.94 ft^2 , whereas the counterflow/cusp collector has an aperture of 11.73 ft^2 . The pressure drop is typically higher than in the heat pipe manifold. For a flow rate of about 200 lb/hr or 17-18 lb/hr per square foot aperture, ΔP for the flat plate and for the counterflow/cusp collectors are 3-5 and 5-8 psi, respectively.

3. EXPERIMENTAL PROCEDURES

Comparative outdoor testing of the collector modules was performed at the Desert Sunshine Exposure Test, Inc., Arizona. Since this test facility has been described elsewhere¹, no account of the instrumentation and the testing will be given here. The method of testing and accuracy of the data acquisition complies to a large extent with the proposed NBS standards.²

Because of the comparative nature of the investigation, collector modules were always tested in pairs simultaneously. Photographs in Figs., 9a and 9b show the installed combinations "Heat Pipe-Flat Plate" and "Counterflow/Cusp-Flat Plate" module pairs on the solar stand at DSET. Because of the tubular geometry of the collectors, definition of modular orientation is essential for all non-tracking applications. In the First Progress Report (No. COO-2608-1) a detailed analysis of the

Figure 9a

HEAT PIPE AND FLAT PLATE MODULES
MOUNTED ON THE TESTING PLATFORM.

COUNTER FLOW AND FLAT PLATE MODULES
MOUNTED ON THE TESTING PLATFORM

Figure 9b

single tube orientation on the total amount of radiation received throughout the days and seasons was given for different latitudes. For the sake of completeness we include here Figs. 10a and 10b for defining some of the characteristic collector orientations for flat plate and the cusp collectors. A multitude of possible experiments were considered to evaluate the relative merits of the collector designs under evaluation. Due to the limited time available for experimentation, the scope of the testing was restricted to the most characteristic parameters. The following test procedures were chosen to understand these parameters.

3.1 Procedure A: Instantaneous Efficiency Tests

Instantaneous efficiency of the modules was obtained by measuring the amount of energy removed by the transfer fluid per unit of effective solar intercept* area over 5 minute periods divided by the total solar radiation onto the collector per unit effective solar intercept area for the same 5 minutes.

*Effective solar intercept is defined as the opening of a solar collector through which the unconcentrated solar energy is admitted and directed to the absorber. In the case of the flat plate collector, the absorber itself is the effective solar intercept whereas for heat pipe and counterflow, it is the aperture of the cusp mirror.

a) HORIZONTAL EAST-WEST

b) VERTICAL EAST-WEST

c) POLAR NORTH-SOUTH

a) HORIZONTAL EAST-WEST

b) VERTICAL EAST-WEST

c) POLAR EAST-WEST

d) POLAR NORTH-SOUTH

ϕ = LATITUDE OF THE GEOGRAPHIC LOCATION

Figure 10b

The modules were mounted on a movable frame and tracked the sun continuously to maintain an incident angle of zero degrees.

The parameters measured and their associated integrated value over the five minute periods are shown in Table I in form of a typical routine data sheet. For each interval of measurement say $\tau_2 - \tau_1 = 5$ min. the instantaneous efficiency η shown in the bottom row is calculated from the relationship

$$\eta = \frac{\dot{m} C_p \int_{\tau_1}^{\tau_2} (T_o - T_i) d\tau}{A_a \int_{\tau_1}^{\tau_2} q_i d\tau}$$

The flow rate ($\dot{m}$) and the specific heat C_p have been taken out of the integration since their value does not fluctuate appreciable during the interval. It is important to note again that the area A_a used for the calculation is the effective solar intercept area as defined earlier. To establish the efficiency curves at least 8 data points were taken up to $\Delta T/H$ values of 0.65. Higher temperatures were not attempted to avoid excessive vapor pressures within the collectors.

Field data showing the flat plate collector efficiency curve in comparison with the heat pipe and counterflow/cusp modules are shown in Figs. 11 and 12, respectively. In both cases the flat plate performance exceeds that of the heat pipe and the counter flow by about 20% up to a $(\bar{T}_f - T_a)/q_i$ value of 0.65. The optical efficiency of the flat plate is about 20% higher than the intercept found experimentally for a single

TABLE I

DATE: August 7, 1977
 COLLECTOR: Heat Pipe/Cusp
 GLAZING: Single Glass Tube
 APERTURE AREA: 11.54 ft²
 TEST METHOD: Instantaneous

t, solar	09:52	11:07	11:15	13:16	13:24
m (lb/hr)	349.40	349.05	347.00	348.81	346.06
T _i (°F)	209.4	170.1	170.9	128.8	128.8
T _o (°F)	214.9	177.1	177.5	136.8	136.8
T _p (°F)	-	-	-	-	-
T _a (°F)	-	-	-	-	-
T _a (°F)	100.9	102.9	102.9	104.0	104.9
Wind Velocity	SW 6	SW 5	SW 10	SW 13	SW 6
Air Over					
Collector (fpm)	150	300	400	760	620
C _p (BTU/lb.°F)	1.00646	1.00207	1.00213	0.99905	0.99905
q _i (BTU/ft ² .hr)	310.43	315.13	315.66	312.42	311.23
q _i , % Diffuse	13.12	12.70	12.91	14.38	14.11
Tilt Angle	32.9	20.5	19.6	24.7	26.0
Incidence Angle	0	0	0	0	0
Azimuth Angle	66.3	36.2	31.3	-50.8	-54.0
P _{in} (psi)	19.8	21.0	21.0	19.8	19.8
ΔP (psi)	0	0	0	0	0
ΔT (°F)	5.6	7.0	6.6	8.0	8.0
T _f (°F)	212.1	173.6	174.2	132.8	132.8
T _p (°F)	-	-	-	-	-
T _f -T _a (°F)	111.2	70.7	71.2	28.8	27.9
T _p -T _a (°F)	-	-	-	-	-
T _i -T _a (°F)	108.4	67.2	67.9	24.8	23.9
(T _f -T _a)/q _i *	0.358	0.224	0.226	0.092	0.090
(T _i -T _a)/q _i *	0.349	0.213	0.215	0.079	0.077
(T _p -T _a)/q _i *	-	-	-	-	-
n _{q_i**}	169.78	212.38	197.805	240.50	239.68
Efficiency η	0.547	0.674	0.627	0.769	0.770

*°F/BTU/FT²
 **BTU/FT².hr

Figure 11

DATE: JULY 24, 1977
 COLLECTOR: EVACUATED FLAT PLATE
 RECEIVER AREA: 10.94 FT²
 COLLECTOR: COUNTER FLOW
 RECEIVER AREA: 11.73 FT²

Figure 12

tube collector as reported in the second progress report.³

The additional increase in optical efficiency is due to the contribution from the diffusely scattering background (ca. 16-18%, see Ref. 3) and the mutual scattering between the neighboring tubes at normal incidence as computed in the first progress report (ca. 4%, see Ref. 4). The discrepancy between the heat pipe and the counterflow optical efficiencies, i.e. 82% versus 76% is unexpected because of the identical geometry. The accuracy of the data excludes a statistical fluctuation of this magnitude. The expected value for the optical efficiency of the single tube cusp collector is ca. 70% (Ref. 3). The additional gain in the case of the counterflow is due to the mutual scattering between the adjacent tubes forming the module. The heat pipe is even higher by 6%. This effect is associated with the heat contribution from the much larger manifold. Under no wind conditions and normal solar incidence the manifold box cover may heat up about 20-30°F above ambient temperature. As will be shown later, there is another manifold term $q = S k (\bar{T}_c - T_f)$ which has to be considered in the overall heat balance equation of the whole module. The quantities $\bar{T}_c$ and $\bar{T}_f$ represent the average manifold box temperature and average fluid temperature, respectively, k is the thermal conductivity of the manifold insulation, and S is a shape factor and dependent on geometry. For counterflow and flat plate modules, S is very small compared to the shape factor of the heat pipe module. Thus, depending on the relative values of

$\bar{T}_c$ and $\bar{T}_f$, two kinds of effects are expected to influence the efficiency curve: For $\bar{T}_c > \bar{T}_f$ a positive contribution to the useful heat increases η . For $\bar{T}_c < \bar{T}_f$, q represents a heat loss and reduces η . A comparison of the efficiencies at $\Delta T/q_i = 0.5$ shows for example that the counterflow collector is about 8% higher than the heat pipe. This is in accordance with the expected excess manifold heat loss, i.e., a decrease in the heat pipe efficiency.

The flow rate dependence of the efficiency in the case of the heat pipe has been subject to further investigation. Fig. 13 shows the data obtained at flow rates 239 and 337 lb/hr, respectively. An increase in the flow rate from 170 lb/hr (Fig. 13) to 239 lb/hr improves the efficiency by about 4% at $\Delta T/q_i = 0.5$. However, further increase in flow to 337 lb/hr causes a drop of 9-10% at the same $\Delta T/q_i$. Thus an optimum flow rate must be around 239 lb/hr. The dependence of efficiency on the flow rate does not seem to be significant between ca. 15 to 30 lb/hr-ft².

3.2 Procedure B: Night Heat Loss Measurements

A good way to obtain information on the overall heat losses from a collector is to measure the temperature drop between the inlet and the outlet of the module during the night. The inlet temperature can be controlled to increase ($\bar{T}_f - \bar{T}_{amb}$) in steps over the range of interest. The quantity $\bar{T}_f$ is the average fluid temperature computed from the expression $(T_{in} + T_o)/2$.

Figure 13

At each stage T_{in} and T_o are measured after equilibrium has been reached for that given inlet temperature. Then the overall heat loss per square foot of effective area is computed by the formula $\dot{m} C_p (T_o - T_{in}) A$. A typical data sheet for this procedure is shown in Table II.

Figure 14 displays the heat loss curves for all three modules up to $\bar{T}_f - \bar{T}_{amb} = 160^\circ\text{F}$. This corresponds approximately to an average fluid temperature $\bar{T}_f$ of 245-250°F.

These curves supply complementary information to the efficiency curves shown in Fig. 11, 12 and 13. For example in Fig. 14 we find for the flat plate a loss of 50 BTU/hr-ft² at $\bar{T}_f - T_{amb} = 140^\circ\text{F}$. Using the insolation data on Fig. 11 we find that $\bar{T}_f - \bar{T}_{amb} / q_i = 0.46$ (°F-ft²-hr/BTU) and the corresponding efficiency is 78%. We can now verify this efficiency value using the loss data from Fig. 14 as follows. Using the optical intercept, i.e. $F_R \bar{\alpha} \tau = .96$ and an insolation of 306.94 BTU/ft²-hr (Fig. 11) we can calculate the efficiency from

$$\eta = \frac{\eta_{op} q_i - Q_{loss}}{q_i}$$

$$= \frac{(.96)(306.94) - 50}{306.94} = 0.797$$

This result is within two percent of $\eta=0.78$ found by instantaneous measurements described in Procedure A.

The relative behavior of the loss curves in Fig. 14 is not in agreement with the single tube experiments reported earlier. The heat pipe, because of the thermal diode feature

TABLE II

DATE: August 4-5, 1977

COLLECTOR: Flat-Plate Tubular

GLAZING: Single Glass Tube

APERTURE AREA: 10.94 ft²TEST METHOD: Night Heat Loss
Measurement

t, solar	21:29	21:37	22:54	23:02	01:09
$\dot{m}$ (lb/hr)	152.11	152.44	153.68	154.29	159.09
T_i (°F)	223.5	223.6	187.9	187.5	161.6
T_o (°F)	220.3	220.4	185.9	185.4	160.0
T_p (°F)	-	-	-	-	-
T_p (°F)	-	-	-	-	-
T_p (°F)	-	-	-	-	-
T_a (°F)	93.7	94.8	93.0	92.5	85.4
Wind Velocity	0	0	0	0	0
Air Over					
Collector (fpm)	0	0	0	0	0
C_p (BTU/lb.°F)	1.00118	1.00119	1.00350	1.00345	1.00090
q_i (BTU/FT ² .hr)	-0.38	-0.43	-0.48	-0.59	0
q_i , % Diffuse	69.19	65.95	232.57	192.45	-
Tilt Angle	-	-	-	-	126.7
Incidence Angle	-	-	-	-	-
Azimuth Angle	-	-	-	-	20.4
P_{in} (psi)	F.S.+	F.S.+	F.S.+	F.S.+	20.8
ΔP (psi)	-	-	-	-	3.5
ΔT (°F)	-3.2	-3.2	-2.1	-2.1	-1.6
T_f (°F)	221.9	222.0	186.9	186.5	160.8
T_p (°F)	-	-	-	-	-
$T_f - T_a$ (°F)	128.2	127.2	93.9	94.0	75.5
$T_p - T_a$ (°F)	-	-	-	-	-
$T_i - T_a$ (°F)	129.8	128.8	94.9	95.0	76.3
$(T_f - T_a)/q_i^*$	-333.813	-294.490	-195.622	-159.847	-
$(T_i - T_a)/q_i^*$	-337.933	-298.169	-197.780	-161.596	-
$(T_p - T_a)/q_i^*$	-	-	-	-	-
$n_{q_i}^{**}$	114.714	102.652	60.840	49.515	-
Efficiency	-44.048	-44.343	-29.201	-29.113	-22.98

*°F/BTU/FT².hr**BTU/FT².hr

Figure 14

should exhibit better insulation properties toward higher temperatures, but it does not. Also the counterflow and heat pipe collectors have half the absorber area of the flat plate collector and therefore the emissive losses which are dominant towards higher temperatures should be less for these collectors. A discussion of the difference between expectations and results is given under "Manifold Losses".

3.3 Procedure C: Diurnal Efficiency Measurements at Fixed Angle

All three collector designs are essentially of the non-concentrating type. Their utilization will involve a stationary i.e. non-tracking mode. Therefore, it is essential to investigate their daily performance at a fixed orientation. Such a test is especially useful to understand their optical sensitivity to the diurnal movement of the sun and to diffuse light. Start up and shut down features of the collectors are of further interest for true-life applications.

Because of the identical collector geometry of the heat pipe and the counterflow, a comparison was made only between the heat pipe and the flat plate modules.

The non-tracking testing was performed by adjusting the tubes to Polar North-South orientation as shown in Fig. 10a and 10b. This was done by adjusting the angle between the horizontal and the modular plane i.e., make the slope equal to 34° which is the local latitude at DSET, Inc. and make the surface azimuth angle zero.

The experiments were performed on clear days from sunrise to sunset. Throughout the tests the inlet fluid temperature T_i to both modules was kept constant at 120°F. A typical data sheet for these procedures is shown in Table III. Solar insolation and the energy output of collector modules as a function of solar time are plotted in Fig. 15. Each data point represents a 15 minute interval. Solid black dots on the figure display the solar insolation onto the collector plane. Squares and circles represent the output of flat plate and heat pipe, respectively.

The flat plate collector module responds to the increase in insolation rate remarkably well, whereas the heat pipe is characteristically delayed in its start up and loses heat until at about 7 o'clock. There are two reasons for this slow response: one, the larger thermal inertia of the heat pipe module due to the heavier absorber and manifold material and two, the delay in the start up of the heat pipe at the operating temperature of 125°F. Fluctuations in both collector outputs occurring before solar noon do not seem to be related to insolation conditions. The data indicates a sharp increase in wind velocity up to 8.4 miles/hr from 9:30 to 10:30 solar time. Although evacuated tubes have been shown to be insensitive to winds, manifold losses can be appreciable. This applies more to the heat pipe collector module. The sharp drop in solar flux around 2:00 P.M. causes the flat plate output to decrease, whereas the heat pipe module maintains its output. This is because of the larger thermal inertia of the absorber which supplies the heat necessary for the heat pipe action.

TABLE III

DATE: August 5, 1977
 COLLECTOR: Flat-Plate Tubular
 GLAZING: Single Glass Tube
 APERTURE AREA: 10.94 ft²
 TEST METHOD: Diurnal Efficiency
 Measurements at 34°S

t, solar	09:13	09:28	09:43	09:58	10:13
$\dot{m}$ (lb/hr)	160.66	162.62	158.04	158.62	158.64
T_i (°F)	120.4	120.7	121.6	121.1	120.9
T_o (°F)	134.7	135.9	137.9	138.1	138.4
T_{p1} (°F)	-	-	-	-	-
T_{p2} (°F)	-	-	-	-	-
T_{p3} (°F)	-	-	-	-	-
T_a (°F)	99.6	100.4	101.3	102.6	102.6
Wind Velocity	S 6	SE 2	S 1	SW 7	S 8
Air Over					
Collector (fpm)	300	200	100	400	740
C_p (BTU/lb.°F)	0.99879	0.99881	0.99888	0.99887	0.99887
q_i (BTU/FT ² .hr)	223.76	237.42	250.80	263.05	273.42
q_i , % Diffuse					
Tilt Angle					
Incidence Angle					
Azimuth Angle					
34° SOUTH FIXED ANGLE TEST					
P_{in} (psi)	20.2	20.8	20.0	20.1	20.7
ΔP (psi)	3.4	3.8	3.5	3.3	3.5
ΔT (°F)	14.3	15.2	16.3	17.0	17.5
T_f (°F)	127.5	128.3	129.7	129.6	129.6
T_{p1} (°F)	-	-	-	-	-
$T_{p1}-T_a$ (°F)	28.0	27.9	28.4	27.0	27.0
$T_{p2}-T_a$ (°F)	-	-	-	-	-
$T_{p3}-T_a$ (°F)	20.8	20.3	20.3	18.5	18.3
$(T_f-T_a)/q_i^*$	0.125	0.118	0.113	0.103	0.099
$(T_i-T_a)/q_i^*$	0.093	0.086	0.081	0.070	0.067
$(T_p-T_a)/q_i^*$	-	-	-	-	-
$n_{q_i}^{**}$	210.41	225.526	235.207	246.642	253.040
Efficiency, η	0.939	0.950	0.938	0.938	0.925

*°F/BTU/FT².hr**BTU/FT².hr

Figure 15

The slopes of the curves for both modules are essentially the same throughout the day. This means that the cusp mirror design can optically respond to the diurnal movement of the sun the same way a flat plate does.

An interesting piece of information obtained from this experiment is the daily efficiency of the modules. This quantity can be computed by the formula

$$\eta_{\text{day}} = \frac{\Sigma \dot{Q}_u}{A_a \Sigma q_i}$$

where

$\dot{Q}_u$ = useful energy gain per interval

q_i = insolation onto the solar stand

A_a = effective solar intercept area

$\Sigma \dot{Q}_u$ = total useful gain (output)

Σq_i = total incident solar radiation

Computations covering a time interval of 11.15 hours (from 6:58 a.m. to 6:13 p.m.) on August 5, 1977 gave the following results

$$\Sigma q_i = 2287 \text{ BTU/ft}^2$$

$$\text{Heat Pipe: } \Sigma \dot{Q}_u = 1580 \text{ BTU/ft}^2 \quad (A_a = 11.54 \text{ ft}^2)$$

$$\text{Flat Plate: } \Sigma \dot{Q}_u = 2028 \text{ BTU/ft}^2 \quad (A_a = 10.94 \text{ ft}^2)$$

$$\text{Heat Pipe } \eta_{\text{day}} = 69.1\%$$

$$\text{Flat Plate } \eta_{\text{day}} = 88.7\%$$

It is important to note that these efficiencies have been computed by using the effective solar intercept A_a and the average operating temperature of the modules taken to be 125°F.

3.4 Procedure D: Transient Response Experiment

Thermal response time of collectors has been the subject of previous investigations. Due to the transient nature of insolation and weather conditions, the solar system performance will fluctuate correspondingly.

The objective of this procedure was to observe the transient performance of the collector modules (i.e. heat pipe, counterflow and flat plate) when subjected to sudden solar flux changes during operation.

The tests were conducted for a pair of modules simultaneously and preferably under nearly constant solar flux. This was achieved by tracking the sun on a clear day. For a given inlet temperature the modules were allowed to achieve steady state operation under constant fluid flow and normal insolation conditions. The collector tubes of both modules were then shielded from the sun by wrapping them with several layers of corrugated paper. In this way the air can circulate to a certain degree between the glass envelope and the corrugated paper to keep the temperature of the tube close to ambient. Then the heat output of the modules was monitored as a function of time until it reached the new equilibrium with no solar input. To observe the re-start behavior, the modules were uncovered and allowed to reach the new equilibrium under the momentary flux.

Table IV shows a sample of typical time dependent data taken during the testing of the heat pipe module.

TABLE IV

DATE: August 9, 1977
 COLLECTOR: Heat Pipe
 GLAZING: Single Glass Tube
 APERTURE AREA: 11.54 ft²
 TEST METHOD: Transient Response
 Experiments

Solar Time	ΔT (°F)	C_p (BTU (lbm°F))	$\dot{m}$ (lbm hr)	$\dot{m} C_p \Delta T$ (BTU hr)
11:45	13.2	1.00361	175.99	2331.45
11:46	13.3	1.00359	175.99	2349.07
11:47	13.2	1.00359	175.99	2331.41
11:48	13.1	1.00359	175.99	2313.75
11:49	12.9	1.00361	175.99	2278.47
11:50	12.7	1.00361	175.99	2243.14
11:51	12.6	1.00364	175.99	2225.55
11:52	12.7	1.00361	175.99	2243.14
11:53	12.8	1.00361	175.99	2260.80
11:54	13.1	1.00359	175.99	2313.75
11:55	13.1	1.00359	175.99	2313.75
11:56	13.0	1.00361	175.99	2296.13
- - - - - Covered - - - - -				
11:57	12.6	1.00359	175.99	2225.43
11:58	11.5	1.00351	175.99	2030.99
11:59	10.2	1.00342	175.99	1801.24
12:00	8.9	1.00336	175.99	1571.57
12:01	7.7	1.00325	176.60	1364.24
12:02	6.9	1.00321	176.60	1222.45
12:03	6.1	1.00317	176.60	1080.67
12:04	4.9	1.00314	176.60	868.06
12:05	4.0	1.00312	176.60	708.60
12:06	3.2	1.00312	176.60	566.88
12:07	2.7	1.00310	176.60	478.30
12:08	2.4	1.00310	176.60	425.15
12:09	2.2	1.00306	176.60	389.71
12:10	2.2	1.00304	176.60	389.70
12:11	2.4	1.00302	176.60	425.12
12:12	2.4	1.00297	176.60	425.10
12:13	2.6	1.00291	176.60	460.50
12:14	2.5	1.00289	176.60	442.78
12:15	2.3	1.00287	176.60	407.35
12:16	1.8	1.00287	175.76	317.28
12:17	1.2	1.00289	175.76	211.52
12:18	0.9	1.00289	175.76	158.64
12:19	0.9	1.00289	175.76	158.64
12:20	0.8	1.00289	175.76	141.01
12:21	0.6	1.00289	175.76	105.76
12:22	0.3	1.00289	175.76	52.88
12:23	-0.1	1.00289	175.76	-17.63
12:24	-0.2	1.00291	175.76	-35.25

Results for counterflow-flat plate and heat pipe-flat plate combinations are shown in Figs. 16 and 17, respectively. These graphs show the change in heat output of the modules as a function of time. Time zero, i.e., the moment when the collectors were covered, is different because of technical reasons and has no bearing on the results. Due to the small thermal mass, the output of the flat plate module drops within 2 minutes below zero. At this point the outlet is lower in temperature than the inlet because of the overall losses. The average heat rejection rate from the covered module from 12:04 to 12:13 p.m. was about 105 BTU/hr. This means that the magnitude of the losses is about 3.5% of the steady state gain in the uncovered mode. In the case of the counter-flow module, the average heat rejection rate is about 86.2 BTU/hr between 12:44 and 13:01 p.m., again 3.5 percent of the steady state gain with solar flux on. It is conceivable that most of the losses during the covered mode are due to the manifold rather than the collectors because of the increased glass tube wall temperature. The cooling period for the counterflow module is about 4-5 minutes. The slower drop in the output is due to the larger thermal mass of the counterflow absorber design. The recovery or the start-up again shows the effect of larger thermal mass.

Figure 17 displays the transient behavior of the heat pipe and the flat plate modules. The comparatively very slow drop and start up of the heat output from the heat pipe module is due to the large mass of water (ca. 2.4 gallons or 2 lb per square feet collector) contained in the manifold. Strong mixing

Figure 16

Figure 17

and back mixing causes the manifold to act like an isothermal storage unit. In this case, the time differential equation of the manifold can be approximated as follows:

$$(\dot{m}C_p)_f \frac{d\bar{T}_f}{dt} = A_a F^+ [(S - U_L(\bar{T}_f - T_{amb}))] - (UA)_M (\bar{T}_f - T_{amb}) - (\dot{m}C_p)_f (\bar{T}_f - T_{in})$$

where the index f stands for the heat transfer fluid contained in the manifold. The various terms of this equation are shown in the diagram below.

F^+ is the collector efficiency factor. It is related to the overall heat pipe resistance by the formula

$$F^+ = \frac{1}{1 + \frac{A_a U_L}{A_c U_{HP}}}$$

where

A_a = Effective solar intercept

A_c = condenser surface area

$1/U_L$ = Heat transfer resistance between the absorber surface and the ambient

$1/U_{HP}$ = Heat transfer resistance between the absorber surface and the heat transfer fluid in the manifold.

For a gravity fed heat pipe, the overall conductance U_{HP} drops to zero when the average manifold temperature exceeds the absorber temperature. In this case the collector efficiency factor F^+ vanishes to prevent any heat flow from the manifold to the absorber, thus losing it to the ambient. This thermal decoupling of the collector tubes from the manifold is called the "diode action" and reduces the overall losses from the system.

For the covered module, the above equation can be re-written as follows

$$\frac{d\bar{T}_f}{dt} = - \frac{1}{(\dot{m}C_p)_f} [(A_a F^+ U_L + (UA)_M) (\bar{T}_f - T_{amb}) + (\dot{m}C_p)_f (\bar{T}_f - T_{in})]$$

The absorber starts cooling down following the interruption of the insolation and continues transferring heat to the manifold. Depending on U_L and its heat capacity, the temperature of the absorber drops sometime later below $\bar{T}_f$. At this moment, thermal diode action, decouples the collectors from the heat transfer fluid and the temperature drop in the manifold is now determined by the heat leak to the ambient, the heat loss to the external load and the thermal capacity of the manifold. As shown in Fig. 17, it takes about 27 minutes for the heat pipe module to reach the zero output level, whereas the flat plate needs only 2 minutes. This is

very close to the ratio of the total thermal masses including transfer fluid and absorber material in both modules. This means that the main factor in the cooling rate of collectors is the internal heat capacities. The flat plate starts losing heat at 12:05 but returns to zero output level at 12:13. At low wind velocities the corrugated paper cover protects the module from natural convective cooling and causes a warm-up of the glass tube. This in turn reduces the losses associated with U_L . In contrast to the flat plate and counterflow collectors (see Fig. 16), the heat pipe energy output does not undershoot the zero level on the average. This is due to a combination of two factors. One factor is the thermal diode action that prevents U_L losses from the collector tubes. The other factor is the manifold box which may heat up appreciably when there is no wind as was the case during this test. Thus, the second term in the above equation may be negligably small if the temperature gradient between the box surface and the manifold is reduced below 50°F. As night loss experiments have shown, the total loss for the whole module would then be of the order of 100 BTU/hr. The reason why the heat pipe does not undershoot the zero output level can now be understood as follows. Due to the elevated temperature of the box under no wind conditions, the manifold losses at the given T_f are negligible. Also the thermal diode action of the heat pipes prevents any further losses via collector tubes.

After the heat pipe module was uncovered, the energy output stayed unchanged about 3-4 minutes and then started rising.

The energy absorbed was needed to heat up the heat pipe above that of the heat transfer fluid until the thermal diode was turned on again. The module reached the equilibrium 45 minutes after it was uncovered. A peak and a valley was observed in the energy output after about 30 minutes. A similar fluctuation was also evident during the cool down at 15 minutes after the module was covered. An accurate explanation of this oscillating behavior is not possible without monitoring the temperature profile of the heat pipes as a function of time. It is possible, however that certain regions of the heat pipe temporarily undergo nucleate-boiling causing partial dry out and associated drop in the latent heat transport to the condenser.

A measure of the relative performance of the modules during the transient can be obtained from Fig. 16 and 17 by integrating under the power curves. To be able to compare the energy outputs the same time periods before, during, and after the transient must be considered. This can be done by assuming a constant solar flux* and shifting the heat pipe curve to overlap "time zero" and "start up" for both curves. The integration was carried out over a period of 30 minutes for both collectors including a shaded interval of 10 minutes. However, since the heat pipe was covered for a longer period the limits for the integration had to be constructed.

*This assumption is accurate because of clear sky conditions during the experiment and constant tracking of the sun.

This was done by assuming that the collectors were uncovered at the same time i.e. at 12:12 p.m.. At this point the output of the heat pipe with the adjusted $t=0$ is 450 BTU/hr. The integration was then continued starting with the same value on the recovery branch i.e. at 12:42 by omitting the interval between 12:12 to 12:42. It was found that over the integration period of 30 minutes, the heat pipe delivers about 89 percent of the energy that the flat plate puts out. This value is about 12% better than the ratio of the steady state efficiencies obtained at $\Delta T/q_i = 0.3$ (see, Section 3.1 Procedure A). These results are indicative that under certain transient conditions the heat pipe module can provide performance enhancements over the flat plate collector module. For larger collector surface areas the diode action of the heat pipe can reduce the losses to ambient significantly. These preliminary conclusions however should not be generalized to a true life situation where the load varies depending on the mode of usage and the collectors has to start up daily at sunrise and at several other times on partly cloudy days.

4. MANIFOLD LOSSES

Steady state data shown in Fig. 11 are in contradiction to the results obtained with single collector tubes.³ According to experimental data and theory, the less optically efficient heat pipe should improve its performance relative to the flat plate as the temperature increases. A crossover of the efficiency curves is expected at a $\Delta T/q_i$ of 0.7. The steeper

slope of the heat pipe collector module is largely due to the heavy losses from the manifold to the ambient. To evaluate the parasitic losses associated with the manifold, a multi-mode finite element computer program was used to calculate the thermal losses at various operation temperatures. Figure 18 shows the simplified model of the manifold used for the computation and thermal losses to an ambient at 85°F. The upper solid curve represents the results of the experimentally obtained "night loss" data. The lower curve displays the computed loss data at the same ambient and flow conditions. The dashed curve depicts the night losses minus the computed manifold losses. Even after this correction, the night losses for the heat pipe module seem to be higher than the total losses for the flat plate and counter-flow modules shown in Fig. 14. This result is unexpected because of the thermal diode characteristics of the heat pipes. Parasitic losses from the heat pipe collector should be less as compared to counter/flow, where the fluid has to pass through the cooling absorber. Because of the identical geometry the explanation of these losses lies most probably with the manifold itself or coupling of the collectors to the manifold. Assuming the validity of our theoretical loss computations, the following reasons seem to be plausible to explain the excess losses.

- a) Water condensation in the manifold.
- b) Heat leaks to the ambient via glass-to-metal seals.

Figure 18

- c) Heat leaks to the ambient via collector couplings.
- d) Improper insulation of the external inlet and outlet to the manifold.

Water condensation in the manifold box is a possibility. The box is not built to be hermetic. Consequently certain amounts of moisture in the insulation may have caused an increase in the overall heat conductance of the insulation. This is a difficult heat loss to estimate, however, because of the dry desert climate, it is expected to be negligible. The second possibility, namely that heat may leak to the ambient via the glass-to-metal seal is plausible. Clear skies in the Arizona Desert can enhance the night radiation cooling of the glass envelope ($\epsilon=.92$). Because of the rather large direct coupling of the heat pipe condenser to the glass envelope, leaks may increase correspondingly. Because of the comparable glass-to-metal contact area, this heat leak would have the same magnitude for the counterflow collector. Thus, due to the discrepancy in the loss data, this possibility can be excluded. The heat leaks which may occur through the couplings shown in Fig. 4 were not included in the theoretical calculations for the losses from the manifold. These couplings are in direct contact with the heat transfer fluid and tightly connected to the plug in end of the collector. This feature is unique to heat pipe and may constitute a contribution to the additional losses of this module. Improper insulation of the manifold

inlet and outlet is a fourth possibility. As shown in Fig. 4 the pipes for inlet and outlet are rather heavy and are coupled to the manifold structure which holds the fluid. Since the sensors are outside the box, any cooling of either pipe can enhance the heat losses.

The final performance comparison of the three collector designs is shown in Fig. 19. The dashed curve represent the experimental data for the heat pipe module corrected for the computed manifold losses shown in Fig. 14. The higher intercept of the heat pipe module at $\Delta T/q_i = 0$ is not caused by the optics, but by the contribution of the manifold box as an "active part" of the collector. If this is corrected for heat pipe and counter flow modules exhibit the same performance. The flat plate module has a 20% higher optical efficiency, but its thermal efficiency has the same slope (i.e. losses) throughout the measured range of operating temperatures. This result is against our predictions from the theoretical considerations and single tube experiments. A cylindrical absorber has half the I.R. emitting surface per unit area of flat plate absorber with the same effective intercept area. Towards the higher temperature regions where the Stephan-Boltzman term dominates the losses, a cylindrical absorber should exhibit a better performance. The problem in the heat pipe module appears to be the large manifold compared with the effective intercept area available, plus the accumulative parasitic losses accounted under a), b), c) and d). In the case of the counter-flow it is most likely internal thermal shorting between the concentric fluid channels in the absorber pipe.

Figure 19

5. CONCLUSIONS AND RECOMMENDATIONS

In summary it can be stated that the heat pipe design as a solar absorber in vacuum is a working concept. For the given modules the thermal behavior matches that of the flat plate indicating that heat pipe action is equivalent to flow through performance. However, due to the mirror losses and not being able to make use of the back scattering, the heat pipe converts 20% less solar radiation per unit effective intercept area. With better light collection and the proper manifold insulation, heat pipe collectors should be a very useful and practical device to convert solar radiation into high temperature thermal energy.

The present work presents the design, fabrication, testing and analysis of the evacuated heat pipe collectors. Due to the present stage of development, an economic analysis at this point appears to be much too early. It is important that high pressure problems due to over heating of the heat pipes under runaway conditions be a task of future research in this field. Also components of the collector tube structure, such as the mirror material, condenser design and glass-to-metal seals should be the subject of further investigations. The manifolding of the heat pipe collectors turns out to be an important factor in the applicability of this concept. Further work is necessary to improve the heat transfer from the condensers. Stability of the manifold structure under pressure and the coupling of the collectors to the manifold are worthwhile tasks for the future.

Development of the heat pipe concept to the stage where system applications with and without concentration is very desirable. The advanced system feature can only be fully realized if long term field testing experience is accumulated.

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9. REFERENCES

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