Testing and Analysis of a Heat-Pipe Solar Collector

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We developed an integral heat-pipe/evacuated-tube solar collector in which the inner receiver tubes form the evaporator sections of glass heat pipes. This paper describes both theoretical analyses and empirical tests, comparing the performance of the glass heat-pipe solar collector with one of today's high efficiency evacuated-tube solar collectors. The comparison demonstrates that the performance of the two collectors is effectively identical. The testing and analysis indicate that the glass-wick-type glass heat pipe is an effective heat transfer system for evacuated-tube solar collectors.

Introduction

The purpose of this paper is to describe the testing and analysis of a heat-pipe, heat-removal system for evacuated-tube solar collectors. Several systems have been employed to remove thermal energy from evacuated tubes [1]. Some manufacturers use a U-shaped copper tube running along the inside wall of the collector tube, as in the General Electric collector (Fig. 1), or a straight tube running all the way through as in the Sanyo collector (Fig. 1). In another design, the collector is filled with water through a thin glass tube running down the center line, such as the Owens-Illinois SUNPAK™ (Fig. 1). Some manufacturers have also explored the use of metal heat pipes in evacuated-tube solar collectors.

A heat pipe is usually composed of a sealed tube with a porous wick that lines the inside. The wick is filled with a volatile fluid, and the chamber is filled with its vapor. Heat applied to the evaporator section vaporizes liquid in the wick. The resulting pressure difference in the chamber drives the vapor toward the condenser section. At the condenser end, the vapor condenses and releases its latent heat to an external heat sink. The condensed fluid returns to the evaporator section due to capillary action in the wick or with the help of gravity in a gravity-assisted heat pipe.

Heat pipes, although seemingly complex, have advantages over other heat removal systems. Heat pipes have a well-known advantage of high effective thermal conductance due to their exploitation of phase-change phenomena. In a heat pipe system, the thermal energy is removed entirely at the condenser end of the unit, which reduces the distance the heat removal fluid must flow. With a reduced path length, parasitic pumping energy losses are reduced. Heat pipes have lower thermal mass, which reduces start-up and shutdown time, and the heat-pipe solar collectors, being lighter than water-filled tube collectors, need less structural support. Since the heat pipe is a thermal diode, heat will not be lost when the receiver is cooler than the heat removal fluid, and risk of freezing in the receiver is reduced [2]. Also, the heat pipe is modular in design; it is a single unit facilitating the assembly of a collector array composed of multiple heat pipes.

Several heat-pipe solar collector designs have been tested. Thermacore, Inc., tested and determined that a metal heat pipe, attached to a cylindrical metal fin which fits snugly inside a General Electric TC evacuated-tube solar collector, is cost-effective, although it has not appeared on the market [2]. Philco Italiana, S.p.A., developed a wickless, gravity-assisted, metal heat pipe, attached to a metal fin that fits in the center of an evacuated-tube (Fig. 1). Corning Corporation made an evacuated cylinder with a parabolic cusp reflector inside; at its focus rests a metal heat pipe [3]. The corporation N. V. Philips reported a relatively comprehensive set of tests on a wickless heat-pipe/evacuated-tube collector of their own design [4]. An integral, glass heat-pipe solar collector, with a fiberglass wick in the center, was patented by Feldman in August 1980 [5].

One expects the efficiencies of well-designed metal heat-pipe solar collectors to be similar to those of the evacuated-tube solar collectors on the market. However, metallic corrosion, the resultant decomposition of the working fluid, difficulties of wick fabrication, and pipe-sealing requirements have hindered the use of metallic-wick heat pipes in solar collectors and in other applications [6]. An all-glass heat pipe is corrosion resistant and easily sealed. Finally, we have developed a relatively simple wick fabrication technique.

We have fabricated and tested an integral glass heat-pipe solar collector. It is composed of an evacuated-tube solar collector whose inside glass tube serves as both the absorber of the collector and evaporator of the heat pipe. We bonded a glass-powder wick to the inside, and fused a cylindrical glass condenser section to the open end of an Owens-Illinois, Inc., SUNPAK™ solar collector tube (Fig. 2) [7]. The wick's primary function is to distribute the heat transfer fluid radially around the inside surface. It also prevents the pipe from drying out and overheating, a potential problem for a wickless heat pipe.

During the summer of 1980, we fabricated and tested a prototype glass-evaporator, heat-pipe solar collector at the Solar Energy Research Institute. The prototype, however, had a copper condenser on an Owens-Illinois (O-I) glass evacuated collector tube with a glass bead wick. We measured an overall

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Contributed by the Solar Energy Division for publication in the JOURNAL OF SOLAR ENERGY ENGINEERING. Manuscript received by the Solar Energy Division October 10, 1982.

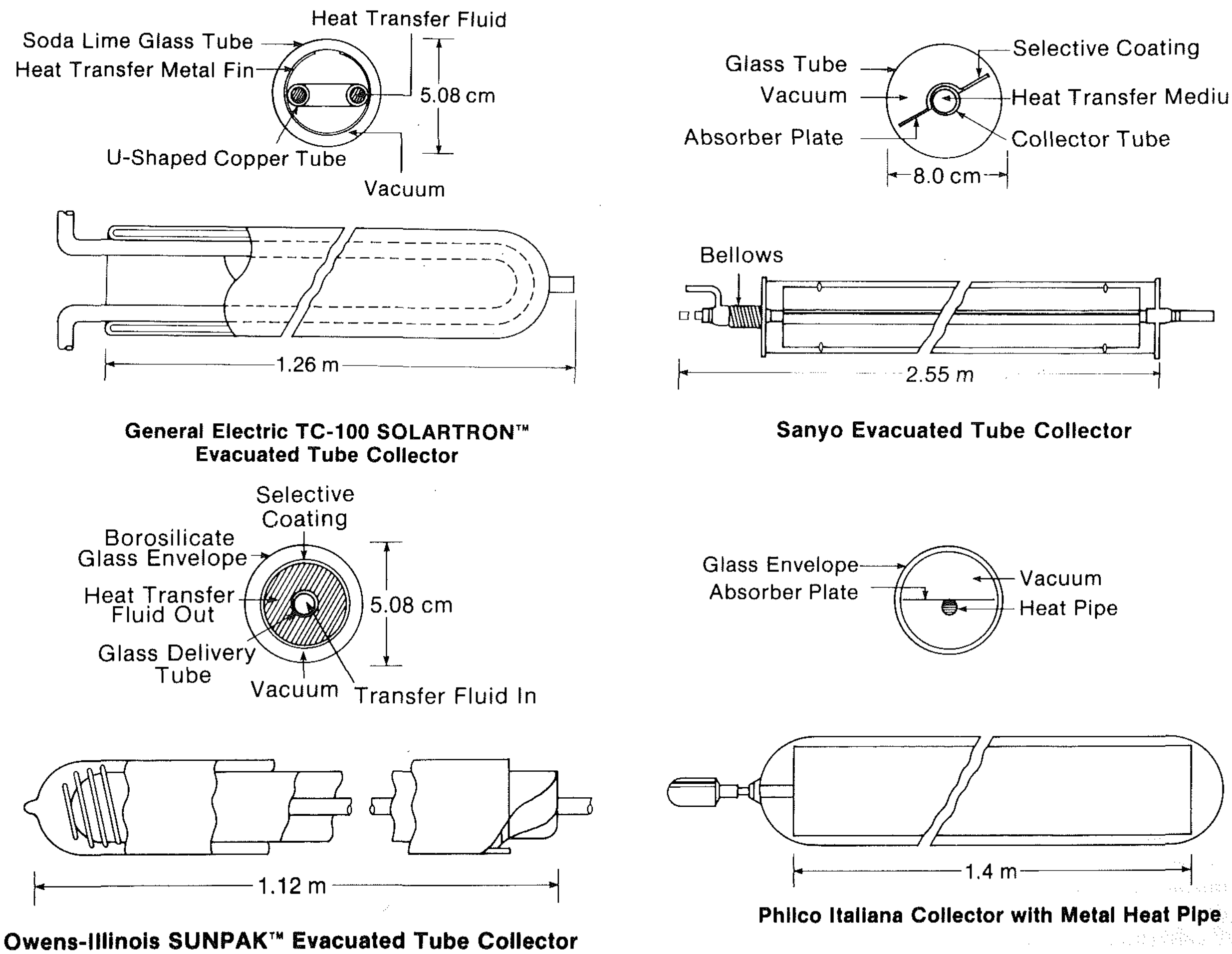
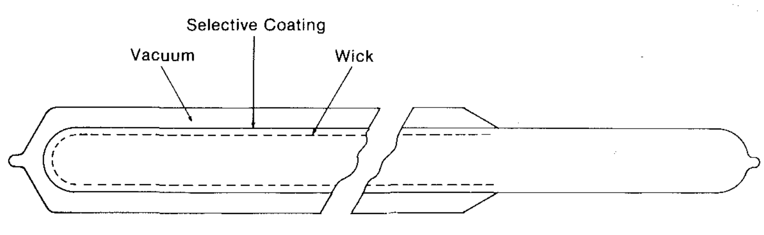


Fig. 1 Heat removal systems for evacuated-tube solar collectors [1]



Integral Glass Heat Pipe Solar Collector

Fig. 2 Integral glass heat-pipe solar collector

efficiency of 55 percent with a time constant of 1 min. The collector looked promising.

Based on the results of the prototype test, we began fabricating and testing an array of eight all-glass, heat-pipe solar collectors. In the following sections, we will present the test results together with a theoretical and empirical analysis of the collectors.

Description and Fabrication of the Collector

We used a glass heat pipe composed of an O-I SUNPAK™ evacuated receiver tube, a wick, a glass heat exchanger/condenser, and methanol as the heat transfer fluid. We made the wick of 270–470 μm soda-lime glass beads, which we adhered to the inside surface of the vacuum tube with a sodium silicate solution (water glass). We then fired the tubes at 573 K for 3 hrs, fusing the wick into place. The condenser section is a 1-mm thick and 20-cm long borosilicate glass cylinder of the same diameter as the open end of the O-I tube. We fused this cylinder onto the lip of the O-I tube and reduced the other end to a 2-mm i.d. We evacuated the inner volume through the small opening in the end and vented in 100 ml of methanol (which is 25 percent more fluid than that which saturates the wick). We then sealed the heat pipe with a flame, fusing the thin entrance shut.

We mounted the heat pipes in the Sunmaster concentrating-parabolic-cusp (CPC) reflectors [8], which were made for the O-I collector tubes, and fabricated a water flow manifold to fit over the condenser sections. Machined from plexiglass, the manifold guided the water circuitously over the condensers in a series configuration (Fig. 3) to maximize heat transfer from the condensers to the water. The primary focus of our tests was on the ability of the glass heat pipe to transport thermal energy to the condenser sections.

We fabricated the original prototype the same way as the all-glass collectors; however, the condenser section was copper and held on with 0-ring seals by the pressure differential between the heat pipe and atmosphere. More fabrication details are published in [9].

Experimental Procedures

We will discuss efficiency measurements made on four days (see Table 1). The first tests (27 August 1980) were run on an individual prototype, several months before testing a full array. We devoted the three test days (12, 13, and 19 January 1981) to the array of eight all-glass heat pipes in the CPC reflectors.

We tested the array with the heat pipes and reflectors in an east-west (E-W) orientation on 12 and 13 January 1981, and in a north-south (N-S) orientation on 19 January 1981. While in the E-W orientation, the heat pipes were mounted at an angle of one degree from horizontal with the condenser end raised to facilitate liquid flow back into the evaporator section after condensation. The array was mounted with the plane of the reflector 60 deg from horizontal, corresponding to normal incidence at solar noon. While in the N-S position the array was also mounted at 60 deg from horizontal, with the heat pipe condensers at the top. The distinction between the E-W and N-S orientations is simply a 89 deg rotation about the solar noon normal incidence with no change in the plane of the array.

We collected data for five variables: manifold inlet fluid temperature, manifold outlet fluid temperature, air temperature, insolation, and water flow rate. We measured inlet and outlet temperatures with T-type immersion thermocouples mounted in the center of the inlet and outlet tubes. We used a shielded T-type thermocouple located in the shade under the collector bank to measure ambient air temperature. We measured insolation with an Eppley precision spectral pyranometer [10] mounted on a surface parallel to the plane of the collector and used a watch and graduated cylinder at the outlet to measure flow rate. We recorded temperatures and insolation on a Kaye datalogger and used a log interval of 10 s during testing.

We calibrated the thermocouples used for inlet and outlet temperatures by placing them in an aluminum block in an insulated container of hot water. As the water temperature slowly dropped, the thermocouple readings were recorded. From these data we generated (by a least squares fit) an equation for the difference in temperature ΔT_o as a function of temperature for the two thermocouples. (In the prototype test, we measured a zero power temperature difference on the covered collector that permitted a correction to the uncovered collector measurements.)

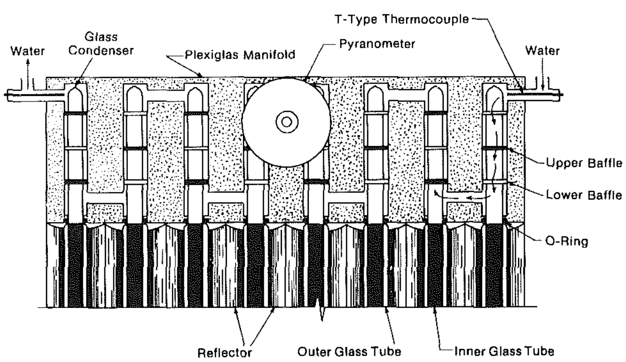


Fig. 3 Integral glass heat-pipe collector array of eight tubes

The efficiency tests consisted of running the system and recording insolation, I, flow rate, $\dot{m}$, ambient air temperature, T_α, and temperature rise, ΔT, across the collector. We repeated this procedure for various flow rates in the range of 8 to 57 g/s. Before each measurement, we ran water through the collector until a steady-state ΔT was reached. After we changed the flow rate, this took 15 to 20 min. We measured the time constant by recording the change of ΔT with time for a constant flow rate after shading the collector during operation.

We calculated precision in our measurements by taking the root mean square of all of the individual uncertainties. The error for absolute temperature measurements with these T-type thermocouples is ± 1.0 K [11]; i.e., any individual thermocouples will be consistently off by some amount within ± 1.0 K. However, at equilibrium, the experimental uncertainty of our temperature measurements was less than ± 0.1 K. Thus, the uncertainty in the measurement of temperature difference between two calibrated thermocouples was less than ± 0.2 K. The relative uncertainty or precision of a ΔT measurement was ± 3.3 percent, based on mean ΔT during testing. (The absolute temperature readouts of all three thermocouples were within 0.7 K of that read on two mercury-and-glass thermometers.) The thermocouples were calibrated both before and after testing, and no drift in the zero point ΔT was observed. Flow rate was measured manually with an uncertainty of ± 0.15 cm^3/s, which yields an accuracy within 1 percent based on mean flow rate. The Eppley precision pyranometer is accurate to within 1 percent [10], and the collector area was estimated to have an uncertainty of less than ± 0.01 m^2 and relative uncertainty of ± 1 percent. For the efficiency, the relative uncertainty is the square root of the sum of the relative uncertainties squared and is equal to ± 3.7 percent.

Precision was dominated by variations in the measurements. Thus, we used the standard deviation (which, in general, exceeded the relative error) as the uncertainty. We calculated standard deviation using the standard form

$$\sigma = \frac{1}{n-1} \sum_{i=1}^{n} (\eta_i - \bar{\eta})^2$$

we computed $\eta_i - \bar{\eta}$ using $\bar{\eta}$ for the particular test day from which the η_i comes. The overall standard deviation in the efficiencies was 5.5 percent.

Results

We calculated the solar noon efficiency (η_n) from the following equation

$$\eta_n = \frac{\dot{m} c \Delta T}{IA} \frac{1}{K_{\tau\alpha}}$$

where

Table 1 Results of the testing of two heat-pipe solar collector configurations: prototype and all-glass array

Test date		No. of heat pipes	Working fluid	Orientation	No. of data points	σ Standard deviation (%)	η mean (%)[a]
8/27/80	Prototype	1	Ethanol	E-W	5		55.0 ± 7.0
1/12/81	1	8	Methanol	E-W	4	4.65	62.1
1/13/81	2	8	Methanol	E-W	13	6.87	53.0
1/19/81	3	5	Methanol	N-S	7	2.61	52.1
	1+2	8	Methanol	E-W	17	6.4	55.2

[a] The overall relative error is ± 3.7 percent, and the overall standard deviation is 5.5 percent (prototype excluded).

$\dot{m}$ = mass flow rate of collector fluid (g/s)
c = heat capacity of collector fluid (J/g K)
ΔT = temperature rise between inlet and outlet (K)
I = insolation (W/m^2)
A = collector area (m^2)
$K_{\tau\alpha}$ = incidence angle modifier: normalizing factor

The temperature rise, ΔT, was calculated from the following equations

$$\Delta T = (T_{\text{out}} - T_{\text{in}}) - \Delta T_o$$

and

$$\Delta T_o = 0.0059(T_m) - 0.047$$

where

T_{out} = measured outlet temperature (K)
T_{in} = measured inlet temperature (K)
ΔT_o = zero power temperature difference between the thermocouples (K)

Incidence angle modifier $K_{\tau\alpha}$ is a function of collector geometry and the effective transmissivity-absorptivity product, $\tau\alpha_e$. Sunmaster derived the $K_{\tau\alpha}$ we used in our calculations by testing their Drainable Evacuated Collector Model DEC-2 [8, 12] in both E-W and N-S orientations. This collector is composed of the same reflector and evacuated tube (the O – I tube) as in our arrays, and therefore has identical external geometry and $\tau\alpha$ product. We therefore assume $K_{\tau\alpha}$ is identical for both collectors. The use of $K_{\tau\alpha}$ in our calculations reduced the measured efficiencies by no more than 10 percent. Table 1 presents the results of the tests described above.

From the data gathered for the time constant (Fig. 4), we generated the following equation by a standard, nonlinear, least-squares fit

$$\Delta T = 3.7e^{-0.365t} + 2.3e^{-0.026t}$$

where t = time (min).

The two exponentials in the equation presumably represent the separate time constants for the heat pipe and the water flow manifold that are 2.74 and 38.47 min, respectively. The flow rate was 35 g/s during the test.

The system time constant is the time at which ΔT decays to $1/e$ of the initial ΔT. The initial ΔT in this case is 6.0°C, and the time constant is 7 min. This system time constant appears to be dominated by the water-filled manifold.

Discussion

In this section, we will compare the performance of the glass heat-pipe solar collector and the Sunmaster DEC-2 solar collector [8, 12]. The DEC-2 Collector is a commercially available, high-efficiency, evacuated-tube collector. First, we will theoretically compare the efficiencies of the receiver ends of the two collectors, and second, we will compare the empirical efficiencies of the two collectors. We will show that the theoretical efficiencies were almost identical, and thus any empirically derived differences in performance are due to factors other than receiver efficiencies.

Theoretical Analysis. Evacuated-tube thermal performance is a function of radiative heat loss from the absorber surface. The effects of convective and conductive heat loss are effectively eliminated by the vacuum jacket around the absorber. Although the absorber is strongly radiatively coupled to the outer glass cover, the outer glass cover experiences convective and conductive heat losses to the ambient air. We assume an absorber heat loss coefficient, U_l of the form [13]

$$U_l = \epsilon\sigma \frac{T_s^4 - T_\alpha^4}{T_s - T_\alpha} \tag{1}$$

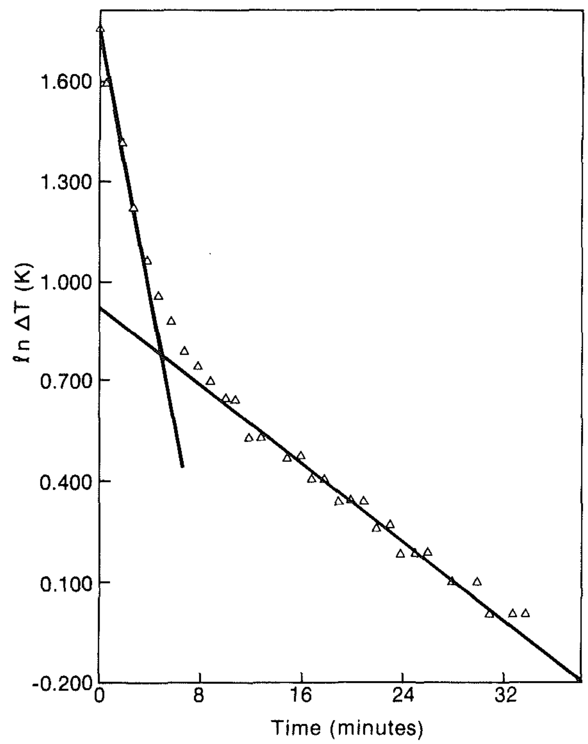


Fig. 4 Time constant data: natural log (ln) of temperature rise across the collector as a function of time

where

ϵ = emittance
σ = Boltzman's constant = 5.67 × 10^{-8} W/m^2 K^4
T_s = absorber surface temperature (K)
T_α = ambient air temperature (K)

We need only consider radiant losses in the receiver end of the collector. Nonetheless, this heat loss is small, since ϵ is only 0.07 [8, 12] when a selective coating is used.

To see inherent differences in the performance of the heat-pipe solar collector and the Sunmaster Model DEC-2 collector, which are externally identical in their receiver sections, we must compare their efficiency curves. The receiver surface temperature will vary between the two collectors, when subject to the same environmental conditions (T_α and heat removal fluid temperature, T_r), if the efficiencies are different. Efficiency depends on U_l, and U_l on T_s. The U_l values and efficiency curves can be derived in the following manner.

Heat transfer between two surfaces is described by the equation

$$Q = (KA\Delta T_s)/t$$

where

Q = energy transfer (W)
K = thermal conductance of material (W/m K)
A = surface area (m^2)
ΔT_s = temperature difference between surfaces (K)
t = thickness (m)

For the Sunmaster collector, the glass receiver tube is the only thermal resistance between the absorber surface temperature and the heat removal fluid temperature.

Thus, the heat transferred to the heat removal fluid can be expressed with the following equation

$$Q = [(K_g A_g)/t_g](T_s - T_r)$$

where g designates glass, T_s = surface temperature (K), and T_r = heat removal fluid temperature (K).

Thus, we see that

$$T_s = Q\left(\frac{t}{KA}\right)_g + T_r$$

Here, Q is the solar energy absorbed on the receiver surface. For simplification, the quantity t/KA, which is a constant for each thermal resistance, will be represented by the letter R with a subscript designating the resistance to which it refers. Thus, we find that

$$T_s = QR_g + T_r \tag{2}$$

By substituting equation (2) into equation (1), we obtain

$$U_l(\text{Sunmaster}) = \epsilon\sigma \frac{(QR_g + T_r)^4 - T_\alpha^4}{QR_g + T_r - T_\alpha} \tag{3}$$

The derivation for U_l of the heat pipe is identical except for the thermal resistances. A heat pipe can be regarded as two thermal resistances (the evaporator wall and the condenser wall) with a thermal diode of uniform temperature in between them.[2] Thus, the total resistance of the heat pipe, R_{hp}, is equal to that of the evaporator wall, R_e [R_e equals resistance of absorber wall (R_g) and wick], plus that of the condenser wall, R_c. We see that

$$R_{hp} = R_e + R_c$$

and

$$U_l(\text{heat pipe}) = \epsilon\sigma \frac{(QR_{hp} + T_r)^4 - T_\alpha^4}{QR_{hp} + T_r - T_\alpha} \tag{4}$$

The efficiency equation for a vacuum-tube solar collector can be expressed in the following form [4, 7]

$$\eta = \eta_o - U_l[(T_s - T_\alpha)/I] \tag{5}$$

for small values of $(T_s - T_\alpha)/I$. η_o is the optical efficiency.

Using values for U_l generated from equations (3) and (4) in equation (5), we generated a plot comparing the efficiency curves for the heat pipe and Sunmaster collectors (Fig. 5).

The difference in the performance of the receiver ends of the Sunmaster and heat-pipe solar collectors is negligible. Where $(T_s - T_\alpha)/I$ is the greatest ($T_s - T_\alpha = 55$ K), the difference in the efficiencies is merely 0.363 percent. This is an expected result, since dU_l/dT_s is small.

Empirical Analysis.

Test results are conventionally presented in both tabular form and in an efficiency curve. The efficiency η curve is based on the following equation [13]

$$\eta = F_r\tau\alpha_e - F_rU_l[(T_{in} - T_\alpha)/I] \tag{6}$$

where

η = collector efficiency
F_r = heat removal factor
$\tau\alpha_e$ = effective transmissivity-absorptivity product
U_l = heat loss coefficient (W/m^{2}K)
T_{in} = inlet fluid temperature (K)
T_α = ambient air temperature (K)
I = insolation (W/m^2)

Equation (6) approximates a straight line in which F_rU_l is the slope and $F_r\tau\alpha_e$ is the η intercept [where $(T_{in} - T_\alpha)/I = 0$]. We do not have data to derive this curve, since all testing was carried out with $(T_{in} - T_\alpha)/I$ too close to zero to calculate U_l. However, F_r can be found by using the values for η at the axis, since $\tau\alpha_e$ is known from Sunmaster data. Thus, we obtain

$$F_r = \eta_o/\tau\alpha_e$$

[2] The vapor between the resistances maintains a uniform temperature because for any temperature gradient in a heat pipe there is a corresponding pressure gradient ($PV = NRT$) that will diffuse at sonic velocity [6].

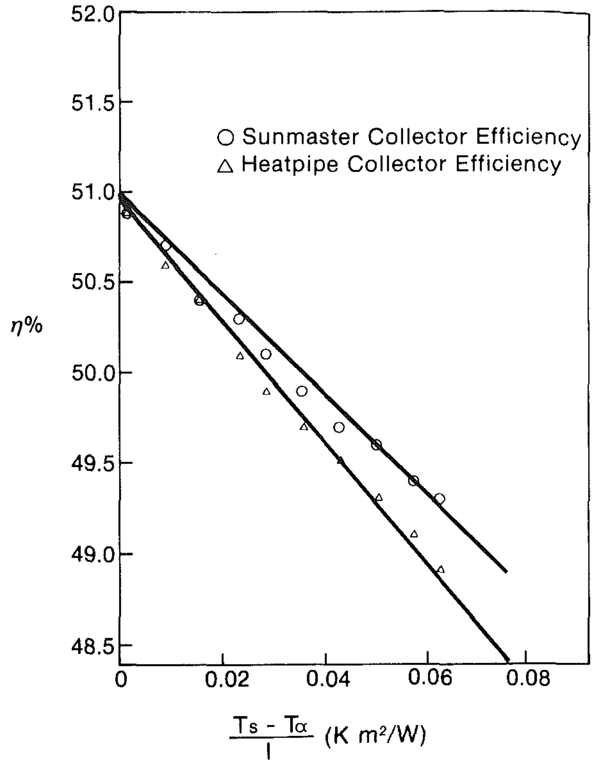


Fig. 5 Theoretical efficiency curve comparison

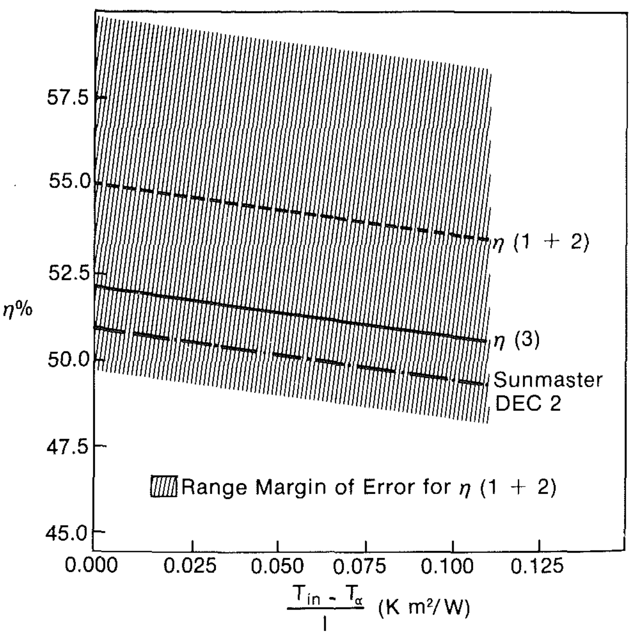


Fig. 6 Comparison of empirically derived efficiency curves

where $\eta_o = \eta$ at $(T_{in} - T_\alpha)/I = 0$, or optical efficiency [1]. In all of these tests, $(T_{in} - T_\alpha)/I$ was sufficiently close to 0 to be negligible. Thus, from the previous equation, we find that F_r(EW) = 0.726 and F_r(NS) = 0.686. EW (east-west orientation) and NS (north-south orientation) represent test days 1/12/81 = 1/13/81, and 1/19/81, tests 1 + 2, and 3, respectively. Results from EW orientation were used together since the collector orientation was the same on both of these days.

For comparison purposes, we will take the heat loss coefficient, U_l, which determines the slope of the efficiency curve, from a previous analysis of the Model DEC-2 Drainable Collector [8, 12]. Thus value is close to any real value that we might expect for our system considering the similarity of the collectors and the theoretical analyses performed earlier. In general, the slopes of evacuated tube solar collector efficiency curves do not vary greatly. Since U_l (Sunmaster) = 0.224, we find that

$$\eta(\mathrm{EW}) = 0.55 - 0.162[(T_{\mathrm{in}} - T_\alpha)/I] \quad (7)$$

$$\eta(\mathrm{NS}) = 0.52 - 0.117[(T_{\mathrm{in}} - T_\alpha)/I] \quad (8)$$

$$\eta(\mathrm{Sunmaster}) = 0.51 - 0.170[(T_{\mathrm{in}} - T_\alpha)/I] \quad (9)$$

Figure 6 is a plot comparing the three efficiency curves, equations (7-9), with the margin of uncertainty of equation (7). The Sunmaster curve is within the lower margin of experimental error. All experimental results agree within experimental error; however, they appear to be cnsistently higher than those of the DEC-2.

Since the difference in theoretical efficiencies is negligible, this analysis implies that with similar manifolds the thermal performance of our system and the Sunmaster system is essentially the same. Therefore, the difference in performance witnessed in the test data probably results from manifold differences, statistical fluctuations in the data, or unidentified systematic errors in testing.

Although this collector performs as well as a commercial high efficiency evacuated-tube collector, a manufacturing cost analysis is still needed to determine its cost effectiveness. In the introduction we described several intrinsic advantages of this collector. These advantages, together with a high efficiency, encourage further investigation into performance and justify a manufacturing cost analysis to determine the system's cost effectiveness.

Conclusion

We have fabricated, tested, and analyzed an integral glass heat-pipe solar collector. This collector has attractive qualities, including low mass, low risk of freezing in the receiver, corrosion-free and less costly materials, low time constant, and modular design. In an array of eight heat pipes oriented in the E-W direction, we determined the optical efficiency to be 55 percent ± 5.5 percent and in the N-S orientation, 52 percent $\pm$ 5.5 percent. These values compare favorably with the 51 percent measured optical efficiency of the Sunmaster DEC-2 drainable collector that exhibits an identical external geometry, $\tau\alpha$ product, and incident-angle modifier. We conclude that the glass-wicked glass heat pipe is an effective heat transfer system for evacuated-tube solar collectors. The intrinsic advantages of an integral glass heat-pipe collector, together with the positive results of this testing and analysis, encourage further research and development to optimize the system of heat pipe receiver, manifold and reflector.

Acknowledgments

We wish to thank the coinventor of the concept, James E. Vansant, of Lawrence Livermore National Laboratory, for his guidance and encouragement. Many researchers at SERI and Hampshire College in Amherst, Mass., have also contributed to this project. We are especially grateful for the support and technical expertise of Joy Barrett, Herb Bernstein, Karl Daninger, Jane Ullman, and Lloyd Williams.

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