ARGONNE NATIONAL LABORATORY
9700 South Cass Avenue
Argonne, Illinois 60439

ANALYSIS OF HEAT-PIPE ABSORBERS
IN EVACUATED-TUBE SOLAR COLLECTORS

by

John R. Hull, William W. Schertz,
and John W. Allen

Renewable Energy Programs

DISCLAIMER

This report was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency thereof, nor any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

February 1986
# CONTENTS

<table>
<thead>
<tr>
<th>LIST OF FIGURES</th>
<th>iv</th>
</tr>
</thead>
<tbody>
<tr>
<td>LIST OF TABLES</td>
<td>v</td>
</tr>
<tr>
<td>NOMENCLATURE</td>
<td>vi</td>
</tr>
<tr>
<td>ABSTRACT</td>
<td>1</td>
</tr>
<tr>
<td>1. INTRODUCTION</td>
<td>1</td>
</tr>
<tr>
<td>1.1 Background</td>
<td>1</td>
</tr>
<tr>
<td>1.2 Use of Heat-Pipe Absorbers in Solar Collectors</td>
<td>3</td>
</tr>
<tr>
<td>1.2.1 General Advantages</td>
<td>3</td>
</tr>
<tr>
<td>1.2.2 Design Concerns</td>
<td>4</td>
</tr>
<tr>
<td>2. ANALYSIS OF CONVENTIONAL FLOW-THROUGH SYSTEMS</td>
<td>5</td>
</tr>
<tr>
<td>3. ANALYSIS OF HEAT-PIPE ARRAYS</td>
<td>8</td>
</tr>
<tr>
<td>3.1 Analysis of Single Heat Pipe Connected to a Manifold</td>
<td>8</td>
</tr>
<tr>
<td>3.2 Analysis of Heat-Pipe Array Connected to a Single Manifold</td>
<td>14</td>
</tr>
<tr>
<td>3.3 Discussion</td>
<td>16</td>
</tr>
<tr>
<td>4. CONDENSER-TO-MANIFOLD HEAT TRANSFER</td>
<td>22</td>
</tr>
<tr>
<td>4.1 Internal Condenser</td>
<td>23</td>
</tr>
<tr>
<td>4.2 External Condenser</td>
<td>26</td>
</tr>
<tr>
<td>4.3 Integrated Storage Tank</td>
<td>29</td>
</tr>
<tr>
<td>4.4 Dependence on Fluid Properties</td>
<td>30</td>
</tr>
<tr>
<td>5. EXAMPLE SYSTEMS</td>
<td>31</td>
</tr>
<tr>
<td>5.1 Past Designs</td>
<td>31</td>
</tr>
<tr>
<td>5.2 ISEC Collector Design</td>
<td>31</td>
</tr>
<tr>
<td>5.3 Absorption Chiller Systems</td>
<td>33</td>
</tr>
<tr>
<td>6. CONCLUSIONS</td>
<td>34</td>
</tr>
<tr>
<td>ACKNOWLEDGMENTS</td>
<td>36</td>
</tr>
<tr>
<td>REFERENCES</td>
<td>36</td>
</tr>
<tr>
<td>Figure</td>
<td>Description</td>
</tr>
<tr>
<td>--------</td>
<td>------------------------------------------------------------------------------------------------------------------</td>
</tr>
<tr>
<td>1</td>
<td>Efficiency of flow-through collector as a function of dimensionless outlet temperature for several dimensionless inlet temperatures, with $F' = 1.0$, $a_0 = 0.7$</td>
</tr>
<tr>
<td>2</td>
<td>Heat flow in a single heat-pipe absorber connected to a manifold</td>
</tr>
<tr>
<td>3</td>
<td>Solar collector array of n heat-pipe absorbers connected to a single manifold (The absorbers are drawn as they might appear in a concentrating solar collector array.)</td>
</tr>
<tr>
<td>4</td>
<td>$F_R$ versus $N_C$ for several $\beta$ at $F' = 1.0$, $n = \infty$</td>
</tr>
<tr>
<td>5</td>
<td>Efficiency of single heat-pipe absorber connected to manifold as a function of dimensionless outlet temperature for several heat transfer ratios $\beta$, with $F' = 1.0$, $a_0 = 0.7$, and $\theta_i = 0.0$</td>
</tr>
<tr>
<td>6</td>
<td>Efficiency of heat-pipe array connected to a single manifold as a function of dimensionless outlet temperature for several values of heat-pipe number $n$, with infinite heat transfer rate to the manifold ($\beta = \infty$), and $F' = 1.0$, $a_0 = 0.7$, and $\theta_i = 0.0$</td>
</tr>
<tr>
<td>7</td>
<td>Efficiency of heat pipe array connected to a single manifold as a function of dimensionless outlet temperature for several values of heat transfer ratio $\beta$, with $F' = 1.0$, $n = \infty$, $a_0 = 0.7$, and $\theta_i = 0.0$</td>
</tr>
<tr>
<td>8</td>
<td>Schematic of internal condenser</td>
</tr>
<tr>
<td>9</td>
<td>Condensation heat transfer coefficient versus heat transfer rate for several condenser areas</td>
</tr>
<tr>
<td>10</td>
<td>Schematic of external condenser</td>
</tr>
<tr>
<td>11</td>
<td>Manifold heat transfer coefficient versus mass flow rate for several manifold diameters</td>
</tr>
<tr>
<td>Table</td>
<td>Description</td>
</tr>
<tr>
<td>-------</td>
<td>------------------------------------------------------------------------------</td>
</tr>
<tr>
<td>1</td>
<td>Fluid heat transfer coefficients for several fluids............................</td>
</tr>
<tr>
<td>2</td>
<td>Estimated heat transfer parameters for several heat-pipe collectors...........</td>
</tr>
<tr>
<td>3</td>
<td>Heat transfer parameters for prototype ISEC with flow-through absorber........</td>
</tr>
<tr>
<td>4</td>
<td>Estimated heat transfer parameters for prototype ISEC with heat-pipe absorber</td>
</tr>
<tr>
<td>5</td>
<td>Pressure drop through different ISEC collector arrays..........................</td>
</tr>
</tbody>
</table>
NOMENCLATURE

A  surface area, m$^2$
 a  cross sectional area, m$^2$
 B  constant
 b  coefficient
 C  heat capacity, J kg$^{-1}$ °C$^{-1}$
 c  coefficient
 D  diameter, m
 e  efficiency
 F  heat transfer factor
 FR collector efficiency factor, which is the actual heat collection rate divided by the heat collection rate attainable with the outer absorber surface at the local mean temperature of the absorber fluid
 FR heat removal factor, which is the actual heat collection rate divided by the heat collection rate attainable with the entire absorber surface at temperature of fluid entering collector
 f  friction factor
 G  heat transfer factor
 g  acceleration of gravity, m s$^{-2}$
 h  heat of vaporization, J kg$^{-1}$
 I  incident solar radiation flux, W m$^{-2}$
 K  thermal conductivity, W m$^{-1}$ °C$^{-1}$
 L  length, m
 m  mass flow rate of manifold fluid, kg s$^{-1}$
 N  number of heat transfer units
 Nu Nusselt number
 n  total number of heat-pipe absorbers in collector array
 ΔP pressure drop, Pa
 Pr Prandtl number
 Q  heat flow, W
 q  heat transfer rate, W
 R  = (UA)$^{-1}$, thermal resistance, °C W$^{-1}$
 Ra Rayleigh number
 Re Reynolds number
 T  temperature, °C
 U  heat transfer rate coefficient, W m$^{-2}$ °C$^{-1}$
 v  velocity, m s$^{-1}$
 W width of optical aperture, m
 y coordinate describing length along manifold tube, m
ANALYSIS OF HEAT-PIPE ABSORBERS IN EVACUATED-TUBE SOLAR COLLECTORS

by
John R. Hull, William W. Schertz, and John W. Allen

ABSTRACT

Heat transfer in evacuated-tube solar collectors with heat-pipe absorbers is compared with that for similar collectors with flow-through absorbers. In systems that produce hot water or other heated fluids, the heat-pipe absorber suffers a heat transfer penalty compared with the flow-through absorber, but in many cases the penalty can be minimized by proper design at the heat-pipe condenser and system manifold. The heat transfer penalty decreases with decreasing collector heat loss coefficient, suggesting that evacuated tubes with optical concentration are more appropriate for use with heat pipe than evacuated or nonevacuated flat-plate collectors. When the solar collector is used to drive an absorption chiller, the heat-pipe absorber has better heat transfer characteristics than the flow-through absorbers.

1. INTRODUCTION

1.1 BACKGROUND

Evacuated-tube collectors have progressed to the point that they are being manufactured by a number of companies (Corning, Hitachi, Philips, Sanyo, Energy Design, O-I Sunmaster) and installed in selected applications in all countries [1,2]. The evacuated collectors out-perform flat-plate collectors at higher temperatures, and have demonstrated greater durability and lifetime. They are currently higher in cost (on a square meter basis) than flat-plate collectors because the tubes are hand made, with minimal tooling for production. Many of the evacuated-tube manufacturers (but not all) use non-imaging optics (CPC) to enhance the effect of the evacuated tubes, particularly for high-temperature operation. The general statement by many of the companies that are making evacuated tubes is that they need a market of about $2 \times 10^5 \ m^2/yr$ production in order to justify mass product equipment expenditures.

Recent research into advanced collectors has shown that a path exists to develop an improved evacuated-tube collector that would have unique advantages over collectors currently on the market [3]. Argonne National Laboratory (ANL) and the University of Chicago (U of C) are working to develop such a collector that will be suitable for mass production. Development of the collector is expected to result in the following:
1. A collector design with a range of applications wide enough to capture a sufficient share of the existing and new markets, so that it could be put into automated manufacture by a company at a production rate of at least $2 \times 10^5 \text{ m}^2/\text{yr}$.

2. The thermal performance of the collector would be greater than conventional (1985 state-of-the-art) evacuated collectors at temperatures above 100°C, equivalent to current flat-plate collectors at temperatures of about 40°C, and competitive with line-focus trackers at temperatures below 250°C.

3. The collector would have an inherently long life because all the critical components, such as mirror and absorber coatings, would be sealed in a vacuum.

4. The successful development of this collector would significantly expand the entire solar heating and air conditioning market, because it can service the same applications as present collectors, plus the higher-temperature applications such as industrial-process heat and solar cooling that present collectors are able to service only marginally.

5. The cost projections for the fully developed collector indicate that the collector should not be any more costly than conventional flat plates, and could be less costly.

The essence of the advanced collector design, which called the Integrated Stationary Evacuated Concentrator (ISEC) [4], is the integration of moderate levels of nonimaging concentration inside the evacuated tube itself. An experimental prototype of the collector has been fabricated and tested at the U of C [5,6]. Analysis of these test results indicates that the efficiency goals of the proposed collector can be readily achieved. A panel of ISEC tubes has been mounted on the ANL solar collector test facility and is being tested to monitor long-term degradation and to measure year-round performance.

Although the performance goals of the advanced collector appear to be in hand, there are a number of problems that deserve further attention before a manufacturer can be expected to commit to large-scale production of the collector tubes. One critical area identified for improvement in the evolution of a manufacturable design is heat-pipe absorber design. The heat transfer design criteria associated with the use of heat pipes in evacuated tubes is the subject of this report. Some of the results have been previously published [7,8].
1.2 USE OF HEAT-PIPE ABSORBERS IN SOLAR COLLECTORS

Heat-pipe absorbers, either wicked or gravity-assisted, have been suggested for flat-plate solar collectors [9-16] and evacuated-tube collectors [9,17-27]. This type of collector has several favorable properties compared with a conventional collector; these are discussed in Sec. 1.2.1. Some of the disadvantages of such a collector are discussed in Sec. 1.2.2.

1.2.1 General Advantages

1. The boiling/condensing heat transfer is very efficient. Large amounts of heat can be moved over long distances with a small temperature difference.

2. In most cases, no external pumping is needed to move heat; i.e., parasitic power requirements are greatly reduced, and the capital expense of the pumping equipment is reduced. The pressure drop in the system is lower because the manifold fluid does not have to pass through the collector. The heat pipe has no moving parts and works quietly.

3. A major fraction of the system can be made freeze-tolerant. The heat pipe fluid can be one that will not freeze under the range of operating temperatures, or one that does not have thermal expansion problems upon freezing or thawing. In addition, many fluids, such as the fluorocarbon refrigerants, are relatively nontoxic and noncorrosive.

4. The heat pipe is self-contained and conducts heat in only one direction. The heat pipe thus acts as a thermal diode, minimizing heat loss when the collector temperature is less than ambient; and in some cases the heat pipe can act as a temperature-limiting device, preventing degradation of the manifold fluid during stagnation. The thermal diode effect, coupled with a lower heat capacity of the absorber and fluid, can result in increased efficiency during periods of transient insolation [12].

5. The heat pipe lends itself to modular design, resulting in particular advantages for evacuated-tube collectors. Installation should be easier. The size of each individual tube is relatively small, and a large number of connections must be made to form an array of sufficient size. The heat pipe opens the possibility for making these connections very easily and inexpensively. With a heat-pipe absorber, each
evacuated tube can be topologically disconnected from the manifold, and the large number of connections in the array can be made more easily [21].

6. Maintenance procedures become more tolerable. Because of the thermal diode effect, if the vacuum in one of the tubes leaks, the tube is not a heat loss. If one of the heat pipes breaks, the system can continue to operate, and in some designs the tube can be replaced while the system is in operation.

7. Connections into the manifold can be made for fool-proof alignment.

8. The heat pipe is connected to the manifold at only one end. This permits free thermal expansion at the other end. In the case of evacuated tube collectors, the expansion can take place inside the glass tube, eliminating the need for bellows.

1.2.2 Design Concerns

The heat pipe absorber system has several disadvantages compared with a conventional flow-through system. One of these is the need for a heat exchange surface between the condenser of the heat pipe and the manifold of the collector. The addition of a heat exchanger in any thermal system always involves a penalty in efficiency, usually treated by incorporation of a heat transfer factor into the efficiency equations [28]. The use of heat transfer factors to describe the effects of the various heat exchange surfaces in a conventional flow-through solar collector is an integral part of the well known Hottel-Whillier-Bliss model (e.g., see References 29 and 30). With heat-pipe absorbers, the constant temperature of the condensation heat transfer at the collector manifold introduces several features into the heat transfer that require slightly different treatment from that of the flow-through system.

Heat pipes may be more expensive than simple tubes. Purity of fluids and noncondensibles are a concern. Degradation of the heat-pipe fluid at high temperatures can cause the formation of noncondensibles, toxics, or corrosives. Reduced heat transfer at the condenser lowers efficiency. For a given manifold temperature, higher collector temperatures are needed, resulting in lower collector efficiency. Of course, the efficiency of the ISEC collector is rather flat as a function of $T_C - T_a$, which helps to mitigate this disadvantage.
2. ANALYSIS OF CONVENTIONAL FLOW-THROUGH SYSTEMS

In this section, we examine the thermal performance of a conventional system that might be used for a solar collector array, utilizing cylindrical absorber tubes and two-dimensional CPC optics. The array of absorber tubes is connected in series, with the same heat transfer fluid passed through each absorber. This is thermally equivalent to one long absorber tube. The thermal efficiency of this system is essentially identical to that of a flat-plate collector, the theory of which has been discussed abundantly in the literature (e.g., References 29-32).

In the standard formalism of the Hottel-Whillier-Bliss model, the steady-state thermal efficiency of a solar collector is

\[ e = F_R \left[ a_o - U_L \left( T_i - T_a \right) / I \right], \]

where \( U_L \) is based on the collector aperture area. We assume that \( U_L \) is constant and ignore axial heat conduction, both in the absorber wall and in the heat transfer fluid. For convenience, the discussion often assumes that the collector is composed of cylindrical absorber tubes and two-dimensional concentrating optics; however, the extension of the presentation to the case of absorber tubes with fins is straightforward.

For a flow-through absorber, heat transfer fluid is flowing at rate \( m \) through a cylindrical absorber tube that is immersed in a field of radiation flux of intensity \( I \). Heat transfer from the radiation field to the collector fluid is a balance between the energy absorbed at the absorber surface of the collector and the heat loss to the ambient from the surface. The thermal conductances of the system are: \( U_L \) the conductance from the absorber surface to the ambient, \( U_w \) the conductance across the wall of the absorber tube, and \( U_f \) the conductance from the inside wall of the absorber tube to the collector heat transfer fluid. In general \( U_L \) will depend on the surface temperature \( T_s \), and \( U_f \) will depend on the mass flow rate \( m \) as well as other hydraulic parameters. In this section we assume they are constant. It is convenient for notational simplification to define these conductances in terms of the area of the optical aperture of the collector.

The radiation intensity \( I \) is assumed uniformly incident on the absorber tube. In reality, the CPC optics creates a nonuniform flux, but the conductivity of the wall is usually large enough that this effect is negligible. The local flux \( I \) at the absorber is concentrated from that of the ambient flux by the CPC optics.

For a segment of absorber tube, the heat transferred to the collector fluid is the difference between the radiation input to that segment vs the heat lost (via reradiation). At the inlet to the array, the collector fluid temperature \( T_C \) is low, resulting in maximum efficiency at this point. The heat balance on a segment of length \( dy \) yields
\[ \dot{m} C_p \frac{dT_c}{dy} = F' \left[ I W \alpha_o - U_L W (T_c - T_a) \right] , \] (2)

where the variables are as defined in the nomenclature section.

\[ F' = \frac{U_o}{U_L} , \] (3)

where \( U_o \) is the heat transfer coefficient between the collector fluid and the ambient, given in this case by

\[ \frac{1}{U_o} = \frac{1}{U_L} + \frac{1}{U_w} + \frac{1}{U_f} . \] (4)

In general, for all advanced collectors \( U_L \) will be small (on the order of \( 1 \text{ W m}^{-2} \text{ °C}^{-1} \)), while \( U_w \) and \( U_f \) will be large in comparison. In this case \( F' = 1 \).

At \( y = 0 \), we have the boundary condition \( T_c(0) = T_i \). Solving equation (2) for \( T_c \),

\[ T_c(y) = T_i \exp(-B y) + [T_a + (I \alpha_o / U_L)] \left[ 1 - \exp(-B y) \right] , \] (5)

where

\[ B = \frac{(F' U_L W)}{(\dot{m} C_p)} . \] (6)

We are assuming in this analysis that \( U_L \) is constant. Strictly speaking, this is not true, as \( U_L \) usually increases with temperature due to increased radiative losses. If the entire length of the collector array is \( L \), then the output temperature \( T_o \) of the array is given by

\[ T_o = T_i \exp(-BL) + [T_a + (I \alpha_o / U_L)] \left[ 1 - \exp(-BL) \right] . \] (7)

The usable heat output of the array \( Q_u \) is given by

\[ Q_u = \dot{m} C_p (T_o - T_i) . \] (8)

In the usual formulation this takes the form
\[ Q_u = F_R A_c \left[ I \alpha_o - U_L (T_i - T_a) \right] , \quad (9) \]

where \( A_c = W L \), is the collector aperture area, and \( F_R \) is the heat removal efficiency factor, which is the actual heat collection rate divided by the heat collection rate attainable with the entire absorber surface at the mean temperature of the fluid entering the collector. In this case \( F_R \) is given by the standard equation from the Hottel-Whillier-Bliss model for a flat-plate collector,

\[ F_R = F' \left[ 1 - \exp\left(-N_c\right) \right] / N_c , \quad (10) \]

where

\[ N_c = BL = (F' A_c U_L) / (\dot{m} C_p) . \quad (11) \]

The parameter \( N_c \) is the number of heat transfer units for the collector (often written \( NTU_c \)). To get the maximum amount of energy from the collector, it is desirable to make \( N_c \) small, i.e., to decrease \( U_L \) or increase \( \dot{m} \). Increasing \( \dot{m} \) will decrease the output temperature according to equation (7). In the limit of small \( N_c \),

\[ F_R = F' \left( 1 - N_c / 2 \right) . \quad (12) \]

We adjust the pump rate \( \dot{m} \), so that for a given \( I \) we always get a given temperature \( T_o \) at the outlet of the array. The thermal efficiency is given by

\[ e = Q_u / (I A_c) = F_R \left[ \alpha_o - U_L (T_i - T_a) / I \right] . \quad (13) \]

Defining the dimensionless temperature parameters

\[ \Theta_i = U_L (T_i - T_a) / I , \quad (14) \]
\[ \Theta_o = U_L (T_o - T_a) / I , \quad (15) \]

and using equation (7), \( N_c \) is given by
\[
\exp(-N_c) = \left(\alpha_0 - \theta_0\right) / \left(\alpha_0 - \theta_1\right) . \tag{16}
\]

Substituting equation (16) into equation (10), the efficiency is given by

\[
e = \left(\alpha_0 - \theta_1\right) / N_c , \tag{17}
\]

\[
e = \frac{F' \left(\alpha_0 - \theta_1\right)}{\ln(\alpha_0 - \theta_1) - \ln(\alpha_0 - \theta_0)} , \tag{18}
\]

where \(\theta_0 > \theta_1\). Note that equation (18) is identical to equation (21) of Reference 32. Efficiency as a function of \(\theta_0\) for several \(\theta_1\) is plotted in Figure 1 for \(F' = 1\).

3. ANALYSIS OF HEAT-PIPE ARRAYS

In this section the effects on thermal performance of the incorporation of a heat-pipe absorber array into a solar collector system are examined. The condenser of each heat pipe is connected to a common manifold, and each heat pipe is hermetically sealed. The manifold fluid is pumped in the usual way, and its temperature rises by an increase in sensible heat due to transfer from the heat-pipe condenser through the manifold wall and into the manifold fluid.

3.1 ANALYSIS OF SINGLE HEAT PIPE CONNECTED TO A MANIFOLD

First, the steady-state thermal performance of a solar collector system with a single heat-pipe absorber is analyzed. The analysis of heat transfer within the heat pipe is simplified. Thermal resistances due to vapor and fluid transport within the heat pipe, as well as heat loss from the manifold to the ambient, are assumed negligible. Consider the geometry of Figure 2, in which the heat pipe is drawn in the form of a tube, as in an evacuated-tube concentrating collector. Radiation input minus heat loss to the ambient over the length of the absorber yields the heat \(Q_{hm}\) transferred to the manifold fluid. The temperature \(T_h\) of the absorber fluid (heat pipe fluid) is constant along the length of the absorber. A heat balance on the absorber yields

\[
Q_{hm} = A_c F' \left[\alpha_0 - U_L \left(T_h - T_a\right)\right] , \tag{19}
\]

where the absorber heat loss coefficient \(U_L\) is defined with respect to the collector aperture area \(A_c\), and the rest of the symbols are defined in the
Fig. 1. Efficiency of flow-through collector as a function of dimensionless outlet temperature for several dimensionless inlet temperatures, with $F' = 1.0$, $\alpha_0 = 0.7$. 
Fig. 2. Heat flow in a single heat-pipe absorber connected to a manifold
nomenclature. In this analysis $U_L$ is again assumed constant. Strictly speaking, this is not true, as $U_L$ usually increases with temperature due to increased radiative losses.

As indicated in Figure 2, the condenser of the heat pipe is attached to the outside of the manifold over a distance $L$. Axial heat conduction is ignored, both in the manifold wall and in the manifold fluid. An important consequence of the condensation heat transfer is that the temperature $T_h$ of the heat pipe fluid is independent of $y$. At any point $y$ along the manifold, the heat transfer is given by

$$m C_p \frac{dT_m}{dy} = \left( \frac{A_{hm}}{L} \right) U_{hm} (T_h - T_m),$$

where $T_m$ is the mean local temperature of the manifold fluid (i.e., the fluid temperature averaged over the manifold cross section at point $y$), and $U_{hm}$ is the total heat transfer coefficient from the condenser to the manifold (which includes condensation, conduction across the manifold wall, and transfer from the inside of the manifold wall to the manifold fluid). For the simple geometry indicated in Figure 2, the ratio $(A_{hm}/L)$ is that part of the manifold circumference that is in thermal contact with the condenser at point $y$. If the manifold wall is a good thermal conductor, this ratio can be taken as the manifold circumference. The mean manifold fluid temperature along the length of the manifold is

$$T_m(y) = T_i \exp(-N_m y / L) + T_h [1 - \exp(-N_m y / L)],$$

where

$$N_m = \frac{(U_{hm} A_{hm})}{(m C_p)}.$$  

(22)

The manifold fluid temperature at the outlet of the segment is

$$T_o = T_i \exp(-N_m) + T_h [1 - \exp(-N_m)],$$

(23)

The heat transferred to the manifold is then

$$Q_{hm} = m C_p (T_h - T_i) [1 - \exp(-N_m)].$$

(24)
Equating equations (19) and (24), and solving for $T_h$,

$$
T_h = \frac{(I \alpha_o / U_L) + T_a + [1 - \exp(-N_m)] T_1 / N_h}{1 + [1 - \exp(-N_m)] / N_h},
$$

where

$$
N_h = \frac{(F' A_c U_L)}{(m C_p)}.
$$

Defining

$$
\theta_h = \frac{U_L (T_h - T_a)}{I},
$$

$$
\theta_1 = \frac{U_L (T_1 - T_a)}{I},
$$

$$
\theta_o = \frac{U_L (T_o - T_a)}{I},
$$

$$
F = [1 - \exp(-N_m)] / N_h,
$$

Equation (25) reduces to

$$
\theta_h = \theta_1 + (\alpha_o - \theta_1) / (1 + F).
$$

Using equations (27)-(31) in equation (23),

$$
\theta_o = \theta_1 + N_h F (\alpha_o - \theta_1) / (1 + F).
$$

The usable heat output of the array is

$$
Q_u = m C_p (T_o - T_1),
$$

and the efficiency is defined in the usual way by

$$
e = Q_u / (I A_c) = F_R (\alpha_o - \theta_1).
$$
Fig. 3. Solar collector array of $n$ heat-pipe absorbers connected to a single manifold (The absorbers are drawn as they might appear in a concentrating solar collector array.)
An energy balance requires $Q_u = Q_{hm}$, so that

$$e = F' \frac{\left( \theta_o - \theta_i \right) }{ N_h} ,$$

$$F_R = F' F / (1 + F) .$$

### 3.2 ANALYSIS OF HEAT-PIPE ARRAY CONNECTED TO A SINGLE MANIFOLD

Continuing the analysis in Section 3.1, examine the system illustrated in Figure 3, in which an array of $n$ heat-pipe absorbers is connected to a single manifold. In this case, the output temperature of the manifold segment that is connected to absorber number 1 becomes the inlet temperature for the manifold segment connected to absorber number 2, this process proceeding along the length of the manifold. In terms of the dimensionless temperatures,

$$\theta_1 = \theta_i + \frac{(N_h/n) F_1 (a_o - \theta_i) }{ (1 + F_1) } ,$$

$$\theta_{j+1} = \theta_j + \frac{(N_h/n) F_1 (a_o - \theta_j) }{ (1 + F_1) } ,$$

where $\theta_j$ represents the dimensionless outlet temperature of the $j$th manifold segment, $\theta_i$ is the dimensionless manifold inlet temperature defined in equation (28), and

$$F_1 = \left[ 1 - \exp(-N_m/n) \right] / (N_h/n) .$$

It is noted that $N_h$ and $N_m$ are defined with respect to the total collector area. The output temperature of the array is

$$\theta_o = \theta_n = \theta_i + (1 - G^n) (a_o - \theta_i) ,$$

where

$$G = 1 - \frac{N_h F_1 / n}{1 + F_1} .$$
It is useful to discuss the results of the analysis in terms of the parameter

\[
\beta = \frac{(U_{hm} A_{hm})}{(U_L A_c)},
\]

(42)

where \(\beta\) is the total heat transfer rate per unit temperature difference from the heat pipe fluid to the manifold fluid divided by that from the heat pipe surface to the ambient. In the limit as \(\beta\) approaches infinity,

\[
G = \frac{1}{(1 + N_h)}.
\]

(43)

Equation (43) can be substituted in equation (40) and \(N_h\) solved in terms of \(\Theta_o\) and \(\Theta_i\):

\[
N_h = 1 - \frac{(\alpha_o - \Theta_i)}{(\alpha_o - \Theta_o)}^{1/n}.
\]

(44)

Substituting equation (44) into equation (35),

\[
e = \frac{F'(\Theta_o - \Theta_i)}{n \left[1 - \frac{(\alpha_o - \Theta_i)}{(\alpha_o - \Theta_o)}\right]^{1/n}}.
\]

(45)

We can use the identity

\[
lm \ x^m = 1 + m \ln(x)
\]

\(m \to 0\)

to derive the efficiency as \(n\) goes to infinity in equation (45). The result is

\[
e = \frac{F'(\Theta_o - \Theta_i)}{\ln(\alpha_o - \Theta_o) - \ln(\alpha_o - \Theta_i)}
\]

(46)

which should be useful for \(\beta > 100\). Note that this is the same efficiency as that for a conventional flow-through system.
For finite $\beta$, a similar procedure can be used in the limit $n$ approaches infinity, resulting in

$$N_h = \frac{(\beta + F')}{\beta} \ln\left(\frac{\alpha_o - \theta_i}{\alpha_o - \theta_0}\right),$$

(47)

$$e = \frac{F' \beta (\theta_o - \theta_i)}{(\beta + F')} - \ln\left(\frac{\alpha_o - \theta_i}{\alpha_o - \theta_0}\right),$$

(48)

which should give good results for $n > 100$.

3.3 DISCUSSION

Figure 4 plots $F_R$ as a function of $N_c$ for several $\beta$ with $F' = 1$. Because evaporation and vapor transport from evaporator to condenser are much more efficient than sensible heat transfer to a moving fluid, $U_f$ should be higher for the heat-pipe absorber than for the flow-through absorber, with a corresponding larger value for $F'$. A flow-through system corresponds to $\beta = \infty$. For a given $N_c$, $F_R$ always decreases with decreasing $\beta$, and the decrease is largest at low $N_c$, where collectors are most efficient. In the limit $N_c = 0$,

$$F_R = \frac{(F' \beta)}{(F' + \beta)}.$$  

(49)

A number of studies have emphasized the advantages of stratified thermal storage [e.g., 31], as well as a control strategy that circulates the storage fluid through the collector only once per day [32,33]. In both cases, it is useful to examine system efficiency on the basis of collector outlet temperature. Such an approach is adopted here, and the results are presented in terms of the dimensionless outlet temperature $\theta_o$.

While efficiency has been given in equations (45), (46), and (48) for several limiting cases, a convenient closed form solution of $e$ as a function of $\theta_o$ is generally not available. The procedure is to choose values of $N_h$ from zero to infinity for a given $\beta$, $n$, $F'$, and $\theta_i$. Equation (40) will yield $\theta_o$, and equation (35) will yield the corresponding $e$. For all of the examples discussed here, $\alpha_o = 0.7$, $F' = 1.0$, and $\theta_i = 0.0$.

For the case of a single heat pipe connected to a manifold ($n = 1$), efficiency as a function of $\theta_o$ for several values of $\beta$ is plotted in Figure 5. It is clear from the figure that, except for large $\alpha_o$, one pays a significant penalty in efficiency for values of $\beta$ below 10. Further, a comparison with the efficiency curve of a conventional flow-through collector (dashed line in
Fig. 4. $F_R$ versus $N_C$ for several $\beta$ at $F' = 1.0$, $n = \infty$. 
Figure 5) indicates that, even with very large $\sigma$, one can never do as well with a single heat-pipe absorber as with the flow-through system. (Because $F' = 1.0$ and the comparison is based on outlet temperature, the flow-through system can have a series, parallel, or combination flow design.) Physically, this result arises because we are forcing the entire heat pipe to operate at a temperature equal to the outlet temperature of the manifold, whereas in the flow-through system only the last segment of the absorber (or absorbers) needs to be at the maximum temperature. The mean temperature of the flow-through absorber is lower, resulting in less heat loss to the ambient.

From the above discussion, one might assume that a single heat-pipe collector is probably not practical. However, there is at least one flat-plate collector on the market that does use a single heat-pipe absorber. In addition to these poorly designed systems, the $n = 1$ analysis is applicable to some systems that are potentially feasible. For example, in a space with limited solar access, a solar collector might consist of an array of heat-pipe absorbers all connected to a single primary manifold. In an effort to maximize $\sigma$ with limited $A_{hm}$, the primary manifold might also be made into a heat pipe, where the evaporative heat transfer in the manifold results in a large $U_{hm}$. The condenser of the primary manifold would lead to a secondary manifold that is not a heat pipe, located away from the collector area, and $\sigma$ at this transfer surface would be made large with a large condenser surface area.

Efficiency as a function of $\theta_{o}$ for several values of $n$ is plotted in Figure 6 for $\sigma = \infty$. The efficiency clearly improves for large $n$. The heat pipe acts as a very-high-conductivity fin attached to the manifold. At large $n$, the fraction of the manifold attached to a single heat pipe is small, and over the total length of the manifold, the penalty due to the constant temperature applied to each manifold segment becomes a small effect. In a large collector array where many heat-pipe absorbers are connected to a single manifold ($n > 100$), the penalty associated with a constant condensation temperature for each heat pipe becomes insignificant. One can analyze the array performance with the standard formalism of the Hottel-Whillier-Bliss model for a flow-through collector. The procedure is to incorporate the thermal resistance from the heat pipe to the manifold into the calculation for $F'$. In a small collector array that might be used for domestic hot water heating, $n = 10$ to 20, and the effect due to a finite number of absorber tubes must be incorporated into the analysis.

Efficiency as a function of $\theta_{o}$ for several values of $\sigma$ is plotted in Figure 7 for the case of a very large array ($n = \infty$). As evident from the figure, it is advantageous to make $\sigma$ as large as possible to maximize efficiency. This may present a challenge in the design of a heat pipe collector array, as can be seen by discussion of each of the terms in equation (42). The collector area $A_c$ is assumed to be predetermined by the energy needs of the load. The value of $U_L$ should be minimized, which suggests that the heat pipe may be more appropriate for evacuated-tube absorbers with low-
Fig. 5. Efficiency of single heat-pipe absorber connected to manifold as a function of dimensionless outlet temperature for several heat transfer ratios \( \beta \), with \( F' = 1.0 \), \( \alpha_o = 0.7 \), and \( \theta_1 = 0.0 \).
Fig. 6. Efficiency of heat-pipe array connected to a single manifold as a function of dimensionless outlet temperature for several values of heat-pipe number \( n \), with infinite heat transfer rate to the manifold (\( \beta = \infty \)), and \( F' = 1.0 \), \( \alpha_0 = 0.7 \), and \( \Theta_1 = 0.0 \).
Fig. 7. Efficiency of heat pipe array connected to a single manifold as a function of dimensionless outlet temperature for several values of heat transfer ratio $\beta$, with $F' = 1.0$, $n = \infty$, $\alpha_o = 0.7$, and $\theta_l = 0.0$.
emissivity selective surfaces than for flat-plate collectors. \( A_{hm} \) should be maximized, but unless optical gaps in the collector array are tolerable, the area of the heat pipe condenser is more or less limited by the width of the collector element associated with each heat pipe. This occurs because it is desirable to keep the surface area of the manifold relatively small in order to minimize the cost of the manifold and its insulation. It is also desirable to maximize \( U_{hm} \). Unless boiling is allowed in the manifold, the limiting thermal resistance in \( U_{hm} \) is likely to be the transfer of heat from the inside surface of the manifold wall to the manifold fluid. If necessary, the value of \( U_{hm} \) can then be increased somewhat by the use of augmented heat transfer methods within the manifold [34]. This effect is discussed in more detail in the next section.

As an example, consider a heat-pipe collector array composed of 2.0 m long evacuated tubes, with an optical aperture width of 10 cm and \( U_L = 0.5 \text{W m}^{-2} \text{C}^{-1} \). Assume that water is flowing with a velocity of 10 cm/s in a 4.0 cm diameter manifold, so that \( U_{hm} = 980 \text{W m}^{-2} \text{C}^{-1} \), and \( A_{hm} = 1.26 \times 10^{-2} \text{m}^2 \), based on the manifold circumference. This yields \( \beta = 120 \). Examination of Figure 7 and equations (46) and (48) indicates that the efficiency in this case is about 1 per cent less than that for a flow-through collector. If, on the other hand, \( U_L = 5.0 \text{W m}^{-2} \text{C}^{-1} \), as might be typical of a flat-plate collector, then \( \beta = 12 \), and the efficiency is degraded approximately 8 per cent compared with the flow-through system. In this case it might be appropriate to increase \( U_{hm} \) by the use of internal fins within the manifold. Note that unless the manifold is very long, it does no good to increase \( A_{hm} \) by widening the flat-plate heat pipe. This requires a reduction in \( n \), and subsequent degradation of efficiency, as indicated in Figure 6.

4. CONDENSER-TO-MANIFOLD HEAT TRANSFER

The analysis in Section 3 showed that the heat transfer ratio \( \beta \) is a key parameter in the thermal performance of a heat pipe collector. For a particular collector design with fixed \( A_c \) and \( U_L \), the difference in performance between flow-through and heat-pipe absorbers is determined by the values of \( U_{hm} \) and \( A_{hm} \). Large values are desired for both parameters to increase \( \beta \); however, in a practical design this desire is tempered by the need to reduce cost and keep the system compact. Typically, a wick is not used inside the heat pipe absorber because it adds significantly to the absorber cost and in general does not improve \( U_{hm} \).

The heat transfer from the heat pipe condenser to the manifold can be divided into three parts: vapor condensation on the inside surface of the condenser wall, heat conduction through the wall, and heat transfer from the wall to the manifold fluid. The thermal resistance \( R_{hm} \) is

\[
R_{hm} = R_{con} + R_w + R_f .
\]
Usually, $R_w$ is small and can be neglected. For example, with a copper wall 1 mm thick, $U_w = 343 \text{ kW m}^{-2} \, ^\circ\text{C}^{-1}$, compared with typical values of $U_{\text{con}} = 20 \text{ kW m}^{-2} \, ^\circ\text{C}^{-1}$ and $U_f = 8 \text{ kW m}^{-2} \, ^\circ\text{C}^{-1}$. However, the relatively thick copper clamping block used in the Philips VTR141 has $R_w = 0.08 \, ^\circ\text{C/W}$, which comprises 1/7 of the total for $R_{\text{hm}}$ [23]. The functional relations for $U_{\text{con}}$ and $U_f$ depend on the design of the condenser attachment to the manifold.

4.1 INTERNAL CONDENSER

One possible design places the condenser wall internal to the manifold, as shown in Figure 8, with heat transfer fluid circulated through the manifold by active pumping or by a thermosiphon effect. Ignoring heat transfer from the end cap of the condenser, the thermal conductance of a single heated cylinder subjected to a cross flow of fluid at a uniform temperature and velocity $v_m$ is given by [35]

$$U_f = \frac{\text{Nu} \, K}{D_h},$$  \hspace{1cm} (51)\

and the correlation for Nu is

$$\text{Nu} = 0.683 \, \text{Pr}^{1/3} \, \text{Re}^{0.466}, \quad 40 < \text{Re} < 4000,$$

$$\text{Nu} = 0.193 \, \text{Pr}^{1/3} \, \text{Re}^{0.618}, \quad 4000 < \text{Re} < 40,000,$$ \hspace{1cm} (52)

$$\text{Nu} = 0.0266 \, \text{Pr}^{1/3} \, \text{Re}^{0.805}, \quad 40,000 < \text{Re} < 400,000,$$

with $\text{Re}$ based on $D_h$, the outer diameter of the heat pipe condenser. The heat transfer can be enhanced by roughening the outer surface of the condenser or by projecting fins from the condenser to increase the surface area. The fins make inserting the condenser into the manifold more difficult and raises the pressure drop in the manifold.

Heat transfer to the inside of a horizontal tube by film condensation is given approximately by [35]

$$U_{\text{con}} = b_{\text{con}} / [D \, (T_v - T_w)]^{1/4},$$ \hspace{1cm} (53)
Fig. 8. Schematic of internal condenser
Fig. 9. Condensation heat transfer coefficient versus heat transfer rate for several condenser areas

\[ A_{\text{con}} = 30 \text{ cm}^2 \]
where \( b_{\text{con}} \) is approximately

\[
b_{\text{con}} = B \left[ \frac{g \rho^2 K^3 h}{\mu} \right]^{1/4},
\]

(54)

where \( B = 0.555 \). The heat transfer coefficient increases slightly with increasing tilt angle from the horizontal. For vertical tubes, \( B = 0.943 \) and in equation (53) \( D \) is replaced by \( L \). Because of the dependence on temperature difference, the coefficient is dependent on the heat flux in the heat pipe, i.e., on the absorbed solar radiation. The thermal resistance of condensation is given by

\[
R_{\text{con}} = q^{1/3} \left[ \frac{D_{\text{con}}}{(b_{\text{con}} A_{\text{con}})^{4/3}} \right].
\]

(55)

Using equation (55) and the definition of \( R \), the value of \( U_{\text{con}} \) as a function of \( q \) at several \( A_{\text{con}} \) is plotted in Figure 9 for water at 90°C as the heat pipe fluid.

As pointed out by de Grijs and de Vaan [24], when film condensation is part of a collector's heat removal, efficiency is dependent on insolation and \( (T_i - T_a) \) independently, and the results of standard methods of testing flat plate collectors give ambiguous results when applied to collectors with heat pipe absorbers, especially if \( R_{\text{con}} \) is a significant fraction of \( R_{\text{hm}} \).

4.2 EXTERNAL CONDENSER

A second design places the condenser wall external to the manifold fluid, as shown in Figure 10, with heat transfer fluid again circulating through the manifold. This configuration may result in an additional wall conductance, but in a good design the thermal resistance will still be dominated by the transfer to the manifold fluid, and equations (50) and (51) will still be valid. Heat transfer from a heated pipe wall to a moving fluid is given by [35]

\[
Nu = 0.023 \Pr^{0.4} \Re^{0.8},
\]

(56)

valid for the range of approximately \( 10,000 < \Re < 100,000 \), where \( \Re \) here is based on the manifold diameter. For \( \Re < 2000 \), in fully developed laminar flow with constant temperature wall surface
Fig. 10. Schematic of external condenser
Fig. 11. Manifold heat transfer coefficient versus mass flow rate for several manifold diameters
\[
\text{Nu} = 3.656 . \quad (57)
\]

For intermediate \( \text{Re} \), the heat transfer is between the values given by equations (56) and (57). Using the correlation of equation (56), \( U_f \) is plotted as a function of \( m \) for several manifold diameters in Figure 11, with water at 90°C used as the manifold fluid. Comparing the coefficient values shown in Figure 11 with those in Figure 9, it appears that in most designs \( U_f \) has the limiting resistance.

The heat transfer area can again be increased, this time by projecting fins from the inside manifold wall into the manifold. This can be done easily at the factory and does not affect the mechanical connection of the condenser to the manifold.

Heat transfer to the outside of a single horizontal tube by film condensation is given by equations (53) and (54), but with \( B = 0.728 \) [35]. The functional dependence of \( U_{\text{con}} \) as a function of \( q \) and \( A_{\text{con}} \) is identical to that in Figure 9, but with the vertical scale increased by a factor of 1.436.

4.3 INTEGRATED STORAGE TANK

A third design integrates the storage tank into the collector array, allowing the heat pipe fluid to condense on a portion of the outer wall of the storage tank. The heat transfer rate coefficient is still given by equations (50) and (51), but the correlation for \( \text{Nu} \) is [35]

\[
\text{Nu} = c_1 \frac{\text{Ra}}{4} , \quad (58)
\]

where \( \text{Ra} \) is based on the tank diameter, and \( c_1 \) depends on geometry but usually is in the range 0.3 to 0.5. Condensation heat transfer is the same as for the external condenser with forced convection.

Assuming that the storage tank is completely mixed, this design does not suffer from the liability of a constant condensation temperature, because the temperature in the manifold (the storage tank is the manifold in this case) is constant. The collector always operates in the asymptotic limit of low \( N_c \), and \( \text{FR} \) is given by equation (49). While the natural convection heat transfer coefficient is relatively small, this is compensated by a relatively large area. Designs that stratify the integrated storage tank are interesting. Because the heat pipe fluid condenses on the coldest part of the tank wall, the collector always interacts with only the coldest storage fluid. However, if only a small fraction of the tank contains the coldest water, the heat transfer area will be correspondingly small.
4.4 DEPENDENCE ON FLUID PROPERTIES

As evident by equation (53), the condensation heat transfer coefficient is directly proportional to the coefficient $b_{\text{con}}$ of the heat pipe working fluid. The heat transfer coefficient $U_f$ is also dependent on fluid properties. Using the correlation of equation (56), equation (51) can be rewritten

$$U_f = b_m v^{0.8} / D_m^{0.2}, \quad (59)$$

where

$$b_m = 0.023 C_p^{0.4} \rho^{0.8} k^{0.6} \mu^{-0.4}. \quad (60)$$

There is considerable variation in $b_{\text{con}}$ and $b_m$ for the typical working fluids and temperatures, as indicated in Table 1. When water is not used as the heat transfer fluid, there is a considerable heat transfer penalty with both coefficients. If water is used as the heat-pipe fluid, there is often the danger of freeze damage. At relatively high temperatures ($T > 90^\circ\text{C}$), where evacuated tubular collectors have a large efficiency advantage over flat plates, and where water is often not used, design for the condenser-to-manifold heat transfer is especially important.

<table>
<thead>
<tr>
<th>Fluid</th>
<th>Temperature, °C</th>
<th>$b_{\text{con}}$, kW m$^{-2}$ °C$^{-3/4}$</th>
<th>$b_m$, kJ m$^{-2.6}$ °C$^{-1}$ s$^{-0.2}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Water</td>
<td>40</td>
<td>9.7</td>
<td>2.30</td>
</tr>
<tr>
<td>Water</td>
<td>90</td>
<td>12.2</td>
<td>3.26</td>
</tr>
<tr>
<td>Water</td>
<td>200</td>
<td>13.1</td>
<td>3.89</td>
</tr>
<tr>
<td>Methanol</td>
<td>50</td>
<td>3.40</td>
<td>0.95</td>
</tr>
<tr>
<td>Methanol</td>
<td>150</td>
<td>3.69</td>
<td></td>
</tr>
<tr>
<td>Ethylene glycol</td>
<td>150</td>
<td>2.87</td>
<td>0.876</td>
</tr>
<tr>
<td>F-113</td>
<td>70</td>
<td>1.52</td>
<td></td>
</tr>
<tr>
<td>Therminol</td>
<td>160</td>
<td>5.17</td>
<td>0.561</td>
</tr>
</tbody>
</table>
5. EXAMPLE SYSTEMS

5.1 PAST DESIGNS

Past collector designs using heat-pipe absorbers have addressed the problem of heat transfer from the condenser to the manifold in different ways. Many designs have used relatively large $A_{hm}$, with condenser lengths of 13 cm [20] or longer [15] placed internal to the manifold, perpendicular to the flow, as indicated in Figure 8. This results in a manifold with a large circumference, which is very costly to effectively insulate. A 15 cm long internal condenser was placed parallel to the fluid flow in a serpentine manifold [22]. In addition to difficulties with insulation, this design has the drawback of increased pressure drop in the manifold at each of the serpentine turns. Other designs have used shorter condenser lengths, but require a relatively large mass flow rate [17,24,25], or internal fins in the manifold [25].

Table 2 lists estimated parameters for several designs for which sufficient information has been reported to perform a calculation of $F_R$. Assuming that panels can always be connected in series, the value for $F_R$ is calculated for $n = \infty$, even though the reported results are for arrays with relatively small $n$. For collector arrays with $n < 20$, $F_R$ would be lower. Even with infinite $n$, these collectors suffer a nonnegligible heat transfer penalty compared with the same collector designed with flow-through absorbers. Note that all values are estimated from experimental prototypes and may not be representative of present commercial collectors.

<table>
<thead>
<tr>
<th>Collector</th>
<th>$U_L$, W m$^{-2}$ C$^{-1}$</th>
<th>$A_c$, m$^2$</th>
<th>$U_{hm}$, W m$^{-2}$ C$^{-1}$</th>
<th>$A_{hm}$, m$^2$</th>
<th>$\beta$</th>
<th>$F_R$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Corning</td>
<td>1.0</td>
<td>0.110</td>
<td>135.</td>
<td>0.0101</td>
<td>12.5</td>
<td>0.93</td>
</tr>
<tr>
<td>VTR141</td>
<td>1.46</td>
<td>0.112</td>
<td>388.</td>
<td>0.0066</td>
<td>15.6</td>
<td>0.94</td>
</tr>
</tbody>
</table>

5.2 ISEC COLLECTOR DESIGN

As an example of the flow-through vs heat-pipe comparison, we consider the integrated stationary evacuated concentrator (ISEC collector) [4-6], under development at the University of Chicago and ANL. Each evacuated tube of the ISEC prototype has a net concentration of 1.64X, and an active aperture area $A_c = 0.0452$ m$^2$. The absorber is a 9.5 mm OD type 314 stainless steel tube with a wall thickness of 0.9 mm and an end to end length of 1.19 m. The prototype panel contains 45 tubes with a net aperture of 2.0 m$^2$. The panel contains 4.0 m of interconnecting tubing with a heat loss per unit length of 0.1 W m$^{-1}$ °C$^{-1}$. At lower temperatures water was circulated; at higher temperatures Therminol-66 was used. A typical flow velocity was 0.5 m/s. If
equation (59) is used to determine $U_f$, the heat transfer parameters for the flow-through case are given in Table 3.

Table 3. Heat Transfer Parameters for Prototype ISEC with Flow-Through Absorber

<table>
<thead>
<tr>
<th>Fluid</th>
<th>$U_L$, W/(m²°C)</th>
<th>$U_W$, kW/(m²°C)</th>
<th>$U_f$, kW/(m²°C)</th>
<th>$F'$</th>
<th>$N_C$</th>
<th>$F_R$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Water (90°C)</td>
<td>0.50</td>
<td>18.5</td>
<td>4.95</td>
<td>0.99987</td>
<td>0.0106</td>
<td>0.995</td>
</tr>
<tr>
<td>Therminol-66</td>
<td>0.76</td>
<td>18.5</td>
<td>0.806</td>
<td>0.9990</td>
<td>0.0343</td>
<td>0.982</td>
</tr>
</tbody>
</table>

In comparison with the preceding analysis, we examine the same array with heat-pipe absorbers connected to the manifold with external condensers. We assume the evacuated tubes are close packed and take $A_{hm}$ to be the product of the manifold circumference and the evacuated tube width (5.08 cm). The goal is to determine the parameter values needed to obtain similar thermal performance with the flow-through design.

It is desirable to leave manifold pipes as small as possible to keep costs down and reduce insulation requirements. As a reference case, we choose a 2.0 cm ID copper manifold with 0.8 mm thick wall. The surrounding condenser wall is also copper and 0.8 mm thick. The resulting heat transfer parameters are shown in the first two lines of Table 4, with a flow velocity of 0.2 m/s in the manifold, $q = 30$ W/tube, $U_f$ the same as in Table 3, and $U_w = 214$ kW m⁻²°C⁻¹. When water is used as the heat transfer fluid, the thermal performance of the two absorber types is nearly identical, although the mass flow rate is four times higher in the heat-pipe collector. However, at higher temperatures the heat-pipe performance is somewhat degraded from that of the flow-through design.

The dominant thermal resistance in an external manifold is the transfer from the inner manifold wall to the manifold fluid. The performance of the heat-pipe design can be significantly improved if this resistance can be lowered. Numerous techniques have been used to augment convective heat transfer in a pipe [34,36-40]. The use of extended surfaces in the form of internal fins appears to offer the best augmentation for our purpose. Internally finned pipes are relatively easy to fabricate, are commercially available, and offer one of the highest heat-transfer/pumping-power ratios. For laminar forced convection, the available results have been summarized by Shah and London [41]. As an example, we assume that internal fins divide the manifold into 12 equal pie-shaped areas. The last line in Table 4 indicates that this is sufficient to yield a value of $F_R$ superior to that for the flow-through absorber.
Table 4. Estimated Heat Transfer Parameters for Prototype ISEC with Heat-Pipe Absorber

<table>
<thead>
<tr>
<th>Fluid</th>
<th>$U_{con}$</th>
<th>$U_f$</th>
<th>$U_{hm}$</th>
<th>$\beta$</th>
<th>$N_c$</th>
<th>$F_R$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Water (90°C)</td>
<td>15.0</td>
<td>1.97</td>
<td>1.73</td>
<td>244.</td>
<td>0.0039</td>
<td>0.994</td>
</tr>
<tr>
<td>Therminol-66 (200°C)</td>
<td>6.36</td>
<td>0.321</td>
<td>0.305</td>
<td>28.7</td>
<td>0.0110</td>
<td>0.960</td>
</tr>
<tr>
<td>Therminol-66 (200°C)</td>
<td>6.36</td>
<td>2.88*</td>
<td>1.96</td>
<td>184.</td>
<td>0.0110</td>
<td>0.989</td>
</tr>
</tbody>
</table>

*With 12 internal fins.

Comparing $F_R$ of the designs listed in Tables 2 and 4, the proposed ISEC heat-pipe collector appears to perform best. This is expected for two reasons. First, the ISEC has a lower $U_L$. Second, the area of the ISEC prototype tube is considerably smaller than that of the designs in Table 2. When the ISEC tube is made longer, as it most likely would be in a commercial design, the performance advantage will decrease.

The pressure drop in a pipe over a length $L$ is given by

$$\Delta P = 0.5 f \rho v^2 L / D.$$ (61)

For laminar flow

$$f = 64 / \text{Re},$$ (62)

whereas for fully developed turbulent flow in a smooth pipe

$$f = 0.184 / \text{Re}^{0.2}.$$ (63)

For the flow-through collector $L = 57.55$ m, and for the heat-pipe collector $L = 2.29$ m. The pressure drops for the several preceding ISEC designs are listed in Table 5. For the designs studied here, the flow-through collector has a pressure drop approximately 400 times larger than the heat-pipe collector and 40 times larger than the heat-pipe collector with an internally finned manifold. This ratio is higher than would be found for other collectors and arises because of the small-diameter absorber tube, necessary for small values of $U_L$.

5.3 ABSORPTION CHILLER SYSTEMS

The use of heat-pipe absorbers is especially attractive when coupled to the generator of an absorption chiller. Here a heat-pipe absorber is connected to a heat-pipe manifold, and the condenser of the manifold is
coupled directly to the coils in the chiller. The evaporative heat transfer in the manifold couples well to the condensation heat transfer in the absorber, and the evaporation in the generator couples well to the condensation within the manifold. The manifold condensate can be returned to the collector array either through a manifold wick, or could even be pumped.

Table 5. Pressure Drop through Different ISEC Collector Arrays

<table>
<thead>
<tr>
<th>Design</th>
<th>Temperature, °C</th>
<th>ΔP, Pa</th>
</tr>
</thead>
<tbody>
<tr>
<td>Flow-through</td>
<td>90</td>
<td>24600.</td>
</tr>
<tr>
<td>Heat pipe</td>
<td>90</td>
<td>60.8</td>
</tr>
<tr>
<td>Heat pipe with finned manifold</td>
<td>90</td>
<td>490.</td>
</tr>
<tr>
<td>Flow-through</td>
<td>200</td>
<td>22000.</td>
</tr>
<tr>
<td>Heat pipe</td>
<td>200</td>
<td>54.4</td>
</tr>
<tr>
<td>Heat pipe with finned manifold</td>
<td>200</td>
<td>532.</td>
</tr>
</tbody>
</table>

When boiling heat transfer is used in the manifold, the U values increase by at least a factor of 10 over convective heat transfer. This results in very high values for Δ and very little temperature drop from collector to generator. This is particularly important in this application, because the efficiency of the new absorption chillers under development at Lawrence Berkely Laboratory increase with increasing temperature, and it is desirable to deliver as high a temperature as possible. It is fortunate that heat pipes are well suited to the evacuated-tube collectors, which are capable of providing the necessary high temperatures to make the chillers more efficient.

In the chiller application, the flow-through absorber suffers a heat transfer penalty. If sensible heat transfer occurs in the manifold at the wall separating the manifold from the generator, the manifold fluid must enter the chiller at a temperature significantly greater than the generator temperature for effective heat transfer. This is analogous to a heat-pipe collector with just one heat pipe, and a substantially larger heat transfer penalty is incurred than that from the reduction in Δ arising from convective, rather than condensation heat transfer.

6. CONCLUSIONS

Analytical methods have been presented that examine the heat transfer factors and predict the thermal efficiency of a solar collector composed of a heat-pipe absorber array connected to a common manifold. Compared with a conventional flow-through absorber with the same collector heat-loss coefficient, the heat-pipe array suffers a heat transfer penalty in most
applications because of the constant-temperature heat transfer at the heat-pipe condenser.

Two key parameters in the analysis are $n$, the number of heat pipes connected to the manifold, and $\beta$, the ratio of heat transfer rate per unit temperature difference from the heat pipe to the manifold divided by that from the heat pipe surface to the ambient. Both $n$ and $\beta$ must be large for the thermal performance of the heat-pipe collector to approach that of an otherwise identical flow-through system. Sufficiently large $n$ and $\beta$ appear to be realizable in practical systems. Because of the low collector heat-loss coefficients, evacuated-tube collectors have larger $\beta$ than their flat-plate counterparts.

When providing thermal energy in the form of a hot fluid, solar collector designs with heat-pipe absorbers always incur a heat transfer penalty compared with designs with flow-through absorbers. However, with careful design at the condenser-to-manifold part of the system, the heat removal factor of a heat-pipe collector can readily equal that of an equivalent flow-through collector. In an analogous manner, when coupled to an absorption chiller, the heat-pipe collector performs better than an equivalent flow-through collector.

The heat-pipe absorber is particularly appropriate for evacuated-tubular collectors. Compared with a flat-plate collector, the heat transfer ratio $\beta$ is larger, due to a smaller $U_L$. Physically, the thermal resistance at the condenser causes the absorber to operate at higher temperature than the manifold. In the evacuated tube, the efficiency as a function of $T_h-T_a$ is relatively flat, and an increase in absorber temperature does not decrease the total efficiency too much. A second reason why heat pipes are most appropriate for evacuated tubes is that the decrease in pumping power, compared with the flow-through case, is greatest.

Several heat-pipe interconnects (i.e., condensers) have been designed and fabricated based on the analysis reported here. They will be tested on several types of manifolds to validate the design criteria that have been developed. Several additional areas associated with the use of heat pipes in evacuated-tubular solar collectors need study and clarification. These problems will be examined in the future as funding permits. Some of these areas are:

1. How does a normal system compare economically, component by component, and installation cost, etc., with projected costs of the CPC designs?

2. Does the heat pipe help regulate unbalanced flow problems in arrays [42,43]?

3. Does the heat pipe change collector heat capacity effect on performance [44-46]?

4. What is the effect of boiling in the manifold? That is, the use of heat pipes in a primary manifold, connected to the
collector heat pipe, with the primary manifold heat pipe condensing into a secondary manifold. This may have advantages in large arrays.

5. What are the design constraints on a horizontal heat-pipe evaporator? This question is especially important for east-west oriented designs. Also, how does the uneven solar flux on the horizontal absorber in the CPC design affect heat-pipe performance?

ACKNOWLEDGEMENTS

This work was supported by the Assistant Secretary for Conservation and Renewable Energy, Office of Solar Heat Technologies, Active Heating and Cooling Division of the U.S. Department of Energy under Contract No. W-31-101-ENG-38.

The author has had many useful discussions with J. O'Gallagher and R. Winston. The author is also grateful to M. Wahlig of LBL for discussion regarding the use of heat-pipe collectors coupled to absorption chillers.

8. REFERENCES


Distribution for ANL-86-16

Internal

J.W. Allen W.T. Sha TIS File (5)
R.L. Cole A.S. Wantroba
E.J. Croke R. Zeno
A.J. Gorski S. Zussman
J.R. Hull (20) ANL Patent Office
A.I. Michaels ANL Contract File
W.W. Schertz (5) ANL Libraries

External

DOE-TIC for distribution per UC-59a (241)
Manager, Chicago Operations Office, DOE-CH
Air Conditioning and Refrigeration Institute, Arlington, VA
C.A. Bankston, Washington, DC
W. Beckman, University of Wisconsin, Madison
B. Brinkworth, University of College, Cardiff, U.K.
Desert Sunshine Exposure Tests, Inc., Phoenix, AZ
G. Ford, Energy Design Corp., Memphis, TN
J.D. Garrison, San Diego State University, CA
S.J. Harrison, National Research Council of Canada, Ottawa, Ontario
N. Kenny, Corning Glass Works, Corning, NY
G.O.G. Lof, Colorado State University, Fort Collins
G. Mather, Owens-Illinois, Inc., Toledo, Ohio
A.E. McGarity, Swarthmore College, PA
E. Morofsky, Public Works Canada, Ottawa, Ontario
F. Morse, U.S. Department of Energy, Washington, DC
D.A. Neeper, Los Alamos National Laboratory, N.M.
J.J. O'Gallagher, Enrico Fermi Institute, The University of Chicago
M. Platt, Sunmaster Corp., Corning, NY
A. Remnitz, Solar Rating and Certification Corp., Washington, DC
C.J. Swet, Mt. Airy, MD
H. Tabor, National Physical Laboratory of Israel, Jerusalem
G. Thodos, Northwestern University, Evanston, IL
B. Wood, Arizona State University, Tempe
M. Yarosh, Florida Solar Energy Center, Cape Canaveral, FL