Solar Energy Vol. 50, No. 6, pp. 491–498, 1993
Printed in the U.S.A.

0038-092X/93 $6.00 + .00


# THERMAL PERFORMANCE OF A FLAT-PLATE HEAT-PIPE COLLECTOR ARRAY

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**Abstract**—This article presents a theoretical model for the determination of the efficiency, the heat removal factor, and the outlet water temperature of a single collector and an array of flat-pipe heat-pipe collectors. The model is validated by on-site testing of 16 heat-pipe collectors under the typical weather conditions in Singapore. Within the operational range of interest of the hot water temperatures, the results show that the proposed model is sufficient to describe the steady-state performance of the collector array.

## 1. INTRODUCTION

A solar collector is characterised by its efficiency and the water temperature that it can provide, and when many collectors are used in a plant, the performance of the array of collectors depends very much on how the collectors are connected together. The most common type of solar collector is a flat-plate collector of flow-through design, where water is heated directly as it flows through the tubes or passages in the absorber surface. The relative merits of this type of collector have been thoroughly discussed in many research papers and textbooks[1–3].

There is another type of solar collector that is increasingly being used. Its absorber consists of one or more heat pipes and the circulating water is heated indirectly by the heat-pipe fluid which condenses in a heat exchanger. Even though the addition of a heat exchanger in a heat-pipe collector can reduce the efficiency of the energy collection, it also has its advantages. A heat-pipe solar collector operates like a thermal-diode where the flow of the heat is in one direction only. This minimizes the heat loss from the hot water when the incident radiation is low. On the other hand, when the maximum-designed temperature of the collector has been attained, it can also cut off its heat transfer capability. Such a feature is useful to prevent the over-heating of the circulating water, a problem often faced by many solar plants[4,5]. It was shown that the number of heat-pipe absorbers and the heat transfer rate at the condenser were the key factors in determining the efficiency of a collector array[6,7], and that a heat-pipe collector has better heat transfer characteristic than a flow-through collector for driving an absorption chiller[8,9].

Heat-pipe collectors can be of either flat-plate or evacuated-tube design. The thermal performance of evacuated-tube collectors is described in references[10–12]. This article is a study of the performance of an array of flat-plate heat-pipe collectors. The array has been in operation for several years, serving a solar-powered air-conditioning system. The performance of the system has been previously reported[13]. The collector array is shown in Fig. 1. An analytical model of the performance of a collector array is developed, and the model is then verified experimentally. The efficiency of the collector at present is also compared with the data initially provided by the manufacturer. The model enables the prediction of the performance of the collector array, when the collectors are connected differently; this information is useful in studying the performance of the current system.

## 2. THEORETICAL ANALYSIS

A flat-plate heat-pipe collector is made up of several heat-pipes. It differs from the conventional flow-through collector in that there is a phase-changing fluid inside the absorber that transfers the heat absorbed by the heat-pipe fluid to the circulating water in a heat exchanger. The evaporators of the heat-pipes together form the absorber of the solar collector, and their condensers are connected to a common heat exchanger manifold at the upper end of the collector. Fig. 2 shows a sectional view of one of the heat-pipes and its heat exchanger. Its evaporator has a black-finned surface. At the evaporator of the heat-pipe, the refrigerant evaporates by absorbing solar energy, then it condenses in the condenser, and transfers the heat to the water that circulates in the heat exchanger manifold.

A single panel of flat-plate heat-pipe collector can be considered as an array of heat-pipes connected to a manifold. If each section of the collector having a single heat pipe can be assumed to have the same heat loss coefficient, $U_L$, to the ambient, then the heat transfer analysis can follow that described by Hull[8]. The outlet water temperature and collector efficiency can be derived as follows. The energy collected by a single heat-pipe, shown in Fig. 2, is the difference between the solar energy absorbed and the heat loss to the ambient over the length of the absorber. The equation for the energy collection is given by

$$Q_{hw} = A_a F' \{ I_t(\tau\alpha)_e - U_L(T_h - T_a) \}. \quad (1)$$

The surface heat loss coefficient between the collector and the ambient, $U_L$, has been assumed constant and $F'$ is the efficiency factor of the collector.

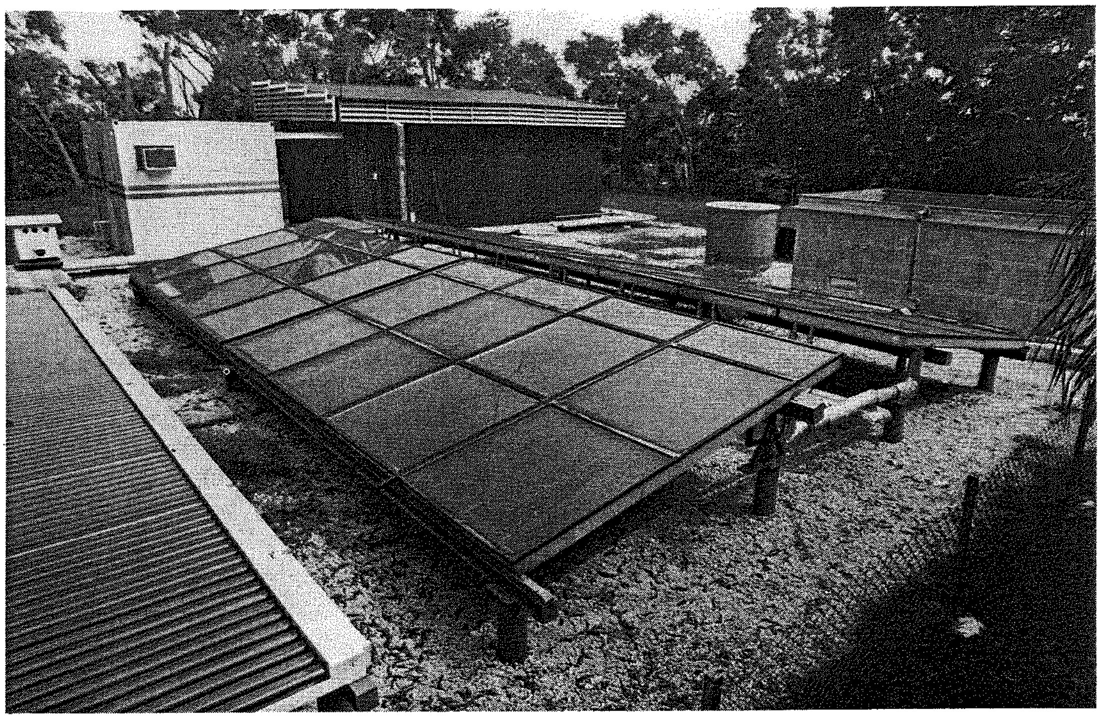

Fig. 1. The heat-pipe flat-plate solar collector array.

Assuming there is no heat loss between the heat exchanger and the ambient, according to energy balance, the amount of heat transferred to the water at the heat exchanger can also be expressed as

$$Q_{hw} = mC_p(T_o - T_i). \qquad (2)$$

If the thermal resistances due to vapor-liquid transport within the heat-pipe are negligible, the temperature of the refrigerant inside the heat-pipe is the same everywhere. The energy gained by water per unit length of the heat-pipe condenser as the water flows over it is given by

$$MC_p \frac{dT_w}{dy} = \left\{\frac{A_{hw}}{L}\right\} U_{hw}(T_h - T_w). \qquad (3)$$

By solving the above equations, the water temperature, after it flows over a single heat-pipe, is

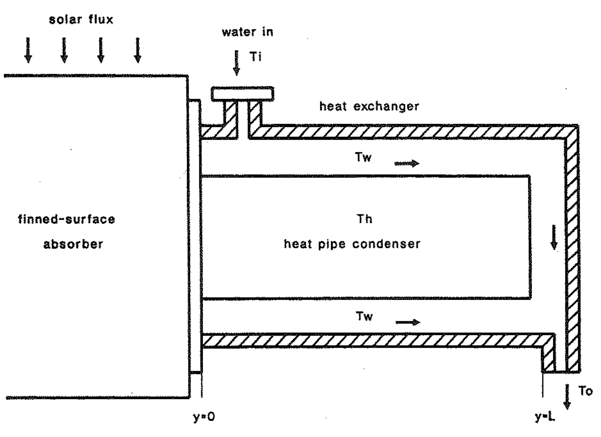


Fig. 2. Single heat-pipe connected to heat exchanger.

$$T_o = T_i\exp(-N_w) + T_h[1 - \exp(-N_w)], \qquad (4)$$

and the heat-pipe fluid temperature can be expressed as follows:

$$T_h = \left\{ \frac{\dfrac{I_t(\tau\alpha)_e}{U_L} + T_a + T_i \dfrac{[1 - \exp(-N_w)]}{N_h}}{1 + [1 - \exp(-N_w)]/N_h} \right\} \qquad (5)$$

where

$$N_h = \frac{(F'A_a U_L)}{(MC_p)}, \qquad (6)$$

$$N_w = \frac{(A_{hw}U_{hw})}{(MC_p)}. \qquad (7)$$

For a collector panel with several heat-pipes, as shown in Fig. 3, the water flows from one heat pipe condenser to another. The outlet water temperature of the first heat-pipe becomes the inlet water temperature of the second pipe if the heat loss of the connecting pipe in between is ignored. For a flat-plate heat-pipe collector having $n$ number of heat pipes, by combining eqns (1), (2), (4), and (5), the outlet temperature of the circulating water can be expressed as

$$\theta_o = (G'')\theta_i + (1 - G'')(\tau\alpha)_e/U_L, \qquad (8)$$

where

$$\theta_i = (T_i - T_a)/I_t \qquad (9)$$

$$\theta_o = (T_o - T_a)/I_t \qquad (10)$$

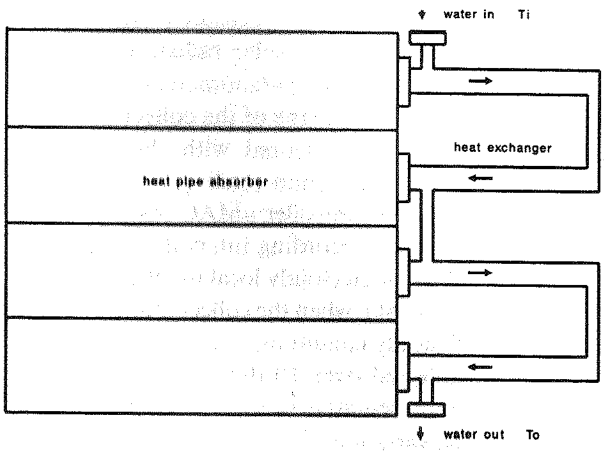


Fig. 3. Heat-pipe array connected to heat exchanger.

$$G = 1 - \frac{(N_h F_1/n)}{(1 + F_1)} \quad (11)$$

$$F_1 = \left[1 - \exp\left(\frac{-N_w}{n}\right)\right] \Big/ \left(\frac{N_h}{n}\right). \quad (12)$$

$N_h$ and $N_w$ are defined in earlier equations. It is noted that these variables are defined in terms of the aperture area and the heat exchange area between the heat-pipe fluid and water of the collector panel. The parameter representing the outlet temperature, $\theta_o$, varies linearly with that representing the inlet temperature, $\theta_i$. When $\theta_o$ is plotted versus $\theta_i$, the slope of the plot is $G^n$ and the $\theta_o$-intercept is $(1 - G^n)(\tau\alpha)_e/U_L$.

The collector efficiency when defined with respect to the gross collector area is as follows:

$$\eta = \frac{Q_{hw}}{I_t A_g}, \quad (13)$$

where the symbols used are as defined in the nomenclature. Substituting eqns (2), (6), and (8) into (13), the efficiency of a flat-plate heat-pipe collector panel then becomes

$$\eta = \left(\frac{A_a}{A_g}\right)\left(\frac{MC_p}{A_a}\right)[(1 - G^n)((\tau\alpha)_e/U_L - \theta_i)], \quad (14)$$

or

$$\eta = \left(\frac{A_a}{A_g}\right) F_R U_L((\tau\alpha)_e/U_L - \theta_i), \quad (15)$$

where

$$F_R = MC_p(1 - G^n)/(A_a U_L). \quad (16)$$

As presented in eqn (16), $F_R$ is the heat removal factor representing the ratio of the actual energy collected to the energy that will be collected if the entire collector area is at the same inlet temperature of the water.

From the above analysis, the efficiency and the outlet water temperature of a collector panel can be determined as long as the design variables of the system are known. The thermal efficiency equation of a flat-plate heat-pipe collector, as given in eqn (15), is similar to that of a flow through flat-plate collector. When the efficiency is plotted against the inlet water temperature, $\theta_i$, the products $\frac{A_a}{A_g} F_R(\tau\alpha)_e$ and $\frac{A_a}{A_g} F_R U_L$ and $(\tau\alpha)_e/U_L$ are the slope, the $\eta$-intercept and the $\theta_i$-intercept, respectively of the efficiency line.

## 2.1 *Collectors in series*

For collectors in series, heat-pipe or otherwise, the outlet water temperature of a collector becomes the inlet water temperature of the following collector. If the heat losses of the interconnecting pipes can be neglected, the outlet water temperature of $N$ number of flat-plate heat-pipe solar collector connected in series, with each panel of collector having $n$ number of heat-pipes, can be determined by rewriting eqn (8) as follows:

$$\theta_o = (G^n)^N \theta_i + (\{1 - G^n\})^N (\tau\alpha)_e/U_L. \quad (17)$$

In the case of two collectors in series, the efficiency of the first collector is

$$\eta_1 = MC_p(T_{o1} - T_i)/(A_g I_t)$$
$$= \left(\frac{A_a}{A_g}\right) F_{R1}[(\tau\alpha)_e - U_L(T_i - T_a)/I_t], \quad (18)$$

and the efficiency of the second collector is

$$\eta_2 = MC_p(T_{o2} - T_{o1})/(A_g I_t)$$
$$= \left(\frac{A_a}{A_g}\right) F_{R1}[(\tau\alpha)_e - U_L(T_{o1} - T_a)/I_t], \quad (19)$$

where $F_{R1}$ denotes the heat removal factor of a single panel. By combining eqns (18) and (19), it can be shown that the efficiencies $\eta_2$ and $\eta_1$ are related by

$$\eta_2 = \eta_1[1 - A_g U_L F_{R1}/(MC_p)], \quad (20)$$

or

$$\eta_2 = \eta_1(1 - K), \quad (21)$$

where

$$K = (A_g F_{R1} U_L)/(MC_p). \quad (22)$$

$K$ is a sole function of the design parameters of the single panel only. The combined efficiency of the two-in-series array is simply given by the arithmetic average of the efficiency of each collector, i.e.,

$$\eta = MC_p(T_{o2} - T_i)/(2A_g I_t) = (\eta_1 + \eta_2)/2, \quad (23)$$

and it can be simplified to an expression containing the relevant parameters of the single collector, i.e.,

$$\eta = \eta_1[1 - A_g U_L F_{R1}/(2MC_p)], \tag{24}$$

or

$$\eta = \left(\frac{A_a}{A_g}\right) F_R U_L[(\tau\alpha)_e/U_L - (T_i - T_a)/I_t], \tag{25}$$

where

$$F_R U_L = F_{R1} U_L\left[1 - \frac{K}{2}\right]. \tag{26}$$

Duffie and Beckman [1] quote the result of Oonk *et al.* for the heat removal factor of $N$ identical flat-plate flow-through collector panels placed in series,

$$F_R = F_{R1}[1 - (1 - K)^N]/(NK) \tag{27}$$

where $K$ is as defined in eqn (22). Since the expression for the efficiency of a flat-plate heat-pipe collector as given by eqn (15) is exactly similar to that of a flat-plate flow-through collector, eqn (25) is also applicable to the flat-plate heat-pipe collectors.

### 3. EXPERIMENTAL VERIFICATION OF MODEL

An experiment was conducted using (a) the array of solar collectors of a solar air-conditioning plant and (b) the results presented following the expression given by eqns (17) and (25). The collector array consists of 16 flat-plate heat-pipe collectors, each having six heat-pipes ($n = 6$). They are arranged 2-in-series ($N = 2$) and having 8 parallel flow paths. The collectors are inclined at an angle of 14° from the horizontal, and each collector has a gross area of 3.943 m$^2$ and an absorber area of 2.868 m$^2$. The total flow rate through the collector was maintained constant at 0.684 kg/s. The values of five variables were recorded: (a) temperature of the water entering the collector array, $T_i$; (b) temperature of the water leaving collector array, $T_o$; (c) ambient temperature, $T_a$; (d) total incident solar radiation on the collector $I_t$; and (e) total water flow rate. Temperatures were measured with the RTD sensors. The total incident solar radiation was measured with black and white pyranometers mounted on a surface parallel to the plane of the collector, and the water flow rate was measured with the rotameter. Temperatures and insolation readings were recorded by a programmable controller $\mu$MAC-5000 and a microcomputer, with a recording interval of 5 s.

Because of the often cloudy local weather condition, only those sets of data, when the collector array operates under a quasi-steady condition, were used in the study. Data were collected over 10 days when the weather was good. A representative sample set of data is shown in Table 1; the variation of the incident solar radiation was within ± 10 W/m$^2$ over a period of 10 min, and that of the ambient temperature was within ± 0.5°K over the same period. The inlet temperature to the collector array was maintained within ± 0.1°K. The outlet temperature of hot water ranged from 45°C to 85°C. The effect of wind velocity is not considered in these tests but the local meteorological records show that the range of wind speeds for the test periods typically ranged from 0.5 to 2.9 m/s.

In this experiment (Table 1), it was assumed that the water in circulation was uniformly distributed over the eight parallel paths, with each path having 0.085 kg/s of water flowing through the two collectors connected in series. Therefore, the outlet water temperature and collector efficiency that were determined were the average values of the 2-in-series collectors over the eight parallel paths. Fig. 4 is the efficiency plot. It can be seen that the efficiency varies linearly with $(T_I - T_o)/I_t$. The regressed efficiency equation is found from the tests to be as follows:

$$\eta = 0.4432 - 2.855\,\frac{(T_i - T_a)}{I_t}. \tag{28}$$

The uncertainty of the measured parameters will affect the precision of the efficiency. It is therefore necessary to examine the possible error of each measurement to evaluate the reliability of the experimental results. For the temperature measurement, both of the RTD sensors together with the transmitter were calibrated before testing. The error for each absolute tem-

Table 1. Sample of the experimental data listing for the evaluation of one efficiency data point of June 7, 1991

| Time | $I_t$ (W/m$^2$) | $T_i$ (°C) | $T_o$ (°C) | $T_a$ (°C) | $\theta_i$ | $\theta_o$ | $\eta$ (%) |
|---|---|---|---|---|---|---|---|
| 13:50 | 824.8 | 78.81 | 83.24 | 33.82 | 0.0545 | 0.0599 | 27.90 |
| 13:51 | 824.2 | 78.79 | 83.26 | 33.31 | 0.0552 | 0.0606 | 28.24 |
| 13:52 | 824.4 | 78.78 | 83.28 | 33.09 | 0.0554 | 0.0609 | 28.40 |
| 13:53 | 823.1 | 78.79 | 83.28 | 33.21 | 0.0554 | 0.0608 | 28.35 |
| 13:54 | 820.6 | 78.79 | 83.27 | 33.11 | 0.0556 | 0.0611 | 28.40 |
| 13:55 | 817.7 | 78.79 | 83.26 | 33.16 | 0.0558 | 0.0613 | 28.40 |
| 13:56 | 818.2 | 78.81 | 83.26 | 33.12 | 0.0558 | 0.0613 | 28.34 |
| 13:57 | 819.4 | 78.82 | 83.25 | 33.15 | 0.0557 | 0.0611 | 28.10 |
| 13:58 | 817.9 | 78.84 | 83.25 | 33.25 | 0.0557 | 0.0611 | 28.04 |
| 13:59 | 816.8 | 78.85 | 83.25 | 33.10 | 0.0561 | 0.0614 | 28.01 |
| Average | 820.6 | 78.81 | 83.26 | 33.23 | 0.0555 | 0.0609 | 28.20 |

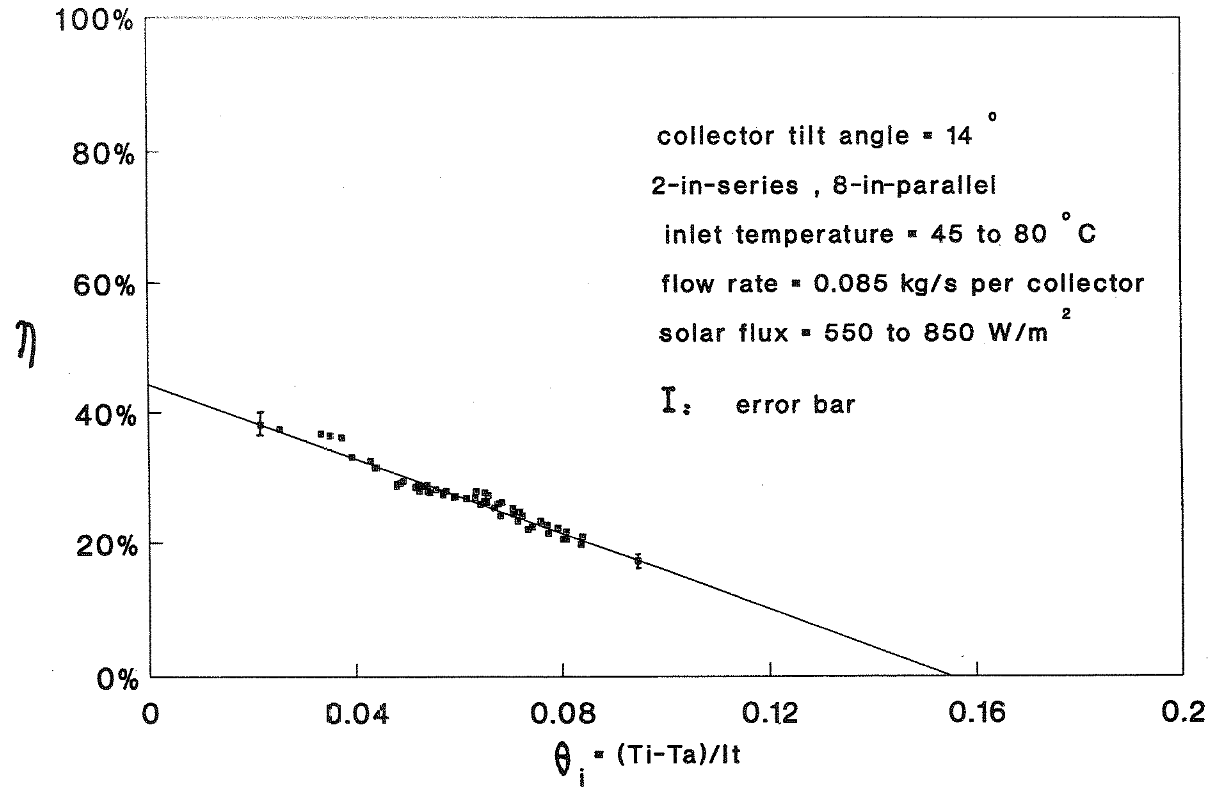


Fig. 4. Thermal efficiency of heat-pipe collector array.

perature measurement was found within 0.10°C. Thus, the uncertainty of the temperature difference is 0.2°C. The relative uncertainty is estimated to be 4.4% for the temperature difference across the collectors, 1% for the difference between the collector inlet temperature and the ambient temperature. The pyranometer was calibrated with a Eppley Precision Spectral Pyranometer, which is accurate to within 1%, so the uncertainty of the black and white pyranometer is 2%. For the rotameter, since it was scaled at 20°C of the water temperature, the actual value of the flow rate must be modified according to the water density at the operating temperature. The relative uncertainty of the flow rate is within 3%. For the efficiency, the relative uncertainty is the square root of the sum of the square of the relative uncertainties [14] and this is estimated to be equal to 5.7%.

From eqn (25), it can be deduced that for 2-in-series collector

$$\left(\frac{A_a}{A_g}\right)F_R(\tau\alpha)_e = 0.4432, \tag{29}$$

$$\left(\frac{A_a}{A_g}\right)F_R U_L = 2.855, \tag{30}$$

and

$$(\tau\alpha)_e / U_L = 0.1552. \tag{31}$$

Since the area ratio, $A_g/A_a = 0.7273$, the variables of interest for the test configuration therefore are:

$$F_R(\tau\alpha)_e = 0.6093, \tag{32}$$

$$F_R U_L = 3.925\left(\frac{\mathrm{W}}{\mathrm{m^2\,K}}\right). \tag{33}$$

Figure 5 is a plot of the dimensionless outlet water temperature, $\theta_o$, versus the dimensionless inlet water temperature, $\theta_i$. It can be seen that $\theta_o$ varies linearly with $\theta_i$.

## 4. DISCUSSION

In the ASHRAE standard 93-1986 [15], there is a section that recommends that the performance of heat-pipe collectors be presented by the following equation,

$$\eta = c + d(T_i - T_a)/I_t + e/I_t + \frac{f(T_i - T_a)^2}{I_t}, \tag{34}$$

where $c$, $d$, $e$, and $f$ are coefficients to be obtained from a test. However, from the results of the present experiment, it appears that a linear relationship, using the first two terms of the equation is suffice to describe the thermal performance of the flat-plate heat-pipe collectors under consideration. The absence of parabolic behavior in the results shows that the effect of heat losses by radiation from the collectors is probably insignificant at this temperature range. The variation of the outlet water temperature with respect to the inlet water temperature can be determined from the infor-

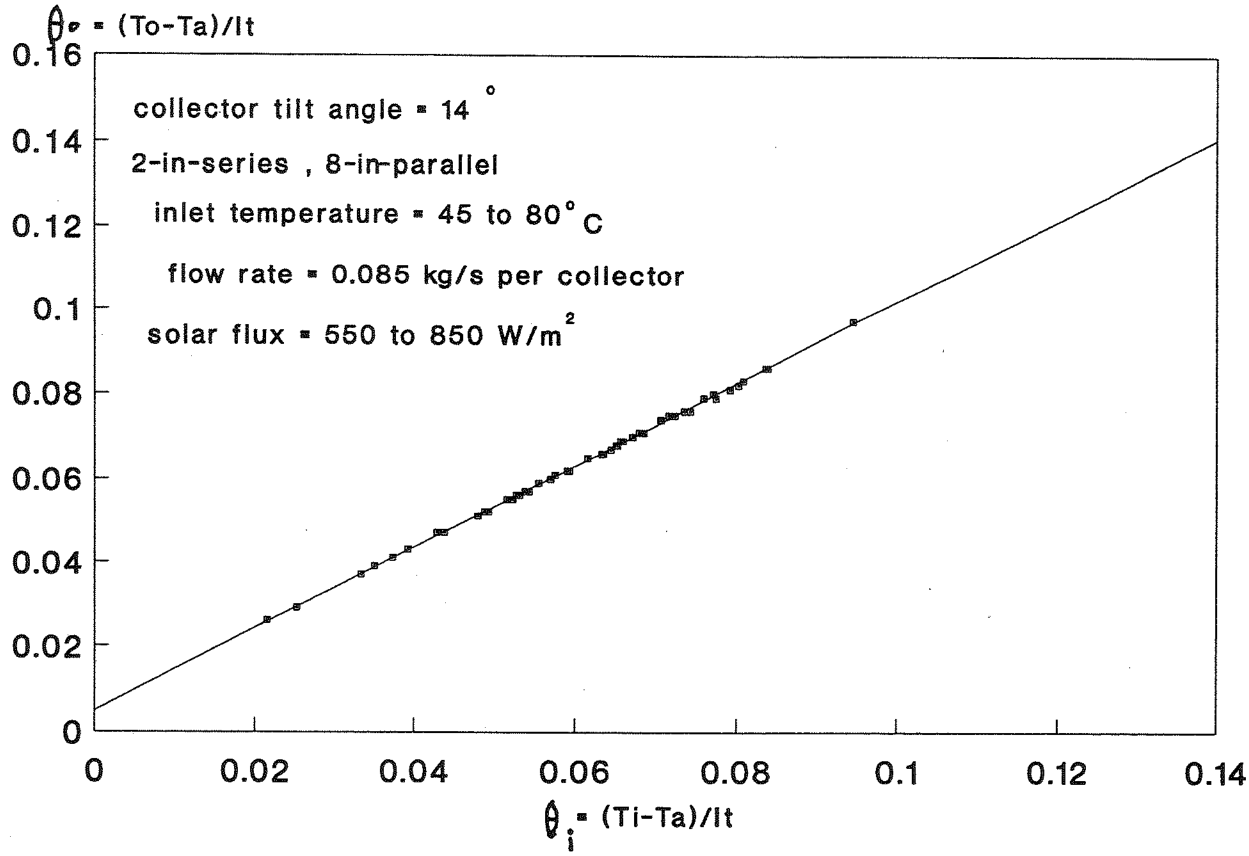


Fig. 5. Outlet temperature of heat-pipe collector array.

mation derivable from the efficiency plot and the flow rate of water per unit area, $M/A_a$, of each collector. The efficiency plot of the 2-in-series collector provides information on the value of $\tau\alpha/U_L$ (=0.1552) and $F_RU_L$ (=3.925). With the value of $F_RU_L$ known, $(G^n)^2$ can be determined from eqn (17), and it is found to be 0.9683 in this instance. Values of $(G^n)^2$ and $\tau\alpha/U_L$ are all that are required for the plot.

The performance of a single flat-plate heat-pipe collector can be deduced from that of a 2-in-series collector, as outlined in the theoretical model. The values of $\tau\alpha/U_L$ in both cases are the same. The $n$-dependency of the model is not verified in the tests because it would involve the replumbing of the collectors. The value of $F_RU_L$ for a single collector can be determined from eqn (26), and in this instance $F_{R1}U_L$ is found to be 4.1125 W/m$^2$ K. The value of the vertical intercept of the efficiency plot, is therefore 0.4642. Based on these values, the efficiency of a single collector is plotted in Fig. 6. The efficiency curve as provided by the manufacturer was plotted versus the difference between the average water temperature and the ambient temperature with the incident solar radiation as the parameter[16]. The data at the relevant radiation value are also plotted in Fig. 6 for comparison. The wear and tear over the past eight years has affected the collector

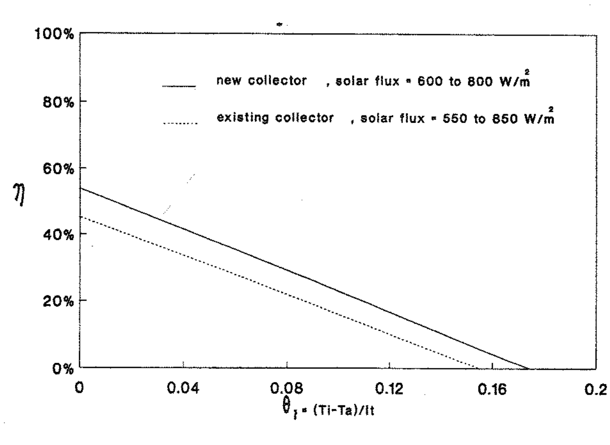


Fig. 6. Comparison of single collector's efficiency.

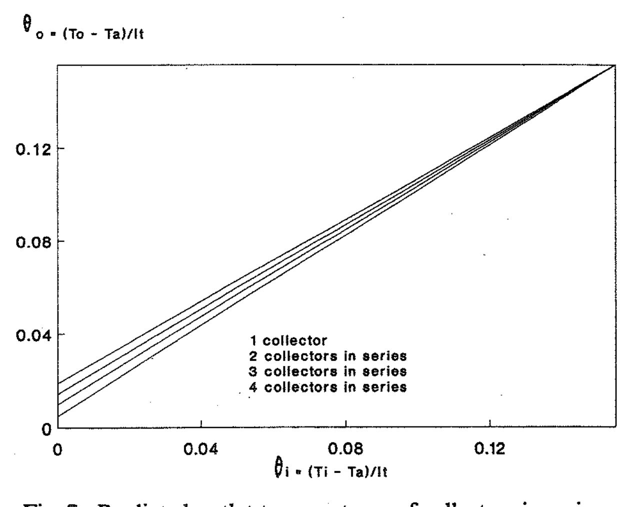


Fig. 7. Predicted outlet temperatures of collectors in series.

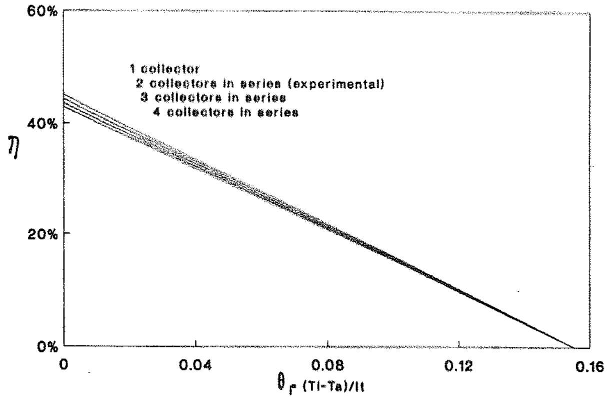


Fig. 8. Predicted efficiency of collectors in series.

efficiency. There has been a decrease in the effective transmittance-absorptance of the collector as indicated by a drop in the efficiency intercept. Since the slope of the efficiency line remains relatively unchanged, there is little change in the value of $U_L$. The lower efficiency is also partly attributed to heat losses of the distribution lines.

When the values of $(A_a/A_g)(F_R)(\tau\alpha)$ and $(\tau\alpha)/U_L$ and $G^n$ are known, the efficiency and the corresponding outlet water temperature of collectors connected in series can be predicted by using eqns (17), (25), and (27). Such information is useful in designing the water circuit of a collector array. In the case of the collectors used in the experiment, the value of $G^n$ for a single collector is found from eqn (16) to be 0.9816. The efficiency and outlet water temperature of up to 4 collectors connected in series are shown in Figs. 7 and 8, respectively. As can be expected, when more collectors are connected in series, a higher water temperature can be obtained, and with a higher water temperature, the collectors lose more heat and consequently, the efficiency of the collectors is decreased. Regardless of the number of collectors that are connected in series, their efficiency becomes zero at the same value of $(T_i - T_a)/I_t$, i.e., when it is equal to $\tau\alpha/U_L$, at which point there is no net energy collection and the outlet water temperature is equal to the inlet water temperature. At low level of radiation, if the heat exchanger section of the flat-plate heat-pipe collector is perfectly insulated, the collector efficiency can become zero but not negative, at which time there is no change in the water temperature as it flows through the collectors.

Although the solar phenomenon is essentially transient in nature, the steady-state characteristics of a collector, as obtained in the experiment, can be used for design purposes, such as the ranging of the outlet water temperatures of the collector and the estimation of the daily energy collection. The steady-state analysis will tend to give a lower estimation of the energy collection in the morning when the solar insolation is increasing, and a higher estimate in the afternoon when the solar insolation is decreasing. However, the overall daily energy estimation tends to agree with the experimental measurements.

## 5. CONCLUSIONS

For a flat-plate heat-pipe collector, or when two or more collectors are connected in series, there is a linear relationship between the efficiency and the parameter $(T_i - T_a)/I_t$. There is also a linear relationship between the parameter representing the outlet water temperature, $(T_o - T_a)/I_t$, and the parameter representing the inlet water temperature, $(T_i - T_a)/I_t$. Under steady-state conditions, the performance of a different number of collectors connected in series can be predicted from one another by using the proposed model.

## NOMENCLATURE

- $A$ collector area ($m^2$)
- $C_p$ specific heat (J/kg)
- $D$ diameter (m)
- $F'$ collector efficiency factor
- $F_R$ heat removal factor
- $i$ specific latent heat (J/kg)
- $I$ incident solar radiation ($W/m^2$)
- $k$ thermal conductivity (W/m K)
- $L$ length of pipe (m)
- $M$ mass flowrate (kg/s)
- $n$ number of heat pipes
- $N$ number of collector panels
- Nu Nusselt number
- $Q$ heat transfer rate (W)
- $R$ thermal resistance (K/W)
- Re Reynolds number
- $T$ temperature (K)
- $U$ heat transfer coefficient ($W/m^2$ K)
- $V$ velocity (m/s)
- $y$ axial length (m)
- $(\tau\alpha)_e$ effective transmittance-absorption product
- $\theta$ temperature parameter (K $m^2$/W)
- $\mu$ kinematic viscosity ($Ns/m^2$)

*Subscripts*

- $a$ aperture or ambient
- $c$ condensate
- $g$ gross area of collector
- $h$ heat-pipe fluid
- $hw$ heat-pipe fluid to water
- $i$ inlet to collector
- $l$ liquid
- $L$ heat loss
- $t$ total radiation
- $o$ outlet of collector
- $w$ water

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