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Thermal model for performance prediction of integrated collector storage systems

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The result of many years of global research on solar water heating systems has outlined the promising approach of integrated collector storage solar water heaters (ICS-SWHs) in cold climates. This research paper aims to model the field performance of a newly developed ICS-SWH for Scottish weather conditions. Field experiments were performed to investigate the performance of the newly developed ICS-SWH and the parameters affecting it. This was followed by developing a thermal macromodel able to compare the internal temperature variation under differing external weather conditions in order to evaluate the transient performance of the ICS-SWH for field conditions. Through this work, important parameters for modeling field performance of ICS-SWH were established. The innovative modeling tool developed was found to accurately predict the bulk water temperature of the ICS-SWH for any orientation and location in the world with good accuracy. © 2011 American Institute of Physics. [doi:10.1063/1.3549148]

I. INTRODUCTION

In a period of rapidly growing deployment of sustainable energy sources the exploitation of solar energy systems is imperative. Integrated collector storage solar water heater (ICS-SWH) systems are one of the simplest forms of solar water heater on the market incorporating the solar collector and thermal storage tank in a single unit. Based on published research1–9 into integrated collector storage systems, an extensive field experimental investigation of a novel finned aluminum ICS-SWH was performed in Edinburgh (56 N, 3.4 W), U.K. to investigate the influence of Scottish weather conditions on the performance and identify the different parameters influencing the ICS-SWH. The various types of ICS-SWH developed over time have impelled the need for a common criterion to measure the effectiveness of solar water heaters. Previous research9–12 showed a realistic representation of ICS-SWH performance however limitations occur as it applies (only) to diurnal system performance. This performance method is applied to flat plate collectors working in steady state mode. It can be applied to ICS-SWH; however, it is not representative of the actual system performance as it is not a steady state situation. Direct comparison of measured system performances is relatively straightforward; however, predicting the performance of an ICS-SWH at another location or with a different installation configuration is more challenging. As a result a visual basic for application (VBA) model was then developed in order to calculate the corresponding water bulk temperature in the collector, therefore predicting ICS-SWH performances.

II. EXPERIMENTAL ASSEMBLY OF ICS-SWH
A. Construction

The designed aluminum ICS-SWH was manufactured with 3 mm thick aluminum sheets and incorporated fins to improve the thermal efficiency and structural stability of the heater. The water
A cold water tank was placed in a hard wooden box insulated with layer of fiber glass wool on all sides and bottom, as shown in Fig. 1. A gap of 35 mm between the absorber plate and the glazing reduces heat losses by restricting air movement.

B. Experimental test facility

The test rig was located on the roof of Edinburgh Napier University at 15.5 m above ground-level or 113 m above sea-level and was placed facing 143° from north. A simulated roof-structure was used to insulate the sides and back of the ICS-SWH. It protected the ICS-SWH from the effects of weather by creating a microenvironment and by decreasing the heat losses from the back of the ICS-SWH. Figure 2 shows an illustration of the adopted test rig.

Kingspan insulation closed-cell polyurethane material was used to insulate the bottom of the ICS-SWH and the side walls of the simulated roof-structure. Mineral fiber wool was then placed inside the simulated roof-structure. Wood was then fixed on top of the insulation as well as a flush system on top of the collector to avoid any water ingress. Pipes were insulated in order to avoid possible freezing during cold days. Figure 3 shows the final installation.

An array of 22 K-type thermocouples of ±0.58 °C accuracy was calibrated and connected to a data acquisition system in order to monitor the water inside the aluminum collector covering middle longitudinal and lateral lines. Other thermocouples were used to record the absorber plate, the glass cover, the ambient, and the inlet and outlet water temperatures. A calibrated Kipp and Zonen BF3 pyranometer was used to record solar radiations at the location of the ICS-SWH. Wind speed and direction were recorded continuously on a 4 s basis next to the location of the ICS-SWH using a WMS302 Vaisala resulting in an average yearly wind velocity of 3.7 m/s. These average data were taken as the reference wind speed in the study. A cold water tank was placed above the...
ICS-SWH to feed the collector with fresh cold water. To investigate the performance of the aluminum ICS-SWH, extensive field measurements were conducted for an initial period of 3 months at an angle of 45°.6,7 Solar radiation and temperatures were recorded at 5 and 10 min intervals, respectively.

C. Experimental measurements

In order to identify the performance of the ICS-SWH, data management and analysis were performed for a typical month of July. Recorded data during the period of 11–26 July are represented in Fig. 4, where $T_w$ is the bulk water temperature, $T_a$ is the ambient temperature, $T_p$ is the...
plate temperature, and \( T_c \) is the cover temperature.

Each day is characterized by heating and cooling profiles which are dictated by solar radiation. The higher the incident solar radiation received by the absorber plate, the higher the water temperature achieved. The decrease in incident solar radiation results in heat losses from the collector to the surroundings leading to a subsequent decrease in water temperature. The thermal mass of the system plays an important role in the effectiveness of the heating charge and cooling discharge operations of the collector and the amount of the stored thermal energy, as shown in Fig. 4 by the gradual heating and cooling profiles. The water stored in the collector accounts for most of the thermal mass of the system. The high specific heat of 4.18 \( \text{kJ/kg} \ \text{°K} \) and moderate density of 1000 \( \text{kg/m}^3 \) of the water result in a relatively high volumetric capacity of 4180 \( \text{kJ/m}^3 \ \text{°K} \). This shows the ability of a given water volume to store internal energy while undergoing a given temperature change. Its low thermal conductivity of 0.58 \( \text{W/m} \ \text{°K} \) also contributes to this effect by slowly storing or releasing heat. The collector material also contributes to the thermal mass through its moderate volumetric capacity of 2424 \( \text{kJ/m}^3 \ \text{°K} \), assuming the density of aluminum at 2770 \( \text{kg/m}^3 \) and a specific heat of 875 \( \text{J/kg} \ \text{°K} \). This results in a thermal mass of 224.7 \( \text{kJ/°K} \) for the collector. An overall efficiency of 57\% for this ICS-SWH was obtained from this analysis.

1. Sky radiation effect on cover temperature

Further analysis of the cover and ambient temperatures in Fig. 5 showed significant outcomes. It can be seen that recorded cover temperatures were lower than ambient temperatures during daytime in the 11 and 15 of July. A possible explanation for this might be that the effective sky temperature can be about 10–20 °C below ambient temperature at ground-level with clear sky conditions and close to or just below the ambient temperature during cloudy conditions. In order to assess this possible explanation the clear sky index (kc) for the location of the ICS-SWH for those dates was taken from the British Atmospheric Data Centre (BADC)\(^4\) and plotted against both temperatures in Fig. 5.

Both days were recorded as clear sky days. The sky acts as a black body during night-time and during an early morning clear sky, resulting in a cover temperature lower than the ambient temperature.

FIG. 5. Sky radiation effect on cover temperature with kc as the clear sky index.
2. Stratification and solar radiation

In order to identify the influence of incident solar radiation on stratification, the rise in temperature registered by thermocouples placed at different locations along the length of the collector is illustrated in Fig. 6. The importance of the shading effect on the collector was observed for the hours 11, 12 and 13 where a decrease in stratification rate was recorded. This supports previous research findings introduced during laboratory study.13

In order to support these findings experimental temperature profiles were scaled to the ratio $T_h/T_{\text{bottom}}$, where $T_h$ is the local and $T_{\text{bottom}}$ is the minimum recorded temperature in the water tank. Dimensionless stratification profiles for different hours are represented in Fig. 7 for the same typical day.

Temperature stratification decreases gradually over night until 6 a.m., at which time incident solar radiation starts being collected by the absorber plate. While the overall collector temperature increases from 7 a.m. to 2 p.m., stratification increases from 7 a.m. to 9 a.m. then decreasing later with time. This is explained by the buoyancy effect when water at a given temperature settles down at an appropriate height in the ICS-SWH in accordance with the prevalent density of the fluid.15 From 7 a.m. high temperature stratification occurs in the ICS-SWH. At 9 a.m. the tem-
perature of the upper layers is established resulting in the lower layer achieving a narrower temperature range, therefore decreasing the stratification in the collector. A decrease in destratification rate was observed at 11–13 h suggesting that the collector was shaded. This is supported by a sudden decrease in incident solar radiation on the collector for this period of time.

III. COMPUTATIONAL MODELING

A. The macromodel

Based on past studies\textsuperscript{15,16}, a macromodel for fluid flow and heat transfer in the ICS-SWH was developed using VBA to model the field performance of ICS-SWH. The model is a thermal energy simulation model, in which analyses combined heat flux environmental networks within the ICS-SWH. It was developed to compare the temperature variation in different ICS-SWH materials, internal water temperature, and external weather conditions for a given aspect ratio. The model generates the water bulk temperature in the collector with a given hourly incident solar radiation (W/m\textsuperscript{2}), ambient temperature, and inlet water temperature, and could cover and average wind velocity for the location.

B. The macronetwork and fundamental heat transfer analysis

1. Thermal network

The model was developed on a steady state approach as the water temperature does not change significantly in very short time intervals. The global thermal network of the ICS-SWH system will remain the same and is shown in Fig. 8 below including the water body “W,” the absorber plate “P,” the cover “C,” the ambient air “A,” and the “sky” nodes.

2. Fundamental heat transfer analysis

Transient performance of the system was predicted for a 360 s time interval by solving the mathematical model consisting of energy balance equations. An iterative process was developed to solve the thermal network. The thermal analysis was undertaken for a 45° collector inclination. A number of assumptions were made in developing the macromodel, for instance, from lack of data or in order to reduce computation. Effects of assumptions were carefully considered in order to ensure that the analysis was appropriate. In order to compare the results with field conditions the following assumptions were made: (a) the absorber plate was assumed to have a constant rate of heat flux for the small time increments taken by the program, (b) an average wind velocity of 3.7 m/s was used for the field experiment based on data recorded on the roof where the ICS-SWH was previously tested, (c) radiative heat losses from the glass cover accounted the glass cover and sky emissivities, (d) clear, overcast, and part-overcast conditions were differentiated to generate the performance of the collector and were set as clear sky conditions: 0 ≤ CC ≤ 2, overcast sky conditions: 6 ≤ CC ≤ 8, and part-overcast sky conditions: 3 ≤ CC ≤ 5, (e) if overcast condition data were not available then the cloud cover (CC) was assumed as part-overcast (CC=4), (f) the dry-bulb temperature for part-overcast conditions was assumed as the average between the overcast and clear sky conditions, and (g) stratification inside the collector was neglected for field experiments; it was thus assumed that water temperature inside the collector was uniform.

The energy balance on the absorber plate at a given instant, assuming steady state conditions, is given by

\[
q_{\text{useful}} = G(\alpha) - q_{\text{p-c}} - q_{\text{w-a}} = \sum (C) \times (T'_w - T_w),
\]

\[
q_{\text{p-c}} + q_{\text{p-w}} = G(\alpha) \quad \text{at } t = 0,
\]

\[
q_{\text{p-c}} = q_{\text{c-a}}.
\]

Heat loss from the plate to the cover. These can be expressed by
\[
q_{p-c} = h_{pc}(T_p - T_c) + \frac{\alpha(T_p^4 - T_C^4)}{e_B} = U_{pc}(T_p - T_c),
\]

where \(e_B\) is the bulk emissivity defined by equation \(e_B = \frac{1}{e_G} + \frac{1}{e_P} - 1\).

Based on previous research\(^\text{17}\) the convective heat transfer coefficient for inclined cavities from the absorber plate to the glass cover, \(h_{pc}\), can be expressed as

\[
h_{pc} = \frac{1 + 1.44 \left[ 1 - \frac{1708}{R_{a_L} \cos \theta} \right]^* \left[ 1 - \frac{1708(\sin 1.8\theta)}{R_{a_L} \cos \theta} \right]^{1.6} + \left[ \left( \frac{R_{a_L} \cos \theta}{5830} \right)^{1/3} - 1 \right]^* \times k_{fa}}{L}.
\]

The notation * implies that, if the quantity in brackets is negative, it must be set equal to zero.

Heat loss from the cover to the surroundings. Losses to surroundings to compute field conditions are expressed as

\[
q_{c-s} = h_{wind}(T_c - T_a) + \frac{\alpha(T_c^4 - T_{sky}^4)}{e_{sur}} = U_{ca}(T_c - T_a) + U_{cs}(T_c - T_{sky}).
\]

The correlation for the external convection occurring on the collector surface is expressed as\(^\text{18}\)
The sky is acting as a black body during night and during clear sky conditions in early morning. The sky temperature $T_{sky}$ was then determined by using the expression developed by Berdahl and Martin in Eq. (6) below where the sky temperature is related to the dew-point temperature $T_{dp}$, ambient temperature $T_a$, and time relative to midnight $t_{mid}$.

$$T_{sky} = T_a^{0.711} + 0.0056 T_{dp} + 0.000 073 T_{dp}^2 + 0.013 \cos \left( \frac{15 t_{mid}}{4} \right).$$

The sky emissivity was calculated by integrating Eq. (7) of clear sky emissivity $\varepsilon_{CS}$, developed by Cucumo et al., into Santamouris and Asimakopolous’s Eq. (8) for sky emissivity in the transparency window, $\varepsilon_{sky}$.

$$\varepsilon_{CS} = 1 + \frac{107 952 \times (1 - \varepsilon)}{T_a^2 - 680.8 \times T_a + 73 594.4},$$

with

$$\varepsilon = 0.747 + 0.6001 \times \frac{T_{dp}}{100} - 0.6668 \times \left( \frac{T_{dp}}{100} \right)^2,$$

where $\varepsilon_{CS}$ is the atmospheric emissivity in the transparency window for clear sky condition and $\varepsilon$ is the global emissivity.

The sky emissivity in the transparency window, $\varepsilon_{sky}$, was then calculated by using the final expression expressed below,  

$$\varepsilon_{sky} = \varepsilon_{CS} \times (1 + 0.0224 \times n - 0.0035 \times n^2 + 0.000 28 \times n^3),$$

where $n$ is the total opaque cloud amount; 0 for clear sky and 1 for overcast sky.

A bulk emissivity $\varepsilon_{sur}$, taking into account the sky and glass cover emissivity of equation $1/\varepsilon_+ 1/\varepsilon_{sky} = 1$, was then introduced to calculate the radiative heat losses from the glass cover to the surrounding for field experiments.

Dew-point and ambient temperatures relationship were developed for Edinburgh, based on one year data from the BADC. These data are plotted in Figs. 9 and 10 from which the following
relationships were developed for the different types of sky:

For clear days: \[ T_{dpC} = 0.71 \times T_a - 0.79, \]  
(11)

For overcast days: \[ T_{dpO} = 0.82 \times T_a - 0.81. \]  
(12)

It was assumed that part-overcast dew-point temperature was the average between clear and overcast conditions resulting in a dew-point expression below,

\[ T_{dpPO} = \frac{T_{dpC} + T_{dpO}}{2}. \]  
(13)

Heat loss from the absorber plate to the water. The heat transfer to the water is mainly a convective process, thus conduction was neglected. The heat loss from the absorber plate to the water is expressed below,

\[ q_{p-w} = h_w A_{pw}(T_w - T_p) = U_{pw}(T_w - T_p). \]  
(14)

A Nusselt expression for trapezoidal-shaped ICS-SWH based on previous studies\textsuperscript{22–24} was used to evaluate \( h_w \) and is expressed as

\[ h_w = \left[ \frac{0.56 (Ra \cos \theta)^{1/4} D^{1/9} H^{1/6}}{U L} \right] k_{fw}, \]  
(15)

where \( H = 1m \) and \( D = U = 0.05m \). A visual description of the geometric parameters of the ICS-SWH is shown in Fig. 11 below.

Heat loss from the water to the ambient.

\[ q_{w-a} = U_{loss}(T_w - T_a), \]  
(16)

with

\[ U_{loss} = \frac{1}{R_{wind} + R_{insulation}}. \]  
(17)
Heat loss from the fins to the water. The inclusion of extended heat transfer surfaces (fins) into the design provided an increased heat transfer to the water. Previous work\textsuperscript{13,25} showed that using elemental strip method to determine the heat transferred by the fin to the water assuming an adiabatic tip condition gave accurate results. The fin temperature distribution along the length $x$ was determined by the following formula:

$$T_{f,x} = \theta_0 \times \frac{\cosh m(d-x)}{\cosh md} + T_w.$$  

The fin heat transfer rate $q_f$ per strip can be expressed as

$$q_f = h_f A_f (T_f - T_w).$$  

The heat transfer to the water is mainly a convective process. Based on previous studies\textsuperscript{26,27} the following regression was used:

$$\text{Nu} = 0.68 + \frac{0.67 \text{Ra}^{1/4}}{[1 + (0.437/\text{Pr})^{9/16}]^{4/9}},$$  

where $L = d/\cos(\theta)$ and a uniform heat flux is assumed.

Based on previous studies\textsuperscript{16} the total heat transferred through the fins to the water, $q_{\text{fins}}$, was calculated by adding up the fin heat transfer rate $q_f$ per strip until reaching the limit effective length defined when 90\% of the actual $\theta_p = T_p - T_w$ was achieved and is expressed as

$$q_{\text{fins}} = \sum_{0}^{90\%} h_x \theta_x A_x.$$  

Calculation of the water bulk temperature. Using the equations developed above, an iterative computer program was developed using VBA with initial conditions of $T_p$ and $T_c$ set as

$$T_p = T_w + 0.5.$$
\[ T_c = T_p - \left( \frac{T_p - T_a}{3} \right), \]

Using Eqs. (2) and (3), the expressions for \( T_p \) and \( T_c \) for \( t=0 \) were developed below in Eqs. (15) and (16),

\[ T_c = \frac{U_w T_p + U_{ca} T_a + U_{cs} T_{sky}}{U_{ca} + U_w + U_{cs}}, \quad (22) \]

\[ T_p = \frac{G(\tau \alpha) + U_{pc} T_c + U_w T_w}{U_{pc} + U_w}. \quad (23) \]

Iterations were made until a balance was obtained between heat losses. Using Eq. (1) a transformed Eq. (17) can be developed to calculate the water temperature rise,

\[ T'_w = T_w + \frac{q_{\text{useful}}}{\sum C} = T_w + \frac{q_{p-w} + q_{\text{fins}} - q_{w-a}}{\sum C}. \quad (24) \]

An interval of 1 h was taken to display the results. After each interval the values of the cover, absorber plate, and water temperature were calculated and used as the input for the next time step. The cycle was repeated for 24 h time increments.

**IV. COMPUTATIONS AND DATA COMPARISON**

**A. Results**

A first analysis carried out for a period of 5 days explores the validity of the model to predict the bulk water temperature inside the ICS-SWH by comparing the computed results with experimental data. Results are plotted in Fig. 12.

Similar to experimental field results, the computed results show that each day is characterized by heating and cooling profiles which are dictated by solar radiation. The higher the incident solar radiation received by the absorber plate, the higher the water temperature computed. It was observed that gradual heating and cooling profiles due to the thermal mass of the system were experienced due to the consideration of thermal mass in the macromodel. However, a lower
cooling rate of the modeled values was observed each day. This could be explained by the propagation of errors and uncertainty by the use of regression analysis and assumed weather data, resulting in a decrease in accuracy of the model.

It was also observed in Fig. 12 that experimental water temperature was followed very closely by predicted water temperature although it showed a slight overestimation by the proposed model. A more detailed picture of a typical day of July is illustrated in Fig. 13.

FIG. 13. 1 day computed and experimental data in July 2007.

In order to examine the validity of the simulation, computed data were plotted as a function of experimental data in Fig. 14.

It was demonstrated that experimental and computer simulation results were in good agreement as they followed a 1:1 relationship. Statistical indicators were then used to evaluate the performance of the model for those 5 days and are resumed in Table I.

The slope of the best fit line suggests a slight overestimation of the computed variable. The high value of $R^2$ indicates a low unexplained variation showing a satisfactory accuracy for the regression model. The Mean Bias Error (MBE) value of $-1.27$ shows that the model has a tendency to overpredict its computed values. The Root Mean Square Error (RMSE) value of 2.19 shows an acceptable actual deviation. The low positive skewed distribution trends to the high end of the scale and indicates a robust model. The positive low kurtosis indicates a peaked distribution of the errors suggesting that there are low outliers in the estimation.

A study of the effect of wind velocity on the bulk water temperature was carried out and is illustrated in Fig. 15, where $V_0$, $V_2$, $V_4$, and $V_6$ represent the wind velocity values of 0, 2, 4, and 6 m/s, respectively. The higher the wind velocity, the higher the heat losses resulting in lower bulk water temperature showing the influence of wind speed on overall performance. Thus, having accurate wind speeds would result in more accurate bulk water temperatures. This shows that the more detailed and complete the weather information inputs, the fewer error sources, and thus confidence in the macromodel results might be achieved.

A second simulation was carried out over a period of 50 days in order to determine the validity of the model to predict the bulk water temperature for an extended period. Computed data were plotted as a function of experimental data to examine the validity of the simulation. It was

<table>
<thead>
<tr>
<th>Statistical indicator</th>
<th>Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Slope</td>
<td>1.03</td>
</tr>
<tr>
<td>$R^2$</td>
<td>0.97</td>
</tr>
<tr>
<td>MBE (°C)</td>
<td>$-1.27$</td>
</tr>
<tr>
<td>RMSE (°C)</td>
<td>2.19</td>
</tr>
<tr>
<td>Skewness (°C)</td>
<td>0.86</td>
</tr>
<tr>
<td>Kurtosis</td>
<td>0.42</td>
</tr>
</tbody>
</table>

FIG. 15. Simulated effect of wind velocity on water temperature for a 5 day period.
demonstrated that experimental and computer simulation results were in good agreement, as shown in Fig. 16.

The same statistical indicators were then used to evaluate the performance of the model to predict the bulk water temperature in the ICS-SWH for the extended 50 day test and are given in Table II.

The slope of the best fit line suggests an overestimation of the computed variable. The high value of $R^2$ indicates a low unexplained variation showing a good adequacy of the regression model. The MBE value of $-1.49$ shows that the model has a tendency to overpredict its computed values by nearly 1.5 °C. The RMSE value of 2.22 shows an acceptable actual deviation. The low positive skewed distribution tails off to the high end of the scale while the positive low kurtosis indicates a peaked distribution of the errors suggesting that there are low outliers in the estimation.

Results validated the field macromodel as a robust model to predict the bulk water temperature in the ICS-SWH under different weather conditions.

**FIG. 16.** Computed vs experimental for a 50 day period.

<table>
<thead>
<tr>
<th>Statistical indicator</th>
<th>Values</th>
</tr>
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<tbody>
<tr>
<td>Slope</td>
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<tr>
<td>$R^2$</td>
<td>0.97</td>
</tr>
<tr>
<td>MBE (°C)</td>
<td>-1.49</td>
</tr>
<tr>
<td>RMSE (°C)</td>
<td>2.22</td>
</tr>
<tr>
<td>Skewness (°C)</td>
<td>0.21</td>
</tr>
<tr>
<td>Kurtosis</td>
<td>0.16</td>
</tr>
</tbody>
</table>
V. CONCLUSIONS

A field experiment was carried out to determine performance dependent parameters of modeling real weather conditions. Parameters such as thermal mass of the system, incident solar radiation, cloud cover, and sky temperature were found to influence significantly the ICS-SWH performance and were of high interest in order to model the performance of such ICS-SWH for field conditions. Current procedures for measuring the effectiveness of solar water heaters emphasize a need to predict the water temperatures and performances achieved from ICS-SWH.

Further work on the initial laboratory thermal model resulted in the development of a thermal model suitable to predict the bulk water temperature in real weather conditions. This model was able to compare the temperature variation and predict the bulk water temperature in diverse ICS-SWH configurations with different geometry, number of fins, and external weather conditions for a given aspect ratio. Simulations for a 5 and 50 day period were undertaken and both showed that computational results were found to be in close agreement with the experimental field measurements. This statement was validated by statistical methods suggesting that the field macromodel is a robust tool to compute the bulk water temperature in the ICS-SWH for any weather conditions. Although this model only gives mean values of water temperature, while in reality this varies along the longitudinal height of the collector, it gives a good estimation of the ICS-SWH performance. Further improvements of the thermal model such as the modeling of water draw-off and stratification could be integrated.

Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$T_w$</td>
<td>water temperature at $t=i$ K</td>
</tr>
<tr>
<td>$T'_w$</td>
<td>water temperature at $t=i+1$ K</td>
</tr>
<tr>
<td>$T_{w,ini}$</td>
<td>initial water temperature at $t=0$ K</td>
</tr>
<tr>
<td>$T_a$</td>
<td>ambient temperature K</td>
</tr>
<tr>
<td>$T_p$</td>
<td>absorber plate temperature K</td>
</tr>
<tr>
<td>$T_c$</td>
<td>glass cover temperature K</td>
</tr>
<tr>
<td>$T_{sky}$</td>
<td>sky temperature K</td>
</tr>
<tr>
<td>$T_{dp}$</td>
<td>dew-point temperature K</td>
</tr>
<tr>
<td>$T_{f,x}$</td>
<td>fin temperature at fin position $x$ K</td>
</tr>
<tr>
<td>$\Delta T_m$</td>
<td>rise in mean water temperature $\Delta T_m = T_w - T_{w,ini}$ K</td>
</tr>
<tr>
<td>$\theta_p$</td>
<td>excess temperature at $x=0$=base of the fin ($\theta_p = T_p - T_w$) K</td>
</tr>
<tr>
<td>$dt_W$</td>
<td>temperature difference of the water at a time interval $dt$ K</td>
</tr>
<tr>
<td>$dt$</td>
<td>the time interval in second s</td>
</tr>
<tr>
<td>$G(\tau a)_e$</td>
<td>rate of incident solar radiation transmitted W</td>
</tr>
<tr>
<td>$U_{pc}$</td>
<td>absorber plate-glass cover overall U-value W/K</td>
</tr>
<tr>
<td>$U_{loss}$</td>
<td>water-ambient overall U-value W/K</td>
</tr>
<tr>
<td>$U_{ca}$</td>
<td>glass cover-ambient overall U-value W/K</td>
</tr>
<tr>
<td>$U_w$</td>
<td>absorber plate-water overall U-value ($=h_w$) W/K</td>
</tr>
<tr>
<td>$R_{ins}$</td>
<td>resistance of the insulation material K/W</td>
</tr>
<tr>
<td>$R_{wind}$</td>
<td>resistance occurring at the box surface in contact with the wind K/W</td>
</tr>
<tr>
<td>$q_{useful}$</td>
<td>useful energy transferred to the water W</td>
</tr>
<tr>
<td>$q_{w-a}$</td>
<td>water-ambient energy lost W</td>
</tr>
<tr>
<td>$q_{p-w}$</td>
<td>absorber plate-water energy lost W</td>
</tr>
<tr>
<td>$q_{c-s}$</td>
<td>glass cover-surrounding heat loss W</td>
</tr>
<tr>
<td>$q_{p-c}$</td>
<td>absorber plate-glass cover energy lost W</td>
</tr>
<tr>
<td>$q_{fins}$</td>
<td>total energy transferred from the fins to the water W</td>
</tr>
<tr>
<td>$q_f$</td>
<td>fin heat transfer rate per stripes W</td>
</tr>
<tr>
<td>$h_{pc}$</td>
<td>absorber plate-cover convection coefficient W/m$^2$ K</td>
</tr>
<tr>
<td>$h_{wind}$</td>
<td>glass cover-ambient external convection coefficient W/m$^2$ K</td>
</tr>
<tr>
<td>$h_w$</td>
<td>absorber plate-water convection coefficient W/m$^2$ K</td>
</tr>
<tr>
<td>$h_f$</td>
<td>fin heat transfer coefficient per stripes W/m$^2$ K</td>
</tr>
<tr>
<td>$k_{fa}$</td>
<td>thermal conductivity of air W/m K</td>
</tr>
</tbody>
</table>
\( k_W \) thermal conductivity of water W/m K
\( C_w \) thermal capacitance of the water J/K
\( C_c \) specific heat of the collector material J/K
\( C_i \) thermal capacitance of the insulation J/K
\( C_{wd} \) thermal capacitance of the wood box J/K
\( C_g \) thermal capacitance of the glazing J/K
\( C_m \) overall thermal capacitance of the material J/K
\( C \) overall thermal capacitance of the system J/K
\( V_W \) wind speed m/s
\( \theta \) tilt angle of the absorber plate rad
\( A_C \) cross sectional area m²
\( A_f \) fins area \((=N \times x \times w)\) m²
\( t \) thickness of the fins m
\( w \) width of the fins m
\( P \) perimeter m
\( x \) fins position m
\( d \) depth of the fins m
\( \sigma \) Stefan–Boltzmann constant J/s m² K⁴
\( t_{mid} \) time relative to midnight h
\( n \) total opaque cloud amount \((0=\text{clear sky}, 1=\text{overcast sky})\)
\( \text{IP} \) improvement factor
\( N \) number of fins
\( \varepsilon \) global emissivity
\( \varepsilon_C \) emissivity of the glass cover
\( \varepsilon_P \) emissivity of the absorber plate
\( \varepsilon_B \) bulk emissivity temperature of the absorber plate
\( \varepsilon_{CS} \) clear sky emissivity in the atmospheric transparency window
\( \varepsilon_{sky} \) sky emissivity in the transparency window
\( \varepsilon_{g_{10}} \) bulk emissivity of the glass cover
\( m^2 = hP/kA_C \)

13. BADC—British Atmospheric Data Centre (2007).