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Design and Optimization of an Air Heating Solar Collector with Integrated Phase Change Material Energy Storage for Use in Humidification-Dehumidification Desalination

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Abstract

Compared to solar water heaters, high-temperature solar air heaters have received relatively little investigation and have resulted in few commercial products. However, in the context of a Humidification-Dehumidification (HD) Desalination cycle, air heating offers significant performance gains for the cycle. Heating at a constant temperature and constant heat output is also important for HD cycle performance. The use of built in phase change material (PCM) storage is found to produce consistent air outlet temperatures throughout the day or night. In this study, the PCM has been implemented directly below the absorber plate. Using a two dimensional transient finite element model, it is found that a PCM layer of 8 cm below the absorber plate is sufficient to produce a consistent output temperature close to the PCM melting temperature with a time-averaged collector thermal efficiency of 35\%, which is significantly less than a collector without built-in energy storage. An experimental energy storage collector with an 8 cm thick PCM layer was built and tested in both a variety of weather and operating conditions. Experimental results show strong agreement with model in all cases.

Key words: Solar Air Heater, Humidification, Dehumidification, Desalination, Energy Storage, Phase Change Material

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Nomenclature

Roman Symbols

\( A_p \)  Collector area \([m^2]\)
\( c_p \)  Specific heat capacity at constant pressure \([J/kg K]\)
\( d \)  Depth \([m]\)
\( F \)  Radiation transmittance factor
\( F_{cg} \)  View factor between collector and ground
\( F_{cs} \)  View factor between collector and sky
\( h_1 \)  Convective heat transfer coefficient between absorber and airstream \([W/m^2K]\)
\( h_2 \)  Convective heat transfer coefficient between glazing and airstream \([W/m^2K]\)
\( h_r \)  Radiation heat transfer coefficient between the absorber and glazing \([W/m^2K]\)
\( h_{sf} \)  Latent heat of solidification \([J/kg]\)
\( I \)  Solar irradiation flux \([W/m^2]\)
\( k \)  Thermal conductivity \([W/m K]\)
\( \dot{m} \)  Mass flow rate of air through the collector \([kg/sec]\)
\( NG \)  Normalized Gain \([K m^2/W]\)
\( Re \)  Reynolds number based on hydraulic diameter
\( S \)  Heat flux into absorber plate \([W/m^2]\)
\( s \)  Relative density of a metal mesh
\( T \)  Temperature \([K]\)
\( t \)  Simulation time index \([sec]\)
\( T_{melt} \)  PCM melting temperature \([K]\)
\( U_b \)  Bottom loss coefficient \([W/m^2K]\)
\( U_t \)  Top loss coefficient \([W/m^2K]\)
\( V \)  Volume \([m^3]\)

Greek Symbols

\((\tau \alpha)\)  Transmittance-absorptance product
\( \Delta T \)  Temperature difference between the absorber and inner glazing \([K]\)
\( \delta T \)  Temperature range over which the PCM melts \([K]\)
\( \eta \)  Collector efficiency
\( \rho \)  Density \([kg/m^3]\)
\( \rho_g \)  Average ground reflectance
\( \sigma \)  Stefan-Boltzman constant \([W/m^2 K^4]\)
\( \tau \)  Time constant \([sec]\)

Subscripts

\( a \)  Absorber Plate
\( Al \)  Aluminum
\( beam \)  Beam irradiation component
\( c1 \)  Inner Glazing
\( c2 \)  Outer Glazing
<table>
<thead>
<tr>
<th>Symbol</th>
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<tr>
<td>$d$</td>
<td>Diffuse irradiation component</td>
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<tr>
<td>$exp$</td>
<td>Experimental</td>
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<td>$g$</td>
<td>Ground reflected irradiation component</td>
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<td>$in$</td>
<td>Air inlet</td>
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<td>$l$</td>
<td>Phase change material liquid phase</td>
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<td>$out$</td>
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<td>$PCM$</td>
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<td>$sim$</td>
<td>Simulated</td>
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<td>$T$</td>
<td>Radiation on a tilted surface</td>
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</tbody>
</table>
1. Introduction

Solar water heaters have been thoroughly investigated and developed commercially [5, 13, 17]. While there are commercial air heaters [28] their use is primarily for low-temperature (40 °C) space heating applications and do not use any form of energy storage. In the context of Humidification-Dehumidification (HD) desalination, heating the air high temperature (50-80 °C), as opposed to the water stream leads to significant performance gains [22]. Heating at a constant temperature also provides more stability to the HD system and increases performance over time [21].

Nayaran et al. [22] laid out a comparison of current solar air heating technologies. Summers et al. [29] investigated the performance of air heaters at steady state and showed improved performance over current designs; however, this analysis did not take into account the transient nature of solar irradiation throughout the day from sunrise to sunset. Rapidly dropping temperatures at the end of the day can severely reduce performance of a HD desalination system, ultimately reducing the amount of water obtained.

Energy storage is an important aspect of collecting intermittent energy like that from the sun, especially for applications like HD desalination in which a stable warm temperature is needed for optimum performance. The goal of a properly optimized storage system is to deliver heat at an approximately constant temperature throughout a 24 hour day. This avoids startup effects in the morning as well as energy loss when the cycle top temperature drops at night.

2. Comparison of Performance

Summers et. al. [29] compared the performance of solar air heaters using the collector efficiency. This was an instantaneous power efficiency that was applicable to a collector with no heat capacity effects running in steady state with constant solar irradiation. Since this study concerns time varying radiation, which is zero at night, a time averaged value of efficiency will be used as defined by Equation 1:

$$\eta = \frac{\int_{t_1}^{t_2} \dot{m}c_p[T_{out}(t) - T_{in}] dt}{\int_{t_1}^{t_2} IA_p dt} \quad (1)$$

This definition of performance is the time average of the efficiency is used in the ASHRAE 93-2003 Standard for solar collector testing [25].

Additionally, to evaluate normalized gain, which measures a normalized temperature rise, or the relative quality of heat provided by the collector, over a period of time during which the solar irradiation may be zero (i.e., night), a time averaged normalized gain is defined by Equation 2:

$$NG = \frac{\int_{t_1}^{t_2} [T_{out}(t) - T_{in}] dt}{\int_{t_1}^{t_2} I dt} \quad (2)$$
3. Built-in Energy Storage

In this design, as shown in Figure 1, a phase change material (PCM) is placed below the absorber plate in direct contact with the absorber. This allows heat to be transferred directly from the absorber plate to the storage medium and then directly from the storage medium to the air when the sun is not shining. This eliminates the need for a separate apparatus and control systems for external storage as well as the associated heat losses. Additionally, the low heat capacity working fluid is not used to transfer heat to the storage. Using a built-in storage system also eliminates the need for a secondary working fluid loop and the inherent increase in complexity and cost.

The use of a PCM as the storage medium stabilizes the output temperature of the collector by storing heat near the melting temperature. The latent heat of a PCM, such as paraffin wax, can be 100 times greater than its sensible heat capacity, which is of the same order of magnitude of most solids. Therefore less material is required to store the same amount of heat. The smaller volume also reduces the need for high contact area or a high thermal conductivity in the storage material.

3.1. Existing Built-in Storage Collectors

Built-in energy storage has commonly been used for solar energy storage. Water heaters have used built-in storage for many years. Early water heaters [27] took advantage of the sensible heat storage in the water itself and made insulated tanks that held hot water heated by the sun during the day and used at night. Garg [12] evaluated the performance of one of these heaters by varying the water tank depth. He found that water at 60°C was available after sunset, even in winter.

In other instances a phase change material was used as part of a water tank to augment the storage capacity of water. Narayan et al. [23] used paraffin wax encapsulated at the bottom of a heated flat plate over the water tank. Kumar [18] used a tank of paraffin wax which absorbed solar radiation, and heat was removed by water flowing through three finned heat exchangers. In some cases, water flowed through pipes between an absorber surface with integrated phase change material. Rabin et al. [24] used a configuration where pipes carrying water were placed in an oil that floated on top of a more dense salt hydrate phase change material. The oil served as an interface between the PCM and water, spreading the heat over the surface. A black absorber plate was above the oil to collect solar energy.

The use of built-in PCM storage in a solar air heater is much less common. Enibe [7] used a natural convection driven heater where air flowed below the absorber and through banks of paraffin wax. The absorber had high thermal conductivity. A minimum temperature rise of 7°C with 200 W/m² of incident radiation was obtained. Jain [16] used an air heater for crop drying that contained a sensible heat storage medium placed below the absorber plate. The collector contained two glazings and air passed between the glazings and then over the absorber plate. It then passed under the bottom side of the absorber storage combination, and into the crop dryer. He achieved a minimum temperature rise of 5°C during the night with 900 W/m² of incident radiation. The efficiency of the dryer never exceeded 35%.
The air heater closest to what is desired here is due to Fath (1995) [11]. He used a series of close packed tubes containing wax, hot salts, or sand (for comparison with sensible heat storage). The tubes were arranged close to each other in parallel to make a flat-plate-style configuration with air flowing above and below them simultaneously. The collector could maintain a minimum of 5°C of temperature difference and ran at design conditions 8 hours longer than a collector without storage. The collector used a wax PCM that melted at 51°C.

3.2. Enhancing PCM Conductivity

Paraffin wax is commonly used as a phase change material because it melts at specific temperatures, is unreactive, and is inexpensive. The properties of wax used [15] are given in Table 3. As it can be seen, it has low thermal conductivity which can result in substantial thermal resistance between the heat transfer surface and melt front. It is important that heat from the absorber plate can melt the entire depth of the PCM. To get around this problem and better utilize the PCM, a metal matrix with high thermal conductivity can be embedded in the PCM [19]. The most cost effective matrix of this type is an aluminum wool or mesh grid. This method has been utilized in heat exchangers with phase change material buffering, but has not been used in a solar collector.

The conductivity of the matrix and PCM combination is calculated by using the relative density, $s$, of the metal wool. For example, commercial metal wool [2] has a density of 128 kg/m$^3$, giving it a relative density, $s$, of 0.05, when compared to solid aluminum. The thermal conductivity of the PCM-matrix system is given by Equation 3:

$$k_{\text{eff}} = (s)k_{\text{Al}}/3 + (1-s)k_{\text{PCM}}$$

The factor of 1/3 takes into account the random internal structure of the wool: only an average of 1/3 of the threads are carrying heat through any given orthogonal direction. A similar equation is used for metal foams [8], which are stiffer but have a similarly random internal structure. Therefore the same equation applies. For wire mesh, where the threads are only aligned in two directions, 1/3 is replaced by 1/2 using similar reasoning.

In this study aluminum wool is assumed to be embedded in the PCM.

3.2.1. Effect of Aluminum on Other Thermal Properties

Addition of an aluminum mesh or wool has an effect on the other thermal properties of the system. It changes the sensible heat capacitance or $(\rho c_p)$ product. This change varies with the relative density of aluminum. Equation 4 shows how the sensible heat capacitance of the added aluminum compares with the wax.

$$\frac{(\rho c_p)_{\text{Al}}s}{(\rho c_p)_{\text{PCM}}(1-s)} \sim \frac{s}{1-s}$$

Since $s$ remains small, on the order of 0.1, the addition of the aluminum matrix does not significantly change the sensible heat storage.
4. Mathematical Model

A simple lumped parameter model was used by Summers et al. [29] to describe the temperatures and heat fluxes in a solar collector operating at constant irradiation with no heat capacity and transient effects. Due to the inclusion of transient effects in this study, there are spatial and temporal temperature variations in the absorber plate, and, as a result, an irregularly shaped PCM melt front that varies along the length of the collector, as well as in the depth of the PCM. Therefore, more detailed modeling is required to determine the temperature distribution and melt front shape in the storage material. Finite element modeling which enables a numerical solution of the heat equation in two dimensions and time is an appropriate choice. Due to its ability to handle temperature dependent material properties, the ADINA finite element modeling system was chosen [1].

4.1. Phase Change Model

Because ADINA does not have a method for handling phase change calculations, the phase change had to be encapsulated as part of the heat capacity of the material. This has been done in previous finite element simulations using two different methods. One method by Leoni and Amon [19], raises the heat capacity at the melting temperature to represent the energy absorption associated with melting the substance over a temperature range from $T_{\text{melt}}$ to $T_{\text{melt}} + \delta T$. It is given in the following equation:

\begin{align}
  c_p &= c_{ps} \quad \text{for } T < T_{\text{melt}} \\
  c_p &= c_{ps} + \frac{h_{sf}}{\delta T} \quad \text{for } T = T_{\text{melt}} + \delta T \\
  c_p &= c_{ps} \quad \text{for } T > T_{\text{melt}} + \delta T
\end{align}

The other method is to neglect the ordinary sensible heat and to treat the latent heat as an increasing heat capacity that changes as the substance gets warmer. This is proposed by Gong and Mujumdar [14] and is given as Equation 6:

$$c_p = \frac{h_{sf}}{(T_{\text{melt}} - T)} \quad (6)$$

Because Equation 6 does not allow for temperatures beyond the melting point and neglects sensible heat storage, the first method was chosen for the numerical model.

4.2. Solar Radiation Model

Solar irradiation was calculated for Dammam, Saudi Arabia for mid-July using the isotropic sky model [4] and constant environmental data given in Table 1. Data for the daily variation of solar radiation as it varies throughout the day in Dammam were obtained from the European Union Joint Research Center PVGIS Database [9]. Data were obtained for both a horizontal surface and a tilted surface at the optimal angle of 26.4°. Values for ground reflectivity were obtained [4, 6, 3] and a value of 0.3 was used for a typical desert
environment. With this known it is possible to obtain an expression for the heat radiation input $S$:

$$S = I_{T, beam}(\tau_\alpha)_beam + I_{T, d}(\tau_\alpha)_d + I_{\rho g} F_{cg}(\tau_\alpha)_g$$

(7)

The $(\tau_\alpha)$ product for beam irradiation was calculated based on the incidence angle of the sun’s rays (based on solar declination and collector tilt angle), the index of refraction of the glazings, the glazing thickness, the absorber absorptivity, and the extinction coefficient of the glazings, given in Table 2. For the other $(\tau_\alpha)$ products, the same calculation was made except that the incidence angle was replaced with an effective tilt angle of diffuse and ground reflected radiation [4]. Each of the $(\tau_\alpha)$ products are listed in Table 5.

4.3. Heat Transfer Model

The solar collector was modeled two-dimensionally. Figure 2 shows the various regions of the finite element model. They communicate thermally by direct contact (conduction) or by the convection/radiation heat transfer coefficients shown in the figure. This model uses a fixed thermal resistance for heat transfer into the air (i.e. the heat transfer coefficients $h_1$, $h_2$), which avoids the complexity and computational power required of a full turbulence model of the air stream. The resistance added by the second glazing due to natural convection between the two glazings is lumped into a single top loss coefficient, $U_t$. This avoids the computational complexity of modeling the convection cell in between the plates.

4.3.1. External Loss Coefficients, $U_t$, $U_b$

The loss coefficients were modeled as convective boundary conditions at the top of the first glazing and the bottom of the insulation. The finite element model uses a single upper thermal resistance (akin to a single inner glazing) which accounts for the thermal resistance of natural convection between the glazings, the forced convection due to wind over the top glazing, and radiative transfer between the glazings and the upper glazing and the sky. The conduction resistance in the second glazing is negligible compared to the other resistances in the cover system. The presence of temperature-dependent radiation and natural convection heat transfer makes this convection coefficient temperature dependent. Boundary conditions in ADINA can be made dependent on the boundary temperature and some fixed ambient temperature.

Since the model only contains the inner glazing, and to avoid modeling the fluid mechanics of the natural convection cell, the temperature of the outer glazing must be estimated in order to calculate the various heat transfer coefficients comprised in the top loss coefficient, $U_t$. To estimate this temperature, the average temperature difference between the glazings from the collector without storage from [29] was used. Since this collector operated at the peak solar radiation expected in a desert climate (900 W/m²), this value was a slight overestimate of the temperature. However, this does not have a large effect on the accuracy of the final result as the natural convection and radiation heat transfer coefficients are loosely dependent on the temperature difference between the plates. This temperature was found to be 15.1°C lower than the top plate. Using this method, $U_t$ is tabulated as a function of the inner
cover temperature, and can be entered into ADINA as a temperature dependent boundary condition. $U_b$ is constant and it is entered as a simple convective boundary condition.

4.3.2. Airflow Model

The bulk temperature of the air stream was calculated without simulating the cross-channel velocity distribution. Instead, heat transfer coefficients $h_1$ and $h_2$ were calculated on the upper and lower surfaces of the channel and these were applied as thermal resistances to the top and bottom. In this design there is a thermal entry length were the heat transfer coefficient is within 20 percent of the fully developed value in 10 hydraulic diameters [20], or 60 cm for this collector. For a 10 m long collector, this length is 6% of the total collector length and the flow can be approximated as thermally fully developed along the entire length. Assuming the flow is mostly fully developed and the heat transfer resistance constant, it is easy to represent these resistances as thin (0.1 mm thick) layers of material between each plate and the air stream of arbitrary conductivity. Figure 3 shows the location of these synthetic materials, indicated by the arrows. They contain the entire temperature gradient between the absorber and bulk air or glazing and bulk air, as shown in the figure. The air is then modeled as a slug flow at a bulk temperature $T_f$. The materials communicate with the plate via conduction and have a conductivity set to match the thermal resistance.

4.3.3. Radiation Heat Transfer Between the Absorber and Glazing

Radiation heat transfer between the absorber and upper glazing can be modeled as a linearized heat transfer coefficient. It is not dependent on the air temperature, only the temperatures of the two plates, and therefore it cannot be easily wrapped into the synthetic materials that encapsulate the convective heat transfer. However, if the bulk fluid temperature is used, the radiation heat transfer can be separated into two linearized coefficients by Equation 8:

$$h_r(T_a - T_{c1}) = h_r(T_a - T_f) + h_r(T_f - T_{c1})$$

With this equation, $h_r$ can be added to the constant convection coefficient and wrapped into the conductivity of the synthetic layer that represent the convection coefficients. The only difference is that $h_r$ is dependent on both plate temperatures, but coefficient attached to one thin material can only be made dependent on the temperature of that material. Therefore it is necessary to estimate temperature difference between the plates, $\Delta T$. The temperature difference $\Delta T$ was obtained from the analysis of the collector without PCM [29] and computing the average temperature difference between the absorber plate and the inner glazing. This value can be used to calculate $h_r$ as defined by Equation 9:

$$h_r = \sigma F[T^2 + (T_a - \Delta T)^2](2T_a - \Delta T) \quad \text{on the absorber plate side} \quad (9a)$$

$$h_r = \sigma F[T^2 + (T + \Delta T)^2](2T + \Delta T) \quad \text{on the glazing plate side} \quad (9b)$$

As shown in Figure 3 there is a temperature gradient in the synthetic materials between the plate temperature and the air temperature. Thus, any temperature dependent thermal...
resistance will vary through the thickness of the synthetic layers. Since the radiation heat transfer coefficient is only a function of a plate temperature, this is only an approximate solution to the radiative transfer; however the temperature difference between the top and bottom of these thin materials varies by only 5 K on average, and the radiation heat transfer coefficient is much more strongly dependent on the absolute temperature of a plate than temperature difference between the two plates; as a result the effect of that temperature difference is averaged with an error in the radiation heat transfer coefficient, \( h_r \), of no more than 4%.

5. Simulation Results

A collector with the cross section in Figure 1 was sized to produce air at 55-60 °C from inlet air at 30 °C. The collector length required to produce this temperature and store enough energy to produce heated air at night was obtained with a simple energy balance on the collector; equating the total solar energy absorbed in a day to the total energy used to heat the air for 24 hours at a given flow rate. From this balance a collector length of 10 m was obtained for the simulation.

5.1. PCM Depth Optimization

To estimate the necessary PCM depth it was assumed that the PCM can be described as a lumped parameter governed by the following differential equation:

\[
\frac{d}{dt} V_{PCM} = S A_p - \dot{m} c_p (T_{out} - T_{in}) - (U_t + U_b) A_p (T_a - T_{amb})
\]

where terms are defined in nomenclature. The differential equation is an energy balance where the change in energy in the PCM (extent of melting) is equal to the solar radiation absorbed less the energy removed by the air and energy lost to the environment. Recognizing that the PCM volume is \( A_p d_{PCM} \), where \( d_{PCM} \) is the PCM depth, and that the first order equation gives rise to a time constant \( \tau \), an estimate for the depth can be derived.

\[
d_{PCM} = \frac{\tau}{\rho_{PCM} h_{sf}} \left[ S - \dot{m} c_p (T_{out} - T_{in}) - (U_t + U_b) (T_a - T_{amb}) \right]
\]

Since the inlet temperature is taken to be the ambient temperature and the plate temperature is close to the fluid outlet temperature (due the stabilizing effects of the PCM) each temperature difference can be approximated to be on the order of \( (T_{melt} - T_{amb}) \). This gives a first estimate of the PCM thickness of order 10 cm.

To optimize the PCM depth, the ADINA simulation was run at the slowest possible flow rate to maintain turbulent flow through the collector (46.8 kg/hr) for 6 different PCM depths on the order of 10 cm: 4 cm to 14 cm in 2 cm increments. Figure 4 shows the temperature profile for several PCM thicknesses at a mass flow rate of 46.8 kg/hr. This shows that the 8 cm depth gives the most consistent temperature output with the least amount of PCM, with temperature output becoming less sensitive to PCM thickness as it increases beyond this depth.
For all the subsequent simulations the optimal PCM thickness of 8 cm was used.

5.2. Outlet Temperature Stabilization

The ADINA model was initially run for 96 hours using the dimensions shown in Figure 1 with various mass flow rates, from 0.013 to 0.052 kg/sec (46.8 to 186.8 kg/hr), representing Reynolds numbers in the duct from 4000 to 16000 in steps of 2000. Figure 5 shows the outlet temperature over time for two consecutive days after warm up transients have been allowed to die out during the first two days of operation. The figure shows that a sustained air output temperature can be obtained by slowing the mass flow rate, which lowers the Reynolds number, allowing more energy to be stored for use at night. The sustained output temperature is near the melting point of the phase change material. The finite element simulation for $\dot{m} = 46.8$ kg/hr shows minimal superheating of the PCM.

However, with sustained output temperature comes a trade-off in efficiency. Figure 6 shows the efficiency as a function of the time averaged normalized gain. The efficiency displays a logarithmic drop-off as shown by the best fit curve with $r^2 = 0.99$ that is shown on the graph. It can be seen that to gain a more constant operating temperature throughout the night the collector has to operate very inefficiently. This is because the collector runs hotter for a longer time when producing a consistent temperature output, so the integrated heat loss over the run period is larger.

5.3. Variation with Latent Heat of Solidification

Phase change materials are available at a variety of latent heat capacities and melting temperatures. They range in latent heats of solidification from 120 kJ/kg to 300 kJ/kg for materials which sufficiently high melt temperatures for HD cycle operation [10, 26]. Selection of the melt temperature is dependent on the top temperature required by the HD cycle. Figure 7 shows the effect of latent heat of the PCM on outlet temperature with the outlet temperature becoming more stable as latent heat goes up. However with a greater latent heat a smaller amount of PCM is melted as seen in Figure 8. Values greater than 1 and less than 0 mean the PCM is completely melted, or completely frozen respectively.

5.4. Variation with Increased Metal Matrix Solidity

Increasing the relative density of the metal matrix allows more heat to be conducted through the metal and deeper into the storage layer. It also increases lateral conduction in the streamwise direction. The end effect is that the plate temperature is more stable and more energy is stored in the latent heat resulting in a more stable outlet temperature. This is shown in Figure 9, where the temperature clearly becomes more stable with increasing relative density.

5.5. Variation with Changes in Surface Roughening

Decreasing the surface roughening decreases the heat transfer coefficient to the air and increases the temperature difference between the air and the absorber plate, which is held close to the melt temperature of the PCM. Figure 10 shows how the outlet temperature varies as a function of the transverse rib spacing. In this case, increased temperature stability goes
with increased spacing, or lower roughness. This allows less heat to transfer to the air stream and more to be stored in the PCM, thus decreasing the dip in temperature at the end of the night. When removing the ribs altogether, the equivalent surface roughness and thus heat transfer coefficient decreases significantly; by about a factor of 5. In this case the night time outlet temperature is significantly lower than the melt temperature, and the collector still has large thermal transients after 3 days. This also suggests that, as the collector runs for more consecutive days, significant superheating of the PCM will occur, resulting in lower temperature stability.

6. Experimental Validation

The numerical model was validated using a scaled-down experimental version of the solar collector tested in Dhahran, Saudi Arabia. Due to the nature of the collector’s design and size and weight constraints for the testing site, the length of the experimental collector was reduced to 1 m from 10 m with the cross section remaining the same. Since this collector was reduced in length, and a minimum flow rate is required to achieve turbulent flow, the collector will not collect enough energy to heat the air to a temperature as high as the longer collector described in the previous section. As a result the air flow will maintain a lower plate temperature than the melting point of the wax previously described. Instead of using the higher melting point wax that was used in the simulation the lowest melting point wax which was commercially available was selected. Due to a variety of compounds in this wax, melting occurred the range of 43-53 °C. This melting range was confirmed experimentally by heating the wax and observing how quickly it cooled over time. The absorber plate in the experimental collector would collect enough energy at the minimum air flow rate to allow for higher plate temperature, creating the thermal gradient necessary to melt the wax. To validate the numerical model presented earlier using the experimental data, new numerical simulations were run for the 1 m length of the experimental unit with the lower melting temperature wax. However, all other aspects of the model remained the same as for the longer collector described in the prior section.

6.1. Design and Construction

As with the simulated collector, the actual absorber plate and glazings area made of glass and metal. The absorber plate was machined and included transverse rib roughening for its simplicity of manufacturing. The roughening specifications were identical to the simulated collector given in Table 4. The top of the absorber was coated in spray paint pigmented with carbon black for high solar absorptivity (α = 0.96, ε =0.88).

The glazings were mounted in wooden guide rails. The glazings were ordinary window glass (Glazing Extinction Coefficient = 32), due to the unavailability of low iron solar glass in Saudi Arabia. Wood was chosen for its low thermal conductivity to avoid conducting heat into the glazings. The grooves were oversized to avoid cracking the glazings under differential thermal expansion. The wooden rails were mounted with screws onto a structural aluminum box frame. This frame is separated from the hot absorber plate by 25.4 mm thick piece of wood acting as thermal break.
To enhance the thermal conductivity of the wax, a folded aluminum mesh was employed. This mesh is made of high conductivity 6061-T6 aluminum. It was folded into an accordion pattern giving a total relative density of \( s = 0.087 \). As a result of its two dimensional structure the fraction of the conductivity of the mesh in Equation 3 was changed from 1/3 to 1/2. The wax was chosen to have similar properties as those used in the simulated collector and listed in Table 3, except the wax melted over a range of 43-53 °C.

Springs are used to compress the entire assembly upward and to compensate for expansion and contraction during freezing and melting. They are sized to provide a force equivalent to 2.5 times the dead weight of the steel and wax combined when mounted. Figure 11 shows a schematic diagram of the wax and mesh as it fits into the whole assembly. Figure 12 shows its physical realization.

The whole assembly is mounted on a modular frame that retains the springs and enables the insulating of the bottom section as shown in Figure 13. Fiberglass insulation \((k = 0.035 \text{ W/mK})\) is used and then covered by a reflective thermal radiation shield.

Temperature sensing was done with T-type thermocouples, manufactured with wire having special limits of error and using a standard calibration curve, which can be easily interfaced with data acquisition systems at the test site. Three evenly spaced thermocouples were installed in Delrin holders on the inlet and outlet, and nine thermocouples were embedded in the plate to read the plate temperature.

6.2. Experimental Results

To prepare the solar collector for measurements, it was equipped with a rolling base and inlet and outlet ducts made of aluminum. The ducts had inner dimensions of 30 cm by 3 cm to match the inlet and outlet dimensions of the solar collector. The solar collector along with its inlet and exit ducts were installed at an angle of 26° to the horizontal. The whole unit is oriented to the South. A centrifugal blower, was attached to the inlet and equipped with a voltage regulator so that the incoming air flow rate can be varied precisely across a wide scale. However, to match the simulated results for the sake of comparison, the measured range was limited to 1.6 to 2.2 m/s.

A flow straightener was used at the inlet of the intake duct to guarantee uniform flow into the solar collector. A pyranometer (self powered, unamplified model SP-110 by Apogee) measuring short wave radiation was connected at the same slope to as the collector to read solar radiation flux \((W/m^2)\) on the inclined surface (for the data set shown in Figure 17(a) irradiation was recorded at a nearby weather station.) A velocity meter (Air/Gas Velocity Transmitter Series EE75 Type: EE75-VTC 635k 1000c12/BN-V12-T15 with ISO Calibration Type EE31/3V, 3 Calibration Points) was installed, and its reading was taken at several locations across a perpendicular plane to the flow direction so that an average velocity is measured. Velocity measurements were confirmed using a pitot-static tube connected to an inclined manometer/digital readout. This measurement is used to determine the air flow rate across the unit. A photograph of the unit under measurement conditions is depicted in Figure 14.

To prepare the unit for the experiment, the instruments were connected to a data acquisition system (DAS; NI SCXI-1000 12 Slot Chassis). Readings were obtained for three
inlet air temperature values and three exit air temperature values, in addition to 9 readings of the temperature of the absorber plate at different locations along the length and across the width. These fifteen readings were made through the use of T-type thermocouples as indicated earlier. The thermocouples were connected to the DAS through a thermocouple amplifier/terminal block arrangement (SCXI 1102 32-channel thermocouple amplifier, signal conditioning module for thermocouples along with SCXI 1303 32-channel isothermal terminal block). In addition, another T-type thermocouple was connected to measure the ambient air temperature. The pyranometer was connected to the DAS and its output reading was converted into a heat flux using the calibration relation provided by the manufacturer. The calibrated air velocity sensor was connected to the DAS through (SCXI-1338 8 channel Current Input Terminal Block which connects current inputs to the SCXI 1120 Module to be connected to the velocity sensor). The DAS is operated using a LabView program that was prepared for recording the measured temperatures, solar radiation flux, and air speed at an interval of 5 minutes.

Once the unit was connected, it was left to run for about three days before the recorded measurements are taken, in order to overcome the initial transient effects and to confirm reliable operation of the unit. Then, the experiment was run at steady state for a period of a week.

Input data, the measured global irradiation, ambient temperature, and inlet temperature were provided as inputs to the model (adjusted for the length of the experimental collector). A ambient “sol-air” temperature was calculated from the dew point temperature [29]. Weather data, such as the dew point temperature and average windspeed were used from data recorded at the nearby Dhahran airport [31]. All these above quantities, including the sol-air ambient temperature were provided to the finite element model. The measured and simulated outlet temperature profiles were then compared.

Data was taken for two different cases. One was a cloudy winter day as shown in Figure 15(a), and on a clear day as shown in Figure 16(a). The figures show the experimental and simulated temperature profiles along with the ambient sol-air temperature.

To validate the phase change characteristics of the wax data was taken for a clear sunny day (high irradiation) with a high inlet temperature. The outlet temperature for this case lies within the melting range of the experimental PCM. The absorber plate reached temperatures over 50 °C during the middle of the day.

The difference between the simulation and experiment is shown for the cloudy and clear days in Figures 15(b) and 16(b) respectively, and is defined in Equation 12:

\[
\text{difference} = (T_{\text{out,sim}} - T_{\text{in}}) - (T_{\text{out,exp}} - T_{\text{in}})
\]

Experimental measurement error is on the order of 1 °C, primarily due to the measurement error of the thermocouples at the relatively low temperatures measured.

The results show very good agreement, especially for the clear day with low inlet temperature. The difference between the simulation and experiment increases on the warm-up and cool-down period, primarily due to the fact that the model does not account for the sensible heat capacity of the framing components of the experimental collector. This capacity cre-
ates a temperature lag of the simulation relative to the experiment when warming up and cooling down. This is most apparent when $\partial T/\partial t$ is large for extended periods of time. The extra heat capacity is also evident on the cloudy day when irradiation fluctuates and the temperatures show a small amount of lag. Overall the experiment fully validates the heat transfer approximations made for the finite element model and the model’s overall accuracy.

For the case with high irradiation and air temperature (shown in Figure 17(b)) for which wax melting occurs, the peak temperature of the air is accurately captured by the simulation, although agreement is less good later in the day. The difference late in the day may be the result of some separation of the wax layer from the absorber plate, which would impede melting of the wax leading to an over-prediction of the model relative to the experimental device. This separation could result from air leaks into the wax chamber, and due to the tilt of the collector most of the separation would be at the inlet where the plate temperature is lower, allowing the wax toward the outlet end to still heat the plate. Despite this, the collector maintains a temperature difference between the inlet and outlet throughout the night with no solar irradiation input indicating energy storage that exceeds the sensible heat that could be stored in the collector components or wax. Sensitivity to other potential differences between the simulation and experiment, such as changes in mass flow rate and reduction in solar irradiation between measurement points from intermittent cloudiness is minimal.

7. Conclusions

An air heating solar collector with integrated phase-change material storage has been designed and optimized. The computational results have been validated by experiment. A novel configuration of phase change material has been presented.

Built-in latent heat energy storage has shown great promise to maintain a consistent temperature output throughout the entire day and night. A finite element model is used to assess the performance of a built-in storage system with paraffin wax and an embedded aluminum matrix to enhance the wax layer conductivity. This case of a phase change material allows the plate temperature to be stabilized and the outlet temperature to be roughly limited to the melt temperature of the phase change material. Optimization of surface roughness characteristics can lead to greater temperature stability, and increasing the conductivity of the PCM layer though a metal mesh increases PCM utilization and temperature stability because energy can penetrate deeper into the layer. However, these benefits come at the cost of thermal conversion efficiency requiring large collector areas for small heat outputs, and has the potential to increase the cost of a large scale system, suggesting that this method of energy storage may not be the workable at large scales. If the cost of such a system can be reduced then this device has the potential to enhance the solar driven humidification-dehumidification distillation cycle by eliminating transients and warm up time associated with other system components by maintaining the top temperature of the system at an optimal operating temperature, thereby increasing overall system performance and water production.
8. Acknowledgements

The authors thank Alexander Guerra for his help with the FEM software, P. Gandhidasan for helpful discussions of the solar design, and also M. K. Adham for his assistance in preforming the experiments. The authors would also like to thank the King Fahd University of Petroleum and Minerals for funding the research reported in this paper through the Center for Clean Water and Clean Energy at MIT and KFUPM.
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Table 1: Constant parameters for simulating baseline design of collector with storage

<table>
<thead>
<tr>
<th>Constants</th>
<th>Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ambient Wind Speed</td>
<td>5 m/s</td>
</tr>
<tr>
<td>Latitude</td>
<td>26.4 °</td>
</tr>
<tr>
<td>Solar Declination</td>
<td>23 °</td>
</tr>
<tr>
<td>Collector Tilt Angle</td>
<td>26.4 °</td>
</tr>
<tr>
<td>Collector Inlet Temperature</td>
<td>30 °C</td>
</tr>
<tr>
<td>Ambient Air Temperature</td>
<td>30 °C</td>
</tr>
<tr>
<td>Dew Point Temperature</td>
<td>4 °C</td>
</tr>
<tr>
<td>Insulation Conductivity</td>
<td>0.02 W/mK</td>
</tr>
</tbody>
</table>

Table 2: Baseline values of material properties

<table>
<thead>
<tr>
<th>Material Properties</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Glazing Refraction Index</td>
<td>1.526</td>
</tr>
<tr>
<td>Glazing Extinction Coefficient</td>
<td>4</td>
</tr>
<tr>
<td>Absorber Solar Absorptivity</td>
<td>0.94</td>
</tr>
<tr>
<td>Glazing IR Emissivity</td>
<td>0.92</td>
</tr>
<tr>
<td>Absorber IR Emissivity</td>
<td>0.86</td>
</tr>
<tr>
<td>Metal Matrix Relative Density</td>
<td>0.07</td>
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</table>

Table 3: Material properties of paraffin wax storage material

<table>
<thead>
<tr>
<th>Material Properties</th>
<th>Solid</th>
<th>Liquid</th>
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</thead>
<tbody>
<tr>
<td>Heat Capacity [J/kgK]</td>
<td>2950</td>
<td>2510</td>
</tr>
<tr>
<td>Density [kg/m³]</td>
<td>818</td>
<td>760</td>
</tr>
<tr>
<td>Thermal Conductivity [W/mK]</td>
<td>0.24</td>
<td>0.24</td>
</tr>
<tr>
<td>Latent Heat Capacity [J/kg]</td>
<td>226,000</td>
<td></td>
</tr>
<tr>
<td>Melting Temperature</td>
<td>58 °C</td>
<td></td>
</tr>
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</table>
Table 4: Roughening parameters

<table>
<thead>
<tr>
<th>Roughening Parameters</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rib Height, $h$</td>
<td>0.0032 m</td>
</tr>
<tr>
<td>Rib Pitch, $p$</td>
<td>0.02 m</td>
</tr>
<tr>
<td>$p/h$</td>
<td>6.3</td>
</tr>
<tr>
<td>Roughening Regime</td>
<td>Fully Rough</td>
</tr>
</tbody>
</table>

Table 5: ($\tau\alpha$) product values for each component of solar irradiation

<table>
<thead>
<tr>
<th>($\tau\alpha$) Product</th>
<th>Value</th>
<th>Effective Angle</th>
</tr>
</thead>
<tbody>
<tr>
<td>Beam</td>
<td>0.78</td>
<td>22.5 °</td>
</tr>
<tr>
<td>Diffuse</td>
<td>0.71</td>
<td>57 °</td>
</tr>
<tr>
<td>Ground Reflected</td>
<td>0.36</td>
<td>77 °</td>
</tr>
</tbody>
</table>

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Figure 6: Time-averaged collector efficiency vs. time averaged normalized gain with curve fit

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Figure 14: Collector experimental setup on site.
(a) Temperature profiles for the solar collector throughout the day. Solar irradiation shown on left axis.

(b) Difference between simulated and measured inlet-outlet temperature difference.

Figure 15: Simulated and experimental collector temperature for a cloudy day.
(a) Temperature profiles for the solar collector throughout the day. Solar irradiation shown on left axis.

(b) Difference between simulated and measured inlet-outlet temperature difference.

Figure 16: Simulated and experimental collector temperature for a sunny day.
Figure 17: Simulated and experimental collector temperature for a sunny day with elevated inlet temperature.