Optimisation of a desiccant cooling system design with indirect evaporative cooler

Goldsworthy, M\(^1\) and White, S\(^2\).

ABSTRACT

Solar desiccant-based air-conditioning has the potential to significantly reduce cost and/or greenhouse gas emissions associated with cooling of buildings. Parasitic energy consumption for the operation of supply fans has been identified as a major hindrance to achieving these savings. The cooling performance is governed by the trade-off between supplying larger flow-rates of cool air or lower flow-rates of cold air. The performance of a combined solid desiccant-indirect evaporative cooler system is analysed by solving the heat and mass transfer equations for both components simultaneously. Focus is placed on varying the desiccant wheel supply/regeneration and indirect cooler secondary/primary air flow ratios. Results show that for an ambient reference condition, and 70ºC regeneration temperature, a supply/regeneration flow ratio of 0.67 and an indirect cooler secondary/primary flow ratio of 0.3 gives the best performance with $COP_e > 20$. The proposed cooling system thus has potential to achieve substantial energy and greenhouse gas emission savings.

KEYWORDS

Desiccant wheel; evaporative system; optimisation; air conditioning.

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<td>a</td>
<td>Channel height</td>
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<td>Desiccant channel width</td>
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<td>Desiccant wall thickness</td>
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<td>$c_p$</td>
<td>Specific heat capacity</td>
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<td>f</td>
<td>Friction factor</td>
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<td>h</td>
<td>Heat transfer coefficient</td>
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<td>$h_m$</td>
<td>Mass transfer coefficient</td>
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<td>k</td>
<td>Thermal conductivity</td>
<td>Wm$^{-1}$K$^{-1}$</td>
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<td>$m$</td>
<td>Mass flow rate</td>
<td>kgs$^{-1}$</td>
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<td>u</td>
<td>Channel velocity</td>
<td>ms$^{-1}$</td>
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<td>A</td>
<td>Desiccant channel cross-sectional area</td>
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<td>$COP_i$</td>
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<td>Surface diffusivity</td>
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<td>Heat of vapourisation</td>
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<td>Heat of adsorption</td>
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<td>Le</td>
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<td>-</td>
<td>-</td>
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<tr>
<td>NTU</td>
<td>Number of transfer units</td>
<td>-</td>
<td>surf Desiccant-air surface</td>
</tr>
<tr>
<td>Nu</td>
<td>Nusselt number</td>
<td>-</td>
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<td>Total cooling capacity</td>
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<tr>
<td>R</td>
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</tr>
<tr>
<td>Re</td>
<td>Reynolds number</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>S</td>
<td>Desiccant primary stream face area fraction</td>
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</tr>
<tr>
<td>$S_{IEC}$</td>
<td>Indirect cooler secondary stream mass flow fraction</td>
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1. **Introduction**

Desiccant cooling can be an environmentally attractive alternative to conventional mechanical air-conditioning. The desiccant cooling cycle contains no harmful synthetic refrigerants and it can be driven by low greenhouse gas emissions solar or low grade waste heat sources. This enables the displacement of fossil fuel derived electricity that would otherwise be used in a conventional system. In addition to greenhouse gas emissions savings, thermally driven desiccant cooling can potentially reduce peak electricity demand and associated electricity infrastructure costs.

Compared with other thermal cooling technologies, desiccant cooling appears to have potential for significant cost reduction. It contains only a small number of simple, robust components, there are no dangerous materials, it operates at atmospheric pressure and the control system is relativity straightforward. However, key challenges facing developers include:

- **Minimising parasitic electricity consumption.**
  
  Recent experience with thermal cooling applications (Ouazia et al., 2009; Thur and Vukits, 2009) has shown that parasitic electricity consumption from fans and other ancillary equipment can be significant compared with those for conventional airconditioners. Indeed if the parasitic electricity consumption of the thermal cooling system is not significantly less than the total electricity consumption of a conventional system, then there is no justification to change from the existing technology. Desiccant cycle selection and flow optimisation are important opportunities for minimising electricity consumption at the design stage.

- **Cost reduction.**
  
  Thermal cooling technologies generally incur higher initial cost than the equivalent conventional system. Cost reduction potential can be achieved through simplified cycle selection, size reduction and increasing thermal efficiency. The latter is particularly important where capital intensive solar collectors are used to provide the required heat. Size reduction can be achieved through optimising air flow-rates and velocities.
Minimising building supply air temperature.

On hot days, the temperature of cool air exiting the desiccant process (building supply air) may not be significantly lower than that desired in the occupied space. Consequently, the system may not be able to achieve the desired comfort conditions unassisted. The application of indirect evaporative coolers (wetted surface heat exchangers) provides opportunities for reducing the supply air temperature from the desiccant cooling process thereby minimising the backup cooling requirement (White et al., 2009).

In attempts to address these challenges, a number of authors have considered numerical models of either desiccant wheels or indirect evaporative coolers. Solutions of the heat and mass transfer partial difference equations for the individual components (Ge et al., 2008; Erens and Dreyer, 1993) have provided fundamental insights into the respective components, but do not address the cycle performance. Simplified decoupled models (e.g. Maclaine-cross and Banks, 1972; Maclaine-cross and Banks, 1983; Stabat and Marchio, 2008), of a specific system have been combined with varying operating conditions to estimate annual energy savings (Charoensupaya and Worek, 1988; Smith et al. 1993). However, in these studies, the heat and mass transfer models remain decoupled from the system model. To date there has been little focus on simultaneous optimisation of design parameters for each of the key components in a complete model of the cooling cycle. Furthermore, the impact of electrical energy consumption and the trade-off between delivered air volume and coldness has not received sufficient attention.

In this study, a simplified desiccant cooling cycle has been selected and a multi-variable optimisation performed to determine both the preferred size and operating points for the key components, and the flow-rates of each air stream in the system. The chosen cooling cycle incorporates a desiccant wheel and an indirect evaporative cooler, and has been selected with the aspiration of reducing capital cost, minimising electricity consumption and providing the potential for lower supply air temperatures.

2. Desiccant Cooling Cycle Description

The selected cooling cycle is illustrated in Figure 1. The cycle combines a desiccant dehumidification wheel with an indirect evaporative cooler and a heat source. A fan blows ‘supply’ air from outside
through the wheel (1p → 2p) where it is dried and slightly heated. This air then passes through the primary channels of the indirect cooler (2p → 3) where it is sensibly cooled by heat exchange with an evaporatively cooled air stream diverted from the main supply stream (3 → 5). Depending on building conditions, the supply air may be further cooled via a direct evaporative cooler (not shown) before it is passed into the conditioned space.

For continuous operation, the water absorbed from the supply air must be removed from the wheel. This is accomplished by passing a second air stream, the ‘regeneration’ air, through the wheel (1r → 2r), which rotates sequentially through the dehumidification and regeneration sections. The regeneration air stream must be hotter than the primary air stream. This can be achieved by heating the regeneration air with a low greenhouse gas emissions heat source such as solar heat or low grade waste heat. Regeneration air has the same humidity as the supply air as they are both sourced from ambient.

This system is similar to those proposed by other researchers (Jain and Dhar, 1996), except that there is no heat recovery exchanger. Particularly at low regeneration temperatures, the absence of the heat recovery heat exchanger has the potential to reduce pressure drop and associated fan electricity consumption, without significantly reducing thermal performance.

3. Desiccant Cooling Cycle Modelling

Key design and operational variables that have potential for optimisation include;

- The fraction \( S \) of the desiccant wheel face area that is used for dehumidifying supply air. This is typically between 0.5 and 0.75 in commercial wheels. Increasing \( S \) produces a higher flow rate of cool air to the occupied space but leads to an increase in the humidity and temperature.

- The fraction \( S_{IEC} \) of primary air leaving the indirect evaporative cooler that is removed as a side-stream and used to evaporatively cool the secondary side of the indirect cooler. Decreasing \( S_{IEC} \) produces a higher flow rate of cool air but leads to an increase in the supply temperature.
• The wheel velocity \( u_p \). Increasing \( u_p \) increases the flow-rate of supply air. However, the penalty is an increased pressure drop, increased fan electricity consumption and increased supply air humidity and temperature.

• The wheel length \( L_d \). Increasing \( L_d \) leads to improved dehumidification and colder supply air. However, this results in an increased pressure drop with a concomitant increase in fan electricity consumption.

• The regeneration temperature \( T_r \). Increasing \( T_r \) leads to improved dehumidification and colder supply air. This enables the flow-rate (and thus the fan electricity consumption) to be reduced for a given cooling load. However, the additional heating corresponds to a reduction in thermal efficiency.

• The rotational speed \( \omega \). Higher rotational speeds generally result in improved dehumidification of the supply air but increased transfer of heat from the regeneration to supply air streams.

It is apparent that for each of these parameters, a maximum cooling effect is expected where additional coldness is balanced by limits to either supply air flow, electricity consumption or regeneration heat demand. A heat and mass transfer model of the complete system incorporating both the desiccant wheel and indirect evaporative cooler has thus been constructed. The model has been used to simultaneously optimise the cooling cycle design with respect to the parameters \( S, S_{IEC}, u_p, L_d, T_r \) and \( \omega \). The complete model incorporated two sub-models; (i) a desiccant wheel heat and mass transfer model and; (ii) an indirect evaporative cooler heat and mass transfer model.

3.1. Desiccant wheel sub-model

A desiccant wheel consists of a large number of axis-symmetric air channels. Supply air flows through a specified fraction of the channels and regeneration air flows counter-current in the remainder. As the wheel rotates, the air flow through each channel cycles between the two streams. The channel walls consist of a water adsorbent material such as silica-gel and are shaped to obtain a high surface area to volume ratio. The desiccant material may be mechanically supported by another (non-adsorbent) material, in which case, there is no mass transfer between channels. Typically for modelling purposes it is also
assumed that the channels are thermally insulated from each other and hence, that each may be considered in isolation. Although circumferential temperature gradients occur near the supply/regeneration changeover, this assumption is reasonable because the circumferential Fourier time for thermal diffusion through the channel walls is small in comparison to the rotation rate. Since the channel air flows are laminar (typically $Re \sim 250$) correlations are available for transfer coefficients and hence, the air velocity, enthalpy and moisture profiles through the channel can be approximated using cross-section mean values. By neglecting channel curvature effects, either axial flow and lumped solid resistance (i.e. 1-D) models, or axial flow and 2-D solid diffusion models are appropriate.

### 3.1.1. Two dimensional desiccant wheel model formulation

A 2-D model of a desiccant channel walls is combined with a one-dimensional model of the air flow in the channels. A sinusoidal air channel cross-section with width $b$ and height $a$ is assumed. The channel length is $L_d$ and the desiccant wall (half-thickness) is $c$. Since $c$ is small in comparison to the overall radius of the wheel, radial curvature effects are ignored.

The desiccant material has a pore structure. The moisture diffusion through and along the wall is due to a combination of (i) gas phase ordinary diffusion and (ii) Knudsen diffusion within the pores of the desiccant and (iii) surface diffusion along the pore surfaces at desiccant/air interfaces. According to Pesaran and Mills (1987) for regular density silica gel, the average pore radius is $11A$ and the ratio of gas phase to surface diffusion is 0.023-0.06. Hence, only surface diffusion need be considered, and thus it is approximate to consider the desiccant as a homogenous material with an overall moisture diffusivity $D$.

For typical wheel parameters (Table 1) $Bi << 1$ and $Bim > 1$. Hence, a lumped capacitance thermal model of the solid material in the radial direction would be appropriate. However, the slow moisture diffusion through the desiccant limits the rate of moisture transfer between the solid and the air stream. Consequently, a 2-D model is required to determine the moisture profile through the wall.
The system is modelled using moisture and energy conservation equations for the air stream moisture content $Y_i$ and temperature $T_i$ and the desiccant material moisture content $W$ and temperature $T_d$. The moisture transfer between the air and the desiccant surface is proportional to the difference between the air moisture content and the gas phase equilibrium moisture content at the surface $Y_{surf}$. Diffusion in the air stream is neglected because convection is the dominant transport mechanism.

\[
\frac{\partial Y_i}{\partial \tau_i} + \theta_i \frac{\partial Y_i}{\partial \eta_i} = \frac{\theta_i NTU_i}{Le_f} (Y_{surf} - Y_i)
\]  
(1)

\[
\frac{\partial T_i}{\partial \tau_i} + \theta_i \frac{\partial T_i}{\partial \eta_i} = \theta_i NTU_i (T_{surf} - T_i)
\]  
(2)

\[
\frac{\partial W}{\partial \tau_i} = F_{m,i} \frac{\partial^2 W}{\partial \eta_i^2} + F_{m,i} \frac{\partial^2 W}{\partial k^2}
\]  
(3)

\[
\frac{\partial T_d}{\partial \tau_i} = F_{o,j} \frac{\partial^2 T_d}{\partial \eta_i^2} + F_{o,k} \frac{\partial^2 T_d}{\partial k^2}
\]  
(4)

where $\Omega_p = \frac{2\pi}{\omega} S$, $\Omega_d = \frac{2\pi}{\omega} (1 - S)$

Equations (1)-(4) form a set of coupled parabolic partial differential equations. The desiccant is modelled as insulated and impermeable at the channel ends and at the boundary between channels. The energy of adsorption is assumed to be released onto the surface when moisture adsorbs onto the desiccant (also assumed to occur at the surface). The upstream air boundary condition is specified by the inlet air properties; the downstream air boundary conditions are extrapolated. The inflow condition and flow direction for each channel cycles between the regeneration and supply cases according to the wheel rotational position. The channels thus reach a cyclically steady state which is independent of the starting conditions.

Desiccant boundary conditions:

\[
\left. \frac{\partial W}{\partial k} \right|_{k=0} = 0, \quad \left. \frac{\partial T_d}{\partial k} \right|_{k=0} = 0
\]

\[
\left. \frac{\partial W}{\partial k} \right|_{k=1} = Bi_m (Y_{surf} - Y), \quad \left. \frac{\partial T_d}{\partial k} \right|_{k=1} = Bi \left( T_{surf} - T_i \right) + \frac{1}{Le_f \epsilon_{pa}} \frac{H}{Le_f} (Y_{surf} - Y)
\]  
(5)

\[
\left. \frac{\partial W}{\partial \eta} \right|_{\eta=0,1} = 0, \quad \left. \frac{\partial T_d}{\partial \eta} \right|_{\eta=0,1} = 0
\]
Air boundary conditions:

\[ \begin{align*}
\eta &= 0, \quad T_p = T_{ip}, \quad Y_p = Y_{ip} \quad \text{for } 0 < \tau_p < 1 \\
\eta &= 1, \quad T_r = T_{ir}, \quad Y_r = Y_{ir} \quad \text{for } 0 < \tau_r < 1
\end{align*} \]  

(6)

Non-dimensional parameters are defined as,

\[ \begin{align*}
\theta_j &= \frac{u_j \Omega}{L_d}, \quad NTU_j = \frac{h}{\rho_a c_{pa} A u_j} = \frac{Nu}{4} \left( \frac{P}{A} \right)^2 \frac{L_d}{u_j} \\
Bi &= \frac{h c}{k}, \quad Bi_m = \frac{h_a c}{\rho_a J D}, \quad Le_f = \frac{h}{h_a c_{pa}} = Le^{2/3} \\
Fo_{\eta,j} &= \frac{k \Omega}{\rho_a c_{pa} L_d^2}, \quad Fo_{m,\eta,j} = \frac{D \Omega_j}{L_d^2} \\
Fo_{\kappa,j} &= \frac{k \Omega}{\rho_a c_{pa} c^2}, \quad Fo_{m,\kappa,j} = \frac{D \Omega_j}{c^2}
\end{align*} \]  

(7)

Equations (1)-(4) together with the boundary conditions (Eqs. (5) and (6)) and the silica gel material and adsorption properties (Table 1) were solved numerically using the Alternating Direct Implicit method. Cyclical steady state conditions were assumed to exist once the total mass of water vapour entering a channel during the supply phase was within 0.5% of the total mass leaving the channel during the regeneration phase.

The equations were solved assuming constant \( H_{ads}, D \) and overall air and desiccant specific heat capacities. These parameters were evaluated at \( T_{ave} = \frac{1}{2}(T_{ip} + T_{ir}) \) and \( W_{ave} = 0.1 \). Sensible heat transfer associated with the moisture transfer process was assumed to be negligible. Collectively these assumptions were found to result in less than 1K change to the average supply side outlet temperature and less than 0.3g/kg\(^{-1}\) change in the average supply side outlet humidity compared with the exact solution for the base case simulation.

The simulations employed constant (average) values of \( Nu \) and \( Sh \) representative of fully developed air thermal and moisture concentration profiles throughout the channel. While larger values may be expected
in the entry length region, for the condition tested \( \frac{u_1}{L_d} > 20 \text{s}^{-1} \), the influence of entry conditions is expected to be small (Antonellis et al., 2009).

For sinusoidal channels with the specified geometry Niu and Zhang (2002) specify \( N_u_\tau = 2.14 \) for the case of constant axial surface temperature and state that typically, \( N_u_H \sim 1.3 N_u_\tau \) for the case of constant axial surface heat flux. For the same geometry Shah (1975) gives \( N_u_\tau = 2.12 \) and \( N_u_H = 2.62 \) although Kakac et al. (1987) specify \( N_u_\tau = 1.54 \) and \( N_u_H = 2.6 \). Here neither the axial surface temperature profile nor the axial surface heat flux profile were constant. Although the solid conduction rate was high \( (\frac{Bi}{<1}) \), the insulated boundary condition employed at the internal wall acted to limit the heat transfer rate to the desiccant \( (i.e. \frac{Fo_k}{<1}) \). For the moisture transfer, the slow diffusion rate \( (Bi > 1) \) limited the surface moisture gradient but the axial surface moisture concentration profile was not constant because of the coupled heat transfer. Hence, the actual values of \( Nu \) and \( Sh \) are likely to be between \( N_u_\tau \) and \( N_u_H \). Based on the more recent data, \( Nu = 2.4 \) was employed here.

For the moisture transfer, \( Le \) was evaluated at \( T_{ave} \) and the heat/mass transfer analogy employed to calculate \( Sh \). Since \( Le < 1 \), the mass transfer across the desiccant surface is enhanced relative to the energy transfer. The influence of varying \( Le \) has been investigated by Van den Bulck et al. (1985). However, in their model, the effect of solid side diffusion was included in the definition of \( Le \), and this led to \( Le > 1 \). Often it may be appropriate to assume that \( Le = 1 \). This assumption does not significantly change the predicted values of \( Y_{zp} \) and \( T_{zp} \).

3.1.2. Desiccant wheel model validation

Validation of the wheel model was achieved by comparing the predicted average supply outlet condition with experimental data (Tsutsui et al., 2008) for varying \( \omega \) and \( L_d \) for set inlet conditions. The parameters and conditions used are listed in Table 1. Simulation results are shown in Figure 2; dots represent the experimental values and solid lines the simulation results. It is evident that the simulation
model can predict the variations of average supply outlet temperature $T_{2p}$ and humidity ratio $Y_{2p}$ with $\omega$ and $L_d$ over a reasonable range of these parameters. Generally, the simulation $T_{2p}$ values are lower than the experimental values by approximately $2K$ and the agreement is closer for lower rotation speeds.

### 3.2. Indirect evaporative cooler sub-model

An indirect evaporative cooler, consists of a series of alternating dry and wet flow channels. In the dry channels, ‘primary’ air stream is cooled sensibly via heat transfer with a ‘secondary’ air stream in the wet channels. In the wet channels a flow of water along the walls evaporatively cools the secondary air stream. Multiple geometric flow-arrangements are possible including, parallel, cross and counter-current flow. For given conditions, the cooling performance may be expressed using the effectiveness defined by

$$
\epsilon = \frac{T_{2p} - T_3}{T_{2p} - T_{wb,3}}. \quad (8)
$$

Note that the denominator in Eq. (8) contains the primary channel exit wet bulb temperature. That is, the maximum effectiveness of 1 is obtained when the inlet air is cooled to its dew point temperature.

A cross flow indirect cooler design is described by Pescod (1974). The water flow in the wet air channel is in the downwards direction opposite to the secondary air flow. Both the wet and dry channels have protrusions spaced at regular intervals to improve the heat transfer by promoting turbulent flow in the channels, even though this also increases the air-flow pressure drops. The key design parameters are the channel spacings, protrusion details, channel lengths, air and water flow-rates and flow velocities.

The performance of a cross-flow regenerative indirect cooler has been analysed numerically by Erens and Dreyer (1993). For a given inlet design temperature and humidity ratio, they present a graph of the cooling capacity as a function of $S_{IEC}$ for a range of plate spacings, assuming equal spacing for the dry and wet channels. The optimum value of $S_{IEC}$ was found to be between 0.25 and 0.4 for plate spacings between 2 and 5mm. This corresponds to secondary stream flow rates significantly below those leading to the maximum effectiveness (for which $S_{IEC} \sim 1$).
Previous numerical and experimental studies of indirect cooler performance have considered inlet flow conditions representative of ambient summer cooling conditions. However, in this case, the primary air stream is sourced from the outlet of the desiccant wheel. These inlet conditions are likely to be significantly hotter and dryer than that used in previous studies. Thus, a 2-D heat and mass transfer model of the indirect cooler has been used to estimate the variation in $\varepsilon$ over a range of inlet conditions and $S_{REC}$ for a cross-flow regenerative cooler.

### 3.2.1. Two dimensional indirect evaporative cooler model formulation

The indirect cooler is shown schematically in Figure 3. Primary air flows through the right (dry) channel from the inlet at ‘2p’ to the outlet at ‘3’. A portion of the air at ‘3’ is diverted to the left (wet) channel and passes from ‘4’ to ‘5’. A water film with thickness $a_w$ runs down the side of the wet channel from ‘5’ to ‘4’. The dry channel and wet channels have thicknesses of $2a_p$ and $2a_s$, respectively.

Because the thermal resistance of the channel walls is small (the Fourier heat transfer time is large), it is appropriate to assume that the wall and water film temperatures are identical at a given location. According to Madhawa et al. (2007), the influence on $\varepsilon$ of conduction in the plane of the water/wall surface is small (<5%) for most conditions. Here all conduction effects are neglected.

Assuming that the variation in $a_w$ is small, the following five equations may be formulated expressing conservation of energy and mass of the primary and secondary air streams and the water flow at the steady state condition. Here $Y_{sat}$ is the saturated air moisture content at the local water temperature $T_w$, $T_p, Y_p$ are the primary channel air temperature and humidity ratio and $T_s, Y_s$ the secondary channel air temperature and humidity ratio. The wetability factor $\psi$ is an adjustable parameter to account for incomplete wetting of the plate surface by the water film.
\[
\frac{\partial T_p}{\partial \eta} = -NTU_p \left( T_w - T_p \right) \\
\frac{\partial Y_p}{\partial \eta} = 0
\]  
(9)

\[
\frac{\partial T_s}{\partial \zeta} = NTU_s \left[ -(T_w - T_s) + \frac{1}{Le_f} \left( Y_{sat} - Y_s \right) \frac{H_{vap}}{c_{pa}} \psi \right]
\]  
(11)

\[
\frac{\partial Y_s}{\partial \zeta} = -\frac{NTU_s}{Le_f} \left( Y_{sat} - Y_s \right) \psi
\]  
(12)

\[
\frac{\partial T_w}{\partial \zeta} = \frac{c_{pw}}{c_{pa}} \beta \left[ NTU_p \left( T_w - T_p \right) + \frac{NTU_p}{S_{IEC}} \left( T_w - T_p \right) \right]
\]  
(13)

Equations (9)-(13) form a coupled set of differential equations. For regenerative operation with a closed loop water cycle, only the inlet temperature \( T_{2p} \) and humidity ratio \( Y_{2p} \) of the primary stream and the inlet water temperature must be specified. Here the water inflow condition employed is equivalent to an external reservoir which is small in comparison to the water circulation rate and which rapidly reaches the temperature of the water exiting the flow channels. The resultant boundary conditions are:

\[
T_p \left( \zeta, \eta = 0 \right) = T_{2p}, \quad Y_p \left( \zeta, \eta = 0 \right) = Y_{2p} \\
T_s \left( \zeta = 0, \eta \right) = T_{p \left( \zeta, \eta = 1 \right)}, \quad Y_s \left( \zeta = 0, \eta \right) = Y_p \left( \zeta, \eta = 1 \right)
\]  
(14)

Non-dimensional parameters are defined as:

\[
NTU_p = \frac{h_p}{\rho_p c_{pa} a_p u_{2p}}, \quad NTU_s = \frac{h_s}{\rho_s c_{pa} a_s u_s}, \quad \beta = \frac{\dot{m}_p}{\dot{m}_w}, \quad S_{IEC} = \frac{\dot{m}_p}{\dot{m}_w}
\]  
(15)

Here it has been assumed that the total air specific heat capacity is constant, that negligible sensible heat transfer is combined with the moisture transfer and, that the variation in \( a_w \) is small. Collectively these assumptions were found to result in less than a 2% change in \( \varepsilon \) compared with the exact solution for the base case simulation conditions listed in Table 2.
The performance of the indirect evaporative cooler, measured in terms of $\varepsilon$ as defined in Eq. (8), is highly dependent on the heat transfer coefficients in the dry and wet channels $h_p$ and $h_s$. Here a correlation of the form $h = Au^b$ (Pescod, 1979) was employed with parameters as given in Table 2.

3.2.2. Indirect evaporative cooler model validation

Validation of the indirect evaporative cooler model was achieved by comparing the predicted effectiveness values with experimental data (White et al., 2009) for primary air inlet temperatures ($T_p = 303,323 K$), humidities ($Y_p = 5,10,15 g kg^{-1}$) and mass flow ratios ($S_{EC} = 0.4,0.6$). Other parameters and conditions used are listed in Table 2 and comparisons between calculated and experimental results are illustrated in Figure 5.

The large uncertainty range associated with the experimental values is due to the high number of sensors from which data was recorded to determine the cross-sectional averaged properties. The model predictions are largely within the range of experimental error, though the predicted values for $\varepsilon$ only approximately match the experimental results over the full range of conditions. In particular, the experimental results indicate a drop in performance at high inlet humidity levels. However, the presence of the desiccant wheel upstream of the indirect cooler in the system simulations insured that the 15g/kg humidity level was not reached in subsequent desiccant cycle simulations.

3.3. Desiccant cycle optimisation

The two sub-models were combined into a single open-loop cooling model with the addition of an optimisation routine used to; (i) calculate and maximise the electrical coefficient of performance $COP_e$ and; (ii) calculate and report the thermal coefficient of performance $COP_t$. The $COP_e$ was defined as the ratio of useful cooling to parasitic fan electricity consumption (Eq. (16)) and $COP_t$ as the ratio of useful cooling to regeneration heat consumption (Eq. (17)).
Here $\Delta H$ is the enthalpy change between the given state points. The useful cooling effect (numerator) depends on the reference condition below which the cooling effect is considered to be useful. In some cases, any enthalpy removal below the ambient (outside) condition is useful (for example for displacement of ventilation air). In other cases, it is only the enthalpy reduction below the indoor condition that is useful. Additional flexibility was provided in the model by allowing the user to set their preferred reference condition $T_{cool}$ and $Y_{cool}$ below which useful cooling is obtained.

In Eq. (16) the fan electricity consumption (denominator) is calculated from (i) the fan efficiency and (ii) the summed pressure drop across both the primary air flow path (desiccant wheel and indirect cooler primary air channels) and the secondary air flow path of the indirect cooler. This assumes that

- the pressure drop across the supply and regeneration sides of the wheel are equal;
- the pressure drop along the building supply air ductwork matches the pressure drop across the secondary side of the indirect cooler.

The pressure drops across each of the components were calculated using Eqs. (18)-(20), and the fan electrical efficiency $\phi$ was set to 0.8 for all calculations.

$$\Delta P_{1p-2p} = F_1 \left[ \frac{f_{IEC} \mu_a}{32} \left( \frac{P^2}{D} \right) u_{1p} L_d + \frac{1}{2} \rho_a u_{1p}^2 K_{1p-2p} \right], \quad F_1 = 1.65$$

$$\Delta P_{2p-3} = F_2 \left[ \frac{f_{IEC} \mu_a}{32} \frac{1}{a_p} u_{2p} L + \frac{1}{2} \rho_a u_{2p}^2 K_{2p-3} \right], \quad F_2 = 1.5$$

$$\Delta P_{4-5} = F_3 \left[ \frac{f_{IEC} \mu_a}{32} \frac{1}{a_s} u_s H + \frac{1}{2} \rho_a u_s^2 K_{4-5} \right], \quad F_3 = 3.5$$

Here $K_{1p-2p}$ is the minor loss coefficient for the desiccant channels. For the (laminar flow) desiccant channels, the variation in pressure drop with channel velocity is close to linear over a range of typical velocities since $f_p$ is constant and the contribution of minor losses is relatively small. For the indirect
cooler, the methodology of Pescod (1979) was employed to express the pressure drop in the turbulent channels as the combination of a friction effect between the protrusions, the drag induced pressure drop due to the protrusions and that due to minor (entrance and exit) losses. Scaling factors $F_1, F_2, F_3$ were introduced to match the theoretical expressions with experimental measurements (White et al., 2009). The particularly large value of $F_3$ arises because the theoretical expression ignored the pressure drop due to passage of air through the demister pad located at the secondary air stream exit. Predicted and experimental pressure drops are compared for an indirect cooler and an equivalent geometry silica-gel desiccant wheel (Rossington et al., 2009), as functions of the stream inlet face velocity, in Figure 5.

### 4. Procedure

The performance of the desiccant cooling system is a function of a large number of parameters. Here the system performance was optimised by varying the parameters,

$$S, S_{REC}, \omega, T, L_d, u_{tp}$$

with fixed values of the following parameters with values as listed in Table 1 and Table 2.

$$\rho_d, c_{pd}, k_d, D_p, \frac{P}{c}, c, N u, u_{te} = u_{tp}$$

$$L, H, a_p, a_s, h_p, h_s, \psi, \beta$$

Because the dimensions of the indirect cooler were fixed, the face area of the wheel was set to a constant value of $0.196m^2$. This is equivalent to setting the scaling between the wheel and indirect cooler face velocities.

In the following text, the optimum parameter values are indicated using a superscript ‘*’. The optimum value of $\omega$ (i.e. $\omega^*$) was chosen as that leading to maximum dehumidification at the desiccant supply side outlet. For all other parameters, $COP_c$ was used as the optimisation criterion. The results are presented in two parts.
1. In Section 5.1 a fixed value of \( L_d \) and \( u_{ip} \) is considered and \( \omega^*, S^* \) and \( S_{IEC}^* \) calculated for 
\[50C \leq T_r \leq 80C\] and 
\[24C \leq T_{cool} \leq 35C\] with \( Y_{cool} = 14.3 \text{gkg}^{-1} \).

2. In Section 5.2, a varying wheel length \( 0.05 \leq L_d \leq 0.4 \) and inlet channel velocity was considered and \( \omega^*, S^* \) and \( S_{IEC}^* \) calculated for 
\( T_r = 70C \) and 
\( T_{cool} = 35C \) with 
\( Y_{cool} = 14.3 \text{gkg}^{-1} \).

In all cases, a fixed inlet (ambient) condition of 
\( T_{ip} = 35C \) and \( HR_{ip} = 14.3 \text{gkg}^{-1} \) was employed.

5. Results

5.1. Fixed desiccant channel length and velocity

The optimum values of \( S \) (solid lines) and \( S_{IEC} \) (dashed lines), (denoted by \( S^* \) and \( S_{IEC}^* \)) are shown in Figure 6 as a function of \( T_{cool} \) for 
\[50C \leq T_r \leq 80C\], for \( L_d = 0.2m \) and \( u_{ip} = 3ms^{-1} \). The \( COP_e \) (solid lines) and \( COP_i \) (dashed lines) values for each case are shown in Figure 7. The following trends are apparent;

i.) as \( T_r \) increases, \( S^* \) generally increases, \( COP_e \) increases and \( COP_i \) decreases,

ii.) as \( T_{cool} \) increases, \( S^* \) increases, \( COP_e \) increases and \( COP_i \) increases,

iii.) as \( T_{cool} \) increases, \( S_{IEC}^* \) decreases,

iv.) \( S_{IEC}^* \) is relatively insensitive to \( T_r \) except at low \( T_{cool} \) for \( T_r = 50C \),

v.) a large increase in \( S^* \) occurs between \( T_r = 60C \) and \( T_r = 70C \) over a range of \( T_{cool} \).

The variation in \( COP_e \) and \( COP_i \) as \( S \) and \( S_{IEC} \) deviate from \( S^* \) and \( S_{IEC}^* \) is displayed in Figure 8. It can be seen that \( COP_e > 22 \) and within 10% of the optimum value for \( 0.55 \leq S \leq 0.85 \), and 
\[0.2 \leq S_{IEC} \leq 0.4\]. Over this range, \( 0.2 \leq COP_i \leq 0.7 \), while \( COP_i = 0.4 \) at the optimum value of \( COP_e \).
5.2. Fixed regeneration and cooling set-point temperatures

It is useful to compare the system performance for varying $L_d$ and $u_{tp}$ with fixed values of $T_r = 70C$ and $T_{cool} = 35C$. However, unlike the previous analysis (optimising $COP_e$ with fixed values of $L_d$ and $u_{tp}$), optimising $COP_e$ with varying $L_d$ and $u_{tp}$ is unhelpful because maximum $COP_e$ values correspond to negligible cooling output. This results from the strong dependence of pressure drop on $L_d$ and $u_{tp}$. It is more constructive to compare the system performance under conditions of fixed total cooling output $Q$.

The variation of $COP_e$ and $COP_i$ with $L_d$ for fixed values of $Q = 1.5, 2$ and $2.5kW$, and with $S^{*}$, $S_{BC}^{*}$ and $\omega^{*}$ is shown in Figure 9. The following trends are apparent;

i.) a trade-off exists between optimising $COP_e$ and $COP_i$,

ii.) for a given $Q$, the maximum $COP_e$ occurs at smaller $L_d$ than does the maximum $COP_i$,

iii.) the values of $L_d$ at which $COP_e$ and $COP_i$ are maximums increases as $Q$ increases,

iv.) a sharp drop in $COP_i$ occurs for small $L_d$ but only a small drop in $COP_e$.

6. Discussion

The open-loop cooling cycle performs best when any cooling effect from ambient is considered beneficial (for example when off-setting infiltration). The results show that, depending on the regeneration temperature, the optimum desiccant wheel supply/regeneration flow rate ratio for infiltration off-set cooling is between the values $S = 0.5$ and $0.75$ commonly associated with commercially available wheels. For cooler conditioning reference temperatures, small values of $S$ would be required; though in some cases, the selection of an open loop cycle may need to be re-assessed.

Irrespective of the reference condition below which the cooling effect is considered useful, when $T_r$ is fixed, (i) $COP_e$ is insensitive to $S$ over the range $0.5 \leq S \leq 0.75$ and (ii) $COP_i$ increases as $S$
increases. This indicates that a small drop in $COP_e$ may be permitted via selection of $S$ slightly larger than $S^*$ to obtain a significant improvement in $COP_t$.

The system performance was slightly more sensitive to $S_{IEC}$ for fixed $T_r$. The optimum range occurred over relatively low ratios; between $S_{IEC}^* = 0.3 – 0.4$ for the high flow, infiltration offset case and increasing to $S_{IEC}^* = 0.5$ for cooler reference conditions.

The values of $S^*$ and $S_{IEC}^*$ predicted in the desiccant wheel study of Chung and Lee (2009) and in the indirect cooler study of Erens and Dreyer (1993) were $S^* \approx 0.6$ at $T_r = 80^\circ C$ and $S_{IEC}^* \approx 0.25 – 0.4$ (for 4mm to 2mm channel spacings) respectively. The similarity with this study is interesting given that these studies (i) considered a single component in isolation, and (ii) employed different parameters and performance indicators (the moisture removal capacity and total cooling power) to those used here. In both cases the performance was evaluated using a reference condition equal to the inlet (i.e. ambient) condition for the component. The similarity suggests that in general, the values predicted here may be approximately correct for a number of systems.

Use of $COP_e$ and $COP_t$ as performance indicators highlights the trade-off which exists, particularly in solar applications, between minimising the operating costs and minimizing the capital cost. By employing an increased collector area, or higher operating temperature collectors, the regeneration temperature can in turn be increased. This increases $COP_e$ (the operating cost indicator) but lowers $COP_t$ (the capital cost indicator).

This trade-off is also evident in the comparison of varying desiccant wheel lengths and inlet velocities. The optimum length of the wheel for maximising $COP_e$, is significantly shorter than the optimum length for maximising $COP_t$. Clearly, the performance of solar air-conditioning systems should be presented in terms of both $COP_e$ and $COP_t$. 
The system value of $COP_e$ is the critical economic or environmental justification for installing a desiccant system. Such a system is only justified if the operating electricity demand is significantly below that of a conventional system (for which $COP_e$ is typically ~3-4). Optimised values for $COP_e$ were typically greater than 20 when ambient conditions are taken as the reference condition below which the cooling effect is considered useful (infiltration offset). Thus, the proposed desiccant cooling system has strong potential to achieve substantial energy and greenhouse gas emission savings when used in place of conventional systems.
REFERENCES


FIGURES

Figure 1 Flow Schematic of an open-loop solid desiccant cooling system.

Figure 2 Comparison of desiccant wheel numerical simulation results (lines) (Eq. (1)–(7)) with experimental results (symbols). Left: Average supply outlet temperature. Right: Average supply outlet humidity ratio.

Figure 3 Indirect evaporative cooler geometry for heat and mass transfer model.
**Figure 4** Comparison of indirect evaporative cooler numerical simulation results (lines) (Eq.(9) - (15)) with experimental results (symbols) for conditions and with parameters as listed in Table 1.

**Figure 5** Left: Indirect cooler pressure drops as a function of channel entrance face velocity. Experimental data (White et al., 2009) and Eqs. (19) and (20). Right: Desiccant wheel pressure drop as a function of face velocity. Experimental data (Rossington et al., 2009) and Eq. (18).

**Figure 6** Variation of optimum desiccant wheel ratio $S^*$ (solid lines) and indirect cooler ratio $S_{iec}^*$ (dashed lines) with cooling set-point temperature $T_{cool}$ for different desiccant regeneration temperatures $T_r$. 
Figure 7 Comparison of $COP_e$ (solid lines) and $COP_t$ (dashed lines) for varying values of $T_r$ and cooling set-point temperature $T_{cool}$ with optimum system parameters for each condition.

Figure 8 Contours of $COP_e$ (left) and $COP_t$ (right) for varying $S$ and $S_{IEC}$ with $T_r = 70C$ and $T_{cool} = 35C$.

Figure 9 Plot of $COP_e$ (solid lines) and $COP_t$ (dashed lines) for constant values of the total cooling output $Q$ as a function of $L_d$ for $T_r = 70C$ and $T_{cool} = 35C$ with $S^*, S_{IEC}^*$ and $\omega^*$. 
### TABLES

<table>
<thead>
<tr>
<th>Wheel material properties</th>
<th>Wheel conditions</th>
<th>Non-dimensional data</th>
</tr>
</thead>
<tbody>
<tr>
<td>$f = 0.7$</td>
<td>$T_p = 305.5K$</td>
<td>Re = 242, $Nu = 2.4$</td>
</tr>
<tr>
<td>$c_p = 921Jkg^{-1}K^{-1}$</td>
<td>$T_{tr} = 353.2K$</td>
<td>$Le = 0.87$ (CRC, 2009)</td>
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<tr>
<td>$\rho_d = 800kgm^{-3}$</td>
<td>$Y_p = 19.5gkg^{-1}$</td>
<td>$NTU = 7.47$</td>
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<tr>
<td>$k_d = 0.2Wm^{-1}K^{-1}$</td>
<td>$Y_{tr} = 11.9gkg^{-1}$</td>
<td>$\theta = 1287$</td>
</tr>
<tr>
<td>$f_d = 44.7$ (Niu and Zhang, 2002)</td>
<td>$\omega = 3–50RPH$</td>
<td>$Bi = 0.028$, $Bi_{tr} = 10.0$</td>
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<tr>
<td></td>
<td>$S = 0.5$</td>
<td>$Fo_s = 6.1 \times 10^{-4}$</td>
</tr>
<tr>
<td></td>
<td>$u_{lp} = u_{tr} = 2.86ms^{-1}$</td>
<td>$Fo_{n,\eta} = 1.6 \times 10^{-6}$</td>
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<tr>
<td></td>
<td></td>
<td>$Fo_{n,\eta} = 4.0$</td>
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### Table 1 Desiccant wheel properties, validation conditions and ‘base’ case parameters.

<table>
<thead>
<tr>
<th>Indirect cooler conditions</th>
<th>Indirect cooler construction</th>
<th>Material data</th>
<th>Non-dimensional data</th>
</tr>
</thead>
<tbody>
<tr>
<td>$T_{2p} = 303–323K$</td>
<td>$L = 0.37m$</td>
<td>$h = 32\alpha \rho h Wm^{-2}K^{-1}$</td>
<td>$Re_p = 237$</td>
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<tr>
<td>$Y_{2p} = 5–15gkg^{-1}$</td>
<td>$H = 0.55m$</td>
<td>$Y_s(T_s)$</td>
<td>$Le = 1$</td>
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<tr>
<td>$u_{wp} = 2.9–3ms^{-1}$</td>
<td>$a_p = 3.5mm$</td>
<td>$H_{vap}(T_s)$</td>
<td>$NTU_p = 3.20,3.24$</td>
</tr>
<tr>
<td>$u_d = 2.9–4.5ms^{-1}$</td>
<td>$a_r = 2mm$</td>
<td>$f_{IEC} = 96$ (Incropera, 2002)</td>
<td>$S_{IEC} = 0.4,0.6$</td>
</tr>
<tr>
<td>$m_u = 0.003Ls^{-1}$</td>
<td>$\alpha$ (per channel)</td>
<td>$\beta = 1.35,2.1$</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>$\psi = 0.3$</td>
<td></td>
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### Table 2 Indirect cooler parameters and validation conditions.