Desalination and Water Treatment
www.deswater.com
doi: 10.1080/19443994.2016.1173383

57 (2016) 23430–23442
October

Taylor & Francis
Taylor & Francis Group

Liquid desiccant dehumidification and regeneration process to meet cooling and freshwater needs of desert greenhouses

Ryan Lefers[a,*], Narasimha Murthy Srivatsa Bettahalli[b], Suzana P. Nunes[b], Nina Fedoroff[c], Philip A. Davies[d], TorOve Leiknes[a]

[a]*Water Desalination and Reuse Center (WDRC), Biological and Environmental Science & Engineering (BESE), King Abdullah University of Science and Technology (KAUST), Thuwal 23955-6900, Saudi Arabia, Tel. +966 128082273; emails: ryan.lefers@kaust.edu.sa (R. Lefers), torove.leiknes@kaust.edu.sa (T. Leiknes)*
[b]*Biological and Environmental Science and Engineering Division (BESE), King Abdullah University of Science and Technology (KAUST), Thuwal, 23955-6900, Saudi Arabia, emails: SRIVATSA.BETTAHALLINARASIMHA@kaust.edu.sa (N.M.S. Bettahalli), suzana.nunes@kaust.edu.sa (S. Nunes)*
[c]*Evan Pugh Professor Emerita, Penn State University, University Park, PA 16802, USA, email: nvf1@psu.edu*
[d]*Sustainable Environment Research Group, School of Engineering and Applied Science, Aston University, Birmingham, B4 7ET, UK, email: p.a.davies@aston.ac.uk*

Received 9 April 2015; Accepted 23 February 2016

ABSTRACT

Agriculture accounts for ~70% of freshwater usage worldwide. Seawater desalination alone cannot meet the growing needs for irrigation and food production, particularly in hot, desert environments. Greenhouse cultivation of high-value crops uses just a fraction of freshwater per unit of food produced when compared with open field cultivation. However, desert greenhouse producers face three main challenges: freshwater supply, plant nutrient supply, and cooling of the greenhouse. The common practice of evaporative cooling for greenhouses consumes large amounts of fresh water. In Saudi Arabia, the most common greenhouse cooling schemes are fresh water-based evaporative cooling, often using fossil groundwater or energy-intensive desalinated water, and traditional refrigeration-based direct expansion cooling, largely powered by the burning of fossil fuels. The coastal deserts have ambient conditions that are seasonally too humid to support adequate evaporative cooling, necessitating additional energy consumption in the dehumidification process of refrigeration-based cooling. This project evaluates the use of a combined-system liquid desiccant dehumidifier and membrane distillation unit that can meet the dual needs of cooling and freshwater supply for a greenhouse in a hot and humid environment.

Keywords: Greenhouse cooling; Dehumidification; Membrane distillation; Liquid desiccant regeneration

*Corresponding author.

Presented at EuroMed 2015: Desalination for Clean Water and Energy Palermo, Italy, 10–14 May 2015. Organized by the European Desalination Society.

1944-3994/1944-3986

1. Introduction

1.1. Global context

As the human population has grown and transportation of food has become easier over the past century, more and more people have chosen to live in areas of the world that do not have a favorable ambient environment for the growth of high-value crops such as fruits and vegetables. Transportation of food over long distances and across borders leaves regions and nations vulnerable to disruptions in food supply, a phenomenon known as food insecurity. A major food importer, the Kingdom of Saudi Arabia (KSA) has vast regions that are not favorable for growth of food in the outdoor climate. Factors that influence the poor production potential in these regions include intense heat, lack of freshwater for irrigation, and poor soil quality.

Growing fruits and vegetables in greenhouses offers a promising solution for regions that are not able to support their outdoor growth. Growing such high-value crops in greenhouses increases the potential for harvest per unit area of land by as much as 20-fold. Despite these advantages, the use of greenhouses in hot climates faces the challenge of providing plants access to solar photosynthetic energy, while rejecting or removing the solar heat energy. For example, Jeddah, the KSA's second-largest city, has an outdoor climate that is only favorable for growth of tomatoes for 25–35% of the year (winter). The remainder of the year is too hot, either inducing heat stress or lethally affecting outdoor production. Jeddah-area producers also face the dual challenges of accessing freshwater for irrigation and poor quality soils. Jeddah is not the only region with these problems: intense heat, lack of long-term access to freshwater for irrigation, and poor soil quality are common throughout the KSA. As an unsustainable practice, agriculture within the KSA currently consumes as much as 70–80% of available freshwater [1]. Unfortunately for the Kingdom, most of this freshwater currently used for agriculture is extracted from non-recharging fossil aquifer systems.

One of the major factors affecting greenhouse production is the ability to keep temperatures within the optimal range, generally 20–25˚C. In hot climates, this means extensive cooling. The most common method of cooling of greenhouses worldwide is evaporative cooling [2]. In evaporative cooling, the sensible heat of ambient air (temperature) is exchanged for latent heat (humidity) such that air entering a greenhouse is cooler and more humid than outside air. It is estimated that the energy use of an evaporative cooling system is four times less when compared with mechanical air conditioning processes (www.energy.gov). However, traditional evaporative cooling consumes a large amount of freshwater, constituting up to 80% of total greenhouse freshwater consumption [3,4]. Naturally occurring freshwater resources are scarce in the KSA desert; most comes from non-recharging aquifers. In spite of this, pad-and-fan evaporative cooling is widely used in the central dry (non-humid) areas of the KSA in both plant and animal production facilities [5]. The productivity of such greenhouses is generally low when compared with technologically sophisticated European greenhouses, and their use of freshwater for cooling is not sustainable.

Energy-intensive methods for greenhouse cooling may provide short-term solutions in light of the KSA's considerable fossil fuel reserves. Such methods may include traditional refrigeration-based cooling (air conditioning) or the use of desalinated water for evaporative cooling. Both solutions are energy-intensive and use energy from the burning of fossil fuels. The continued intensive use of fossil fuel resources is not a long-term solution, and other energy resources are not sufficiently developed at the present time to replace fossil fuels in the KSA context [6].

An additional challenge facing greenhouses on coastal deserts is that the high humidity levels limit the use and effectiveness of evaporative cooling. Ambient air in such areas is already near saturation with water vapor, leaving little capacity to exchange sensible heat for latent heat in the evaporative cooling process. Greenhouse cooling in hot and humid climates is a significant challenge, both in terms of economics and engineering [7]. This research aims to advance the science and knowledge of systems designed for greenhouse cooling in hot and humid climates.

Efforts to build greenhouses using alternative water sources and renewable energy for irrigation and cooling have demonstrated the feasibility of the approach, but have left substantial room for improvement. The "Seawater Greenhouse for Arid Lands" project, constructed in Tenerife in 1993, used evaporative cooling pads perfused with seawater to provide a cooling effect that was satisfactory in the local climate (www.seawatergreenhouse.com [8]). The same method of seawater-evaporative cooling was subsequently applied in the UAE and in Oman through collaborations between Seawater Greenhouse Ltd and Sultan Qaboos University [9]. Seawater Greenhouse Ltd implemented a further project in Australia, which remains in operation under the name of Sundrop Farms (www.sundropfarms.com), also incorporating a solar-PV fan system to reduce fossil fuel consumption

[10]. More recently, the Sahara Forest Project implemented a greenhouse in Qatar that uses seawater evaporative cooling (http://saharaforestproject.com). However, existing designs using seawater for evaporative cooling processes still face the fundamental limitation of evaporative cooling becoming ineffective when outdoor ambient air humidity is already near saturation. In humid regions similar to the coast near Jeddah, this means that average internal greenhouse conditions will often exceed 30°C during the hot and very humid months (August and September), which reduces product quality and prevents year-round cultivation of certain types of produce.

1.2. Liquid desiccant dehumidification

A critical step towards enabling the adoption of seawater-based evaporative cooling for greenhouses, which is both energy and freshwater efficient as compared with other forms of greenhouse cooling, is to make it effective for use in humid climates. One potential solution to reduce the amount of humidity present in ambient air is the use of chemical desiccants to dehumidify the air before it enters the evaporative cooling system. The absorption of water vapor can be accomplished using liquid or solid desiccants directly or indirectly through a membrane contactor [11]. Liquid desiccants are increasingly being used for dehumidification because of their operational flexibility [12]. Suitable liquid desiccants include highly concentrated salt solutions like magnesium chloride, calcium chloride, or lithium chloride. The driving force behind the effectiveness of a liquid desiccant is its vapor pressure. A cool desiccant solution has a lower vapor pressure than the ambient vapor. Under these conditions, moisture is transferred from the air to the desiccant solution. However, when the moisture transfer takes place, latent heat is exchanged for sensible heat as the heat of condensation is released into the liquid and/or the air. The capacity of a liquid desiccant to remove humidity from the air is limited by both its concentration and its temperature: a concentrated cool desiccant is a good dehumidifying solution [13].

Therefore, to improve the performance of any liquid desiccant system, attention must be given to create sufficient contact between the ambient air and the liquid desiccant to induce desiccation, removing the heat of condensation from the desiccator using a heat rejecter or exchanger, and keeping the concentration of the liquid desiccant sufficiently high using a mass regenerator so that the vapor pressure is maintained below that of the ambient air. Freshwater can also potentially be recovered for reuse within the greenhouse for evaporative cooling or irrigation by the liquid desiccant system as shown in Fig. 1. Use of these coupled processes has the potential of markedly reducing the freshwater footprint of greenhouse agriculture within a greenhouse to near zero, resulting in a "self-watering" greenhouse.

1.3. Freshwater recovery from liquid desiccant regeneration

To achieve continuous dehumidification for cooling of greenhouse air, the liquid desiccant must be regenerated by the removal of freshwater mass from the desiccant on a regular basis. To move closer to the goal of creating a greenhouse with a near-zero freshwater footprint, it is absolutely critical that an appropriate regenerator be developed and optimized to recover the freshwater that is extracted from the liquid desiccant.

2. Theory of self-watering greenhouse using liquid desiccant cooling

The principle of the proposed self-watering greenhouse is shown in Fig. 2. The illustration is of a closed greenhouse in which air is recirculated and cooled continuously by an evaporative pad wetted with seawater, brackish water, or recovered freshwater. For effective cooling to be maintained, moisture must be removed from the air: otherwise the air will become saturated and the cooling will no longer be effective. Removal of moisture is the function of the liquid desiccant that comes into contact with the air upstream of the evaporator. As a result, the liquid desiccant becomes diluted slightly, and so it has to be regenerated to the initial concentration. In the process of regeneration, the moisture that is absorbed by the

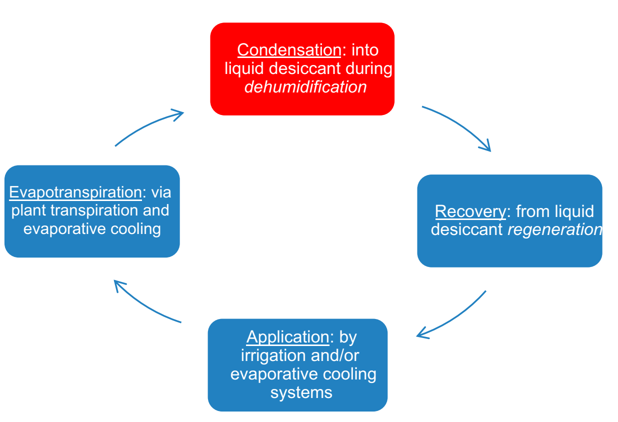

Fig. 1. Process diagram showing freshwater cycle within a proposed self-watering greenhouse.

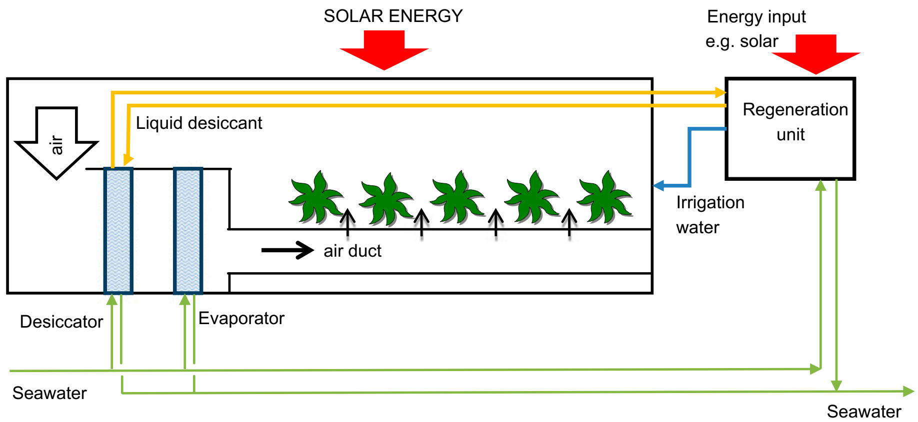

Fig. 2. Schematic of the self-watering greenhouse.

desiccant is separated and returned to the greenhouse for irrigation. This makes the system self-sufficient in freshwater. Whatever water evaporates inside the greenhouse—either from the plants or from the evaporator—is returned to the greenhouse. Thus, no external source of freshwater is required.

Note that removal of moisture by the desiccant results in release of heat, as water vapor is condensed to liquid state. This latent heat is taken away by a heat exchanger embedded in the desiccator. Seawater or other cool, brackish water circulates through the heat exchanger as the cooling medium.

As a whole, the system can be viewed as a refrigerator, which removes heat from a greenhouse and pumps it to the sea, the ground, or the ambient environment. From the general standpoint of the second law of thermodynamics, this process must require an energy input. Specifically, energy is needed for regeneration. In this case we prefer to use solar energy as the input, but other sources are possible, like electricity from the grid. The usual criterion of performance of a refrigeration system is the coefficient of performance (COP). It is the heat removed from the cooled space divided by the work supplied to the system, i.e.:

$$\mathrm{COP} = \frac{Q_{\mathrm{out}}}{W_{\mathrm{in}}} \tag{1}$$

The COP can be calculated for this system using certain simple and reasonable assumptions as follows:

Assumption 1: The heat Q_{out} removed from the greenhouse corresponds to the latent heat of the water vapor absorbed by the desiccant. In reality, the heat removed will be slightly greater because there is also heat associated with the dilution of the desiccant. In addition, there could be a contribution to heat removal if the liquid desiccant enters and leaves at different temperatures. However, both of these contributions are in fact small compared to the large contribution from the latent heat of water vapor:

$$Q_{\mathrm{out}} = m\, h_{\mathrm{fg}} \tag{2}$$

where h_{fg} is the specific enthalpy of evaporation and m is the mass of water evaporated. For water vapor in saturated air at 25˚C, h_{fg} = 2,440 kJ/kg.

Assumption 2: The work input W_{in} of regeneration corresponds to the minimum thermodynamic work of separating the water from the liquid desiccant—in other words the osmotic pressure. This assumption is much less realistic, but it is used here to indicate the ideal performance that can be obtained:

$$W_{\mathrm{in}} = P_{\mathrm{osm}} V = P_{\mathrm{osm}} m/\rho \tag{3}$$

where V is the volume of water removed from the desiccant and ρ is the density of pure water (= 1,000 kg/m^3). An interesting point is that the osmotic pressure is not independent of the desiccant properties of the liquid. On the contrary, it is closely related because both vapor pressure and osmotic pressure are colligative properties. This relationship allows a simple expression to be derived for the ideal COP.

The vapor pressure of a solution can be expressed as a fraction of that of the pure solvent. This fraction is called activity a, or equilibrium relative humidity (ERH %). In this case, the relation needed is [14]:

$$-\ln a = V' P_{\mathrm{osm}} RT \tag{4}$$

where R is the universal gas constant (8.3 kJ/kmol K), T is the absolute temperature (taken here as 298˚K) and V' is the specific molar volume of the solvent. For most aqueous solutions, $V' = 0.018\ m^3/kmol$. Combining the above equations provides the approximate expression:

$$COP = \frac{22}{-\ln a} \quad (5)$$

It is straightforward to estimate the needed value of a because a equals the limiting minimum relative humidity (equilibrium relative humidity) to which the air can be dried in contact with the liquid desiccant. For an evaporator to cool to the desired temperature, the ERH should be low enough to provide a desirable wet bulb temperature as an evaporator cannot cool below the wet bulb temperature of the air. Bearing in mind that the desiccator will not be perfectly effective in drying the air, a lower value of ERH for the desiccant is preferred (e.g. if an air of 50% relative humidity is desired, an appropriate desiccant ERH is 35–40%). The exact choice will depend on the design and sizing of the desiccator, the temperature of the liquid used for cooling, and the target temperature inside the greenhouse.

Fig. 3 shows how the ideal COP varies with ERH based on Eq. (5). With an ERH of 35% a COP of about 20 is achievable and at ERH 20% we get ideal COP = 14. These values of COP are very promising considering that conventional refrigeration equipment typically gives COP in the range of only 3–6.

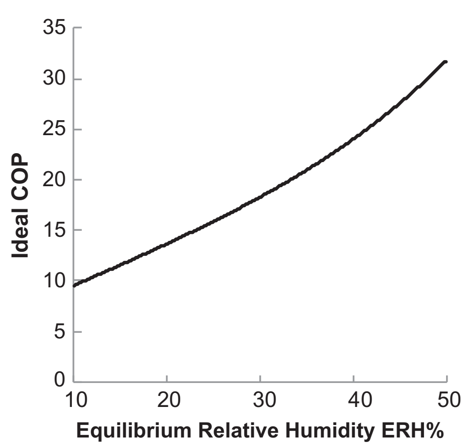

Fig. 3. The ideal achievable COP increases with the ERH of the desiccant solution. This is because a liquid desiccant solution that is better at drying and cooling the air will require more energy input for its regeneration. This relation applies to a range of liquid desiccants, rather than any one in particular, because it is based on general thermodynamic relations.

It is also interesting that Eq. (5) is valid for any aqueous desiccant solution in principle or in fact for any liquid desiccant with V' adjusted accordingly. As long as the correct value of a is used in Eq. (5), this equation is valid regardless of the composition of the liquid desiccant solution.

The real performance of the system depends crucially on the method of regeneration. Possible methods include, but are not limited to:

(1) Open regenerators. These are simple, but inefficient [15].
(2) Membrane distillation (MD). This includes direct contact, air-gap, vacuum gap and multi-stage systems [16,17].
(3) Reverse osmosis or nanofiltration. This is potentially very efficient, but the operating pressures may be prohibitively high [18].
(4) Electrodialysis [19,20].

3. Calculations of greenhouse freshwater needs

3.1. Evapotranspiration of tomato crop

Using the Priestley-Taylor Method to estimate evapotranspiration (ET) and assumptions as described by Valdes-Gomez et al. [21], a peak day in mid-summer was chosen to estimate greenhouse irrigation needs. The required inputs for the calculation come from solar radiation data collected in Thuwal, Saudi Arabia for one day from 15 August 2014 at 19:30 through 16 August 2014 at 19:30 and from assumed indoor conditions necessary for tomato production (Table 1).

Applying these inputs, the estimated ET of a tomato crop just after transplant (crop coefficient (Kc) = 0.4) [22] varies from 2.0 to 2.6 mm and at the start of harvest (Kc = 1.25), from 6.3 to 8.3 mm. This is more than the maximum measured ET of 5 mm in the cited Chilean study [21], probably due to variations in solar radiation and temperature. It should be noted that some water will also be "lost" as crops are harvested and plant material is removed from the greenhouse. The total amount of this water loss is small in comparison to ET, but will require replacement.

3.2. Water use by evaporative cooler

To estimate water use by the evaporative cooler, the limiting conditions within the greenhouse must first be set. The greenhouse under evaluation in this study is a single-pass or recirculating plug-flow type of greenhouse 40 m long by 9 m wide by 4 m tall. Cooled air enters at one end and exits or is

Table 1
Input conditions and expected evapotranspiration at multiple indoor climate goals

Variable	Indoor climate #1	Indoor climate #2	Indoor climate #3	Description
τ (Tau)	0.68	0.68	0.68	Coefficient of solar radiation transmission of covering material
Solar radiation (Rge) (MJ/m^2/d)	22.3	22.3	22.3	Solar radiation measured outside of greenhouse
Indoor average temperature (°C)	29.5	28	25	Average temperature goal in greenhouse
Indoor average relative humidity (%)	78	75	69	Average relative humidity goal in greenhouse
Indoor vapor pressure (kPa)	3.20	2.83	2.19	Average vapor pressure goal in greenhouse
ET after transplant (mm/m^2/d)	2.6	2.4	2.0	Average ET after transplant of tomatoes (min)
ET at start of harvest (mm/m^2/d)	8.3	7.6	6.3	Average ET at start of tomato harvest (max)
Total minimum greenhouse ET, L/d	951	872	720	Assuming 9 × 40 m greenhouse
Total maximum greenhouse ET, L/d	2,970	2,726	2,251	Assuming 9 × 40 m greenhouse

recirculated to the cooling system at another, with 40 m between entry and exit. The maximum temperature will be realized as solar radiation heats the indoor environment and will reach its highest point just before exit at the far side of the greenhouse. For calculations, the average daily maximum greenhouse temperature was set at 30°C and the instantaneous maximum temperature was set at 33°C [23]. Fig. 4 shows the variation in solar radiation and outdoor temperature over the course of the design day from before sunrise to after sunset.

To calculate heat gain, the solar radiation accrued inside the greenhouse was found by multiplying the outdoor solar radiation by the coefficient of solar radiation transmission (τ) of the covering material, in this case estimated at 0.68 for polycarbonate [24]. The solar radiation (J/m^2 s) was converted to energy added to the greenhouse (J/s) by multiplying the radiation by the floor area of the greenhouse (360 m^2) (Table 2).

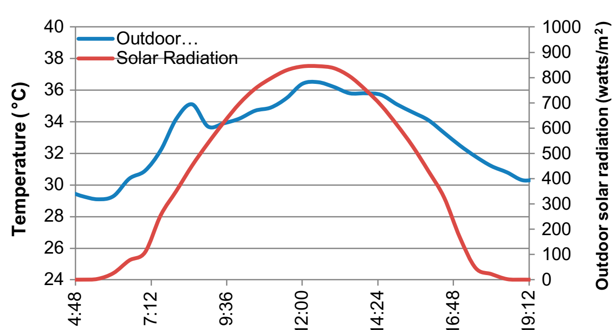

Fig. 4. Outdoor temperature and solar radiation from before sunrise to after sunset on 16 August 2014 in Thuwal, Saudi Arabia.

Energy added to the greenhouse through the walls was not considered, only solar radiation was considered in the calculations of energy flux into the greenhouse between the inlet from the cooling system and the outlet. The amount of energy transferred through the walls is expected to be small compared with the amount of energy added via solar radiation; therefore, it was ignored.

The energy added to the greenhouse is then converted into temperature gain. First, the total energy of air in the greenhouse at the outlet conditions is found at desired conditions of temperature and relative humidity. Then, the total maximum allowable energy at the inlet of the greenhouse can be back calculated by subtracting the energy added through solar gain. Next, a desired relative humidity at the inlet of the greenhouse is chosen. The inlet dry bulb temperature can be calculated using the psychrometric chart and the inlet enthalpy. The absolute humidity of the air entering can be estimated using the inlet dry bulb temperature and relative humidity. The expected transpiration from the tomato crop can then be added to estimate the outlet absolute humidity, which can be converted to relative humidity. After iterating a few times, an estimated inlet temperature can be found to satisfy the energy and mass balances and to provide desired values at the outlet. After these iterations are complete, simply subtracting the calculated inlet temperature from the outlet temperature allows for an estimation of the temperature gain within the greenhouse from inlet to outlet (Table 3).

Using the values obtained for desired temperature and humidity of air from the evaporative cooler to the greenhouse, it is easy to calculate both the air conditions desired for input into the evaporative cooler and

Table 2
Dry bulb temperatures and solar radiation input into calculations

Parameter	Daily peak	Average	Daily low	Comments
Dry bulb temperature in greenhouse (°C)	33	30	25	Goal, input into calculations
Dry bulb temperature outside of greenhouse (°C)	36.5	32.0	29.1	As measured: 16 August 2014, Thuwal, Saudi Arabia
Solar radiation outside (W/m^2)	851	253	0	As measured: 16 August 2014, Thuwal, Saudi Arabia
Solar radiation inside (W/m^2)	579	172	0	Calculated using coefficient of solar radiation transmission (τ) = 0.68 [24]

Table 3
Energy addition and temperature increase in greenhouses based on 360 m^2 floor area

Parameter	Daily peak	Average	Daily low	Comments
Air exchanges per hour	60	30	10	Total volume of air passing through greenhouse each hour divided by greenhouse volume
Energy added by solar radiation per exchange (kJ/cycle)	12,500	7,400	0	Energy added per volume of greenhouse air evacuated
Relative humidity of air from cooling system into greenhouse (%)	85	85	85	Goal, input into calculations
Dry bulb temperature of air from cooling system into greenhouse (°C)	26	26	25	Selected based on solar energy gain (below) and maximum desired conditions (Table 1)
Estimated temperature gain in greenhouse (°C)	7.0	4.0	0	Maximum minus minimum

the amount of water evaporated by the evaporative cooler. The temperature and relative humidity of air into the evaporative cooler are calculated using the expected cooling efficiency (η) [5]. Using psychrometric equations to calculate the absolute humidity before and after the evaporative cooler, the difference in absolute humidity is calculated by subtraction. Multiplying the difference in absolute humidity (kg/kg) by the expected airflow (kg/s) through the evaporative cooler provides an estimate of the amount of water evaporated per unit of time (Table 4). In this case (16 August 2014 in Thuwal, Saudi Arabia), the expected daily water evaporated by the evaporative cooler is estimated at ~3,100 L (assuming preconditioning by a liquid desiccant unit, discussed in the next section).

If freshwater is used as the evaporative cooling liquid, it can be added with the estimated tomato crop ET rates to get the total freshwater use rates for the described greenhouse: 3,888 L/d for a recently transplanted tomato crop to 5,580 L/d at the start of harvest. If seawater or another brackish water source is used for evaporative cooling, then only the irrigation system and plant material removed from the greenhouse (harvested produce) consume freshwater on a daily basis.

4. Calculated desiccator needs and proposed design

4.1. Desiccator design calculations

To meet the needs of humidity and temperature removal such that the inlet conditions to the evaporative cooler are suitable, a desiccator is proposed to remove humidity and adjust temperature. The requisite performance of the desiccator can be calculated using the calculated temperature and relative humidity input into the evaporative cooler, along with the ambient temperature and relative humidity.

The ambient outdoor conditions are available for the experimental day from weather station data. Using the ambient dry bulb temperature and relative humidity, the absolute humidity in kg of water per kg of air can be obtained from psychrometric relations. From the evaporative cooler calculations, desired outlet dry bulb temperature and relative humidity from the desiccator unit are also known and can be used to obtain

Table 4
Calculation inputs and results to estimate water use by evaporative cooler

Parameter	Daily peak	Average	Daily low	Comments
Expected efficiency of evaporative cooling system, (η) (%)	75	75	75	10 cm pad, 45° angles [5]
Max dry bulb temperature into evaporative cooler (°C)	31.9	31.9	30.8	Calculated from pad cooling efficiency
Max relative humidity into evaporative cooler (%)	52	52	52	Calculated from pad cooling efficiency
Calculated water use of evaporative cooler (kg/h)	257	129	42	Calculated based on pad cooling efficiency and airflow rate

the absolute humidity. By simply subtracting the absolute humidity after the desiccator from the absolute humidity before the desiccator, the required humidity removal can be found in kg water/kg dry air. Multiplying this value by the flow of air required (already calculated for the evaporative cooler) allows estimation of the humidity removal required per unit of time desired.

The estimated humidity removal efficiency of the desiccator on the basis of absolute humidity can be calculated from the following equation:

$$\eta_d = \frac{W_i - W_o}{W_i} \tag{6}$$

where η_d is equal to the humidity removal efficiency of the desiccator, W_i represents the absolute humidity at the inlet of the desiccator and W_o represents the absolute humidity at the outlet of the desiccator.

Finally, the required energy removal in kJ/h by the desiccator can be estimated by subtracting the enthalpy of the air after the desiccator from the enthalpy of the air before the desiccator and multiplying this value by the total air flow per unit time. Estimated values for a desiccator to meet the needs of the 16 August design day are shown in Table 5. These values assume that 100% of the air input into the desiccator comes from the outdoors.

As a lower energy alternative, air may be recycled from within the greenhouse to provide a lower energy input into the desiccator. Table 6 summarizes required desiccator performance if 100% of air input into the desiccator is recycled from inside the greenhouse.

4.2. Direct contact desiccator

A desiccator has been designed to meet the dual needs of humidity and energy removal, as shown in Fig. 5. The shown desiccator integrates cellulose pads with embedded heat exchange pipes. It is based on the design described by Lychnos and Davies [15,25]. The cellulose pad provides the surface area for air to desiccant contact and humidity removal. The embedded heat exchange pipes are included to remove energy from the system. Liquid desiccant is distributed over the cellulose pads from the top and flows via gravity to the bottom, where it is collected and (based on concentration) is either pumped to a regeneration system (to remove condensed water) or recycled back to the top of the desiccator (to absorb more humidity). A cooling liquid (brackish or sea water from the sea, the ground, or a cooling tower) circulates through the heat exchange pipes from the top left to the bottom left, to the bottom right, and then finally out through the top right of the proposed system. As the cooling liquid moves through the system it acts as a heat sink, collecting energy from the desiccator system and transporting it out of the system.

4.3. Hollow fiber membrane desiccator

The second desiccator system proposed for testing integrates hollow fiber membranes to separate the liquid desiccant from direct contact with the air. Hollow fiber membranes with a liquid desiccant solution have been used by other researchers to effectively dehumidify lab-scale experiments [11]. An advantage of this system is that it prevents any possibility for aerosols to develop from the desiccant salts and enter the greenhouse. It also prevents any airborne dust from entering the desiccant stream. A disadvantage of this system is that an additional mass transfer barrier exists between the air and the desiccant. Testing of a hydrophobic polyvinylidene fluoride (PVDF)-based triple-bore hollow fiber membrane with a circulating calcium chloride desiccant solution is now underway in the KAUST Water Reuse and Desalination Center.

Table 5
Desiccator design values using 100% outdoor air as input

Parameter	Daily peak	Average	Daily low	Comments
Dry bulb temperature into evaporative cooler (˚C)	32.5	32.5	30.8	Calculated from pad cooling efficiency
Relative humidity into evaporative cooler (%)	52	52	52	Calculated from pad cooling efficiency
Outdoor dry bulb temperature (˚C)	36.5	32	29.1	As measured: 16 August 2014, Thuwal, Saudi Arabia
Outdoor relative humidity (%)	60	73	84	As measured: 16 August 2014, Thuwal, Saudi Arabia
Required humidity removal by desiccator (kg/h)	839	354	126	Based on ambient and input conditions into evaporator
Required humidity removal efficiency of desiccator (%)	35	31	34	% of absolute humidity (kg water/kg dry air) removed
Required energy removal by desiccator (kJ/h)	256×10^4	870×10^3	278×10^3	Based on temperature and heat of condensation removal

Table 6
Desiccator design values using 100% recycled air from greenhouse outlet as input

Parameter	Daily peak	Average	Daily low	Comments
Dry bulb temperature into evaporative cooler (˚C)	32.5	32.5	30.8	Calculated from pad cooling efficiency
Relative humidity into evaporative cooler (%)	52	52	52	Calculated from pad cooling efficiency
Maximum dry bulb temperature in greenhouse (˚C)	33	30	25	Goal, input into calculations
Expected relative humidity at greenhouse outlet (%)	57	67	85	Calculated based on other inputs
Required humidity removal by desiccator (kg/h)	295	147	48	Based on ambient and input conditions into evaporator
Required humidity removal efficiency of desiccator (%)	16	16	16	% of absolute humidity (kg water/kg dry air) removed
Required energy removal by desiccator (kJ/h)	790×10^3	234×10^3	6,220	Based on temperature and heat of condensation removal

5. Liquid desiccant regenerator performance and sizing

A regeneration system is required to remove condensed humidity from the liquid desiccant to enable continuous operation and to maintain a constant desiccant ERH. Peak system regeneration needs were 839 kg/h on the design day with 100% outdoor air input and 295 kg/h with 100% recycled air input (Tables 5 and 6).

5.1. *Hollow fiber vacuum MD laboratory results*

A hydrophobic PVDF hollow fiber membrane was manufactured in the lab using a 12 wt.% polymer solution. After manufacture, the hollow fibers were investigated using a Quanta 600 FEG scanning electron microscope (Fig. 6).

Following characterization of the hollow fibers, a setup was created in the lab to test the regeneration performance of the fibers under vacuum. Calcium chloride desiccant solution was pumped through the lumen of the hollow fiber by a peristaltic pump at a rate of 20 mL/min. Vacuum was applied at the outer surface of the hollow fiber to retrieve the vapor passed through the membrane wall. A pressure meter was installed between the peristaltic pump and the hollow fiber. A temperature and conductivity meter was installed after the hollow fiber to measure solution conditions after the MD process. The solution was

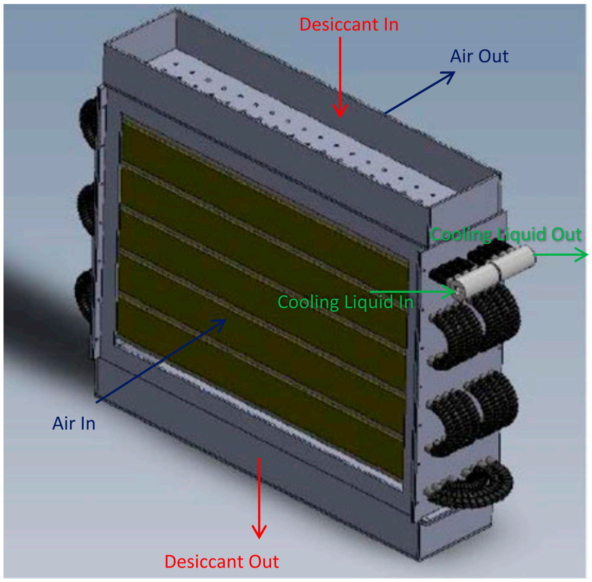

Fig. 5. Integrated cellulose pad desiccator with heat removal pipes built in. The shown desiccator is currently under construction in the KAUST workshop and will be tested for performance during the coming year.

recirculated into the primary container where it mixed with the bulk, lower concentration desiccant solution. The primary container consisted of a jacketed glass tube with connections for a circulating heating or cooling liquid. Water heated to 50˚C was circulated in the outer tube to warm the bulk desiccant. The total amount of water removed from the desiccant solution was measured by passing the vacuum line through a condenser trap cooled by liquid nitrogen. Therefore, all water vapor in the vacuum line condensed and froze inside the trap during the experiment, allowing the final weight of the recovered water to be measured post-experiment. The conductivity of permeate collected was also measured post-experiment to evaluate its potential use as irrigation water. Finally, the flux was calculated.

Preliminary test results, including the amount of permeate (fresh water) collected from the laboratory tests, are shown in the following table (Table 7). The final concentration by weight of the $CaCl_2$ desiccant solution was used to estimate the equilibrium relative humidity of the solution [26].

Assuming that performance of the hollow fiber membrane regenerative system is maintained upon scale-up to field size, a regenerator can be sized to meet the peak demand and the average hourly regeneration demands. Meeting the average hourly regeneration demand for a desiccator dehumidifying outdoor air at input conditions would produce ~8,496 L of freshwater for use within the greenhouse over the course of a day. If the indoor air is recycled through the desiccator, the daily water removal required from the desiccant drops to ~3,528 L/d. As the required freshwater for irrigation ranged from 2,250 to 2,970 L/d (Table 1), a properly functioning liquid desiccant dehumidification system with a regenerator system outputting freshwater would meet crop production needs.

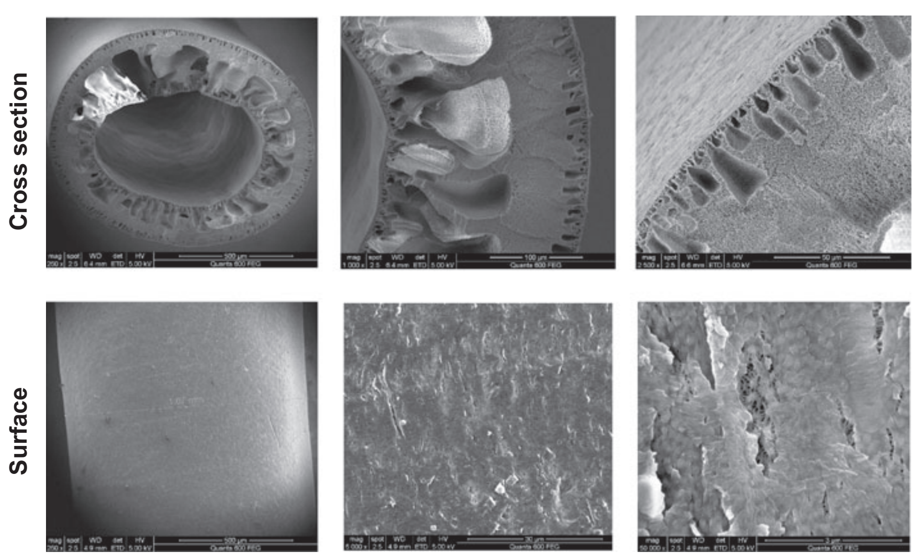

Fig. 6. Scanning electron microscope images of PVDF hollow fiber membrane.

Table 7
Results of hollow fiber vacuum MD experiment

Parameter measured	Test #1	Test #2	Test #3
Fiber inner diameter (mm)	0.62	0.62	0.62
Fiber length (mm)	410	395	395
Active surface area (m^2)	0.00080	0.00077	0.00077
Starting concentration $CaCl_2$ (wt.%)	35.0	25.0	30.0
Ending concentration $CaCl_2$ (wt.%)	41.2	29.2	32.4
Temperature of $CaCl_2$ solution (˚C)	30	27	29
Pressure of $CaCl_2$ solution (bar)	0.80	0.57	0.62
Vacuum pressure (millibar)	~3	2	8
Weight of permeate collected (g)	46.1	70.5	73.0
Conductivity of permeate collected (μs/cm)	26	Not measured	3.2
Flux (g/(m^2 h))	2,015	5,714	4,789
Ending equilibrium relative humidity (%)	37.3	63.1	56.3

6. Discussion

This work has highlighted the potential to realize a self-watering greenhouse system based on liquid desiccation and regeneration of the desiccant by solar thermal energy in a MD system. Effective and economical regeneration is the key challenge in realizing such a system. Advances in MD technology such as those based on PVDF membranes, demonstrated in this work, show great promise in this respect. The hollow fiber reported here was able to withstand the high concentrations of desiccant solution needed to lower the humidity in the greenhouse sufficiently. Nonetheless, there remain several challenges in implementing these advances in a full-scale greenhouse. In a large MD system, localized concentration in the fiber bundles could present a risk of crystallization and blockage, if the system is not designed carefully. In addition, the thermal input requirement to the MD system may be excessive unless the system is configured (by multiple stages or regenerative arrangement) to provide a gain output ratio (GOR) substantially greater than one. Because of the high boiling point of the liquid desiccant solution, substantial driving temperature gradients may be needed to achieve GOR > 1, presenting challenges for the membrane materials as feed temperatures are increased. The on-going program of work will address these challenges through construction of pilot systems at progressively larger scale, connected to solar thermal collectors so that the engineering issues can be identified and resolved. Noting that the thermodynamic analysis allows very high COP values in principle, it is anticipated that—notwithstanding these challenges—an attractive COP allowing a compact solar collector arrangement will be achieved in practice. Thus, the solar thermal collector of compact size compared to the greenhouse footprint would allow for integration with the same structure or an adjoining structure, without adding excessively to capital cost. As the system is driven by solar energy, running costs will be minimal.

7. Conclusions

Calculations have been done to estimate the mass and energy balance within a 360 m^2 greenhouse cooled by a combined liquid desiccant and evaporative cooling system during a design summer day (16 August 2014) in Thuwal, Saudi Arabia. Based on the literature values, theoretical performance, and achieved preliminary lab results for the various components of such a system, we conclude that the freshwater needs can be met. We draw the following specific conclusions:

(1) A liquid desiccant air dehumidifier followed by evaporative cooling provides a potential solution to meeting both cooling and freshwater supply requirements in desert greenhouses.
(2) A COP of 10–30 is theoretically achievable and attractive compared with the efficiency of mechanical refrigeration technology.
(3) Peak crop irrigation needs for tomatoes grown in a 9 × 40 m greenhouse have been estimated at ~2,200–3,000 L/d based on a hot summer day (16 August 2014 in Thuwal, Saudi Arabia).
(4) Recycling air from the greenhouse to the cooling system lowers the energy and humidity removal required from a desiccator when compared with cooling of outdoor air for the design day.

(5) Average flux achieved by a hollow fiber vacuum MD system varied from 2.0 to 5.7 L/m^2 h related to the input concentration of the desiccant.
(6) The MD system was able to produce output desiccant concentrations with an equilibrium relative humidity of ~38%, near to the theoretical recommended value of 35–40%.
(7) The recovered freshwater from the MD system was of suitable quality to be used as irrigation water.
(8) A properly functioning desiccant regenerator can theoretically meet both crop irrigation and desiccant regeneration needs.

Acknowledgments

The research reported in this publication was supported by funding from King Abdullah University of Science and Technology (KAUST). The authors wish to acknowledge and thank Shinthujan Selvanayagam of Aston University, who contributed design drawings for the direct contact desiccator, and Rami Alsouqi for his input related to commercial greenhouse systems.

Symbols

COP	—	coefficient of performance of cooling system
Q_{out}	—	heat removed by cooling system
W_{in}	—	work input into cooling system
m	—	mass of water evaporated
h_{fg}	—	specific enthalpy of evaporation
P_{osm}	—	osmotic pressure
V	—	volume of water removed from desiccant
ρ	—	density of pure water, 1,000 kg/m^3
a	—	activity
ERH	—	equilibrium relative humidity
V'	—	specific molar volume of the solvent
R	—	universal gas constant (8.3 kJ/kmol K)
T	—	absolute temperature in Kelvin
τ	—	coefficient of solar radiation transmission of greenhouse covering material
Rge	—	solar radiation measured outside of greenhouse
Kc	—	crop coefficient for evapotranspiration equation
η	—	efficiency of evaporative cooling system
η_d	—	absolute humidity removal efficiency of the desiccator
W_i	—	absolute humidity of air into the desiccator
W_o	—	absolute humidity of air out of the desiccator

References

[1] M. Al-Saud, Water Sector of Saudi Arabia, in: Presentation at the 2nd Japan-Arab Economic Forum, 11 December 2010, Tunisia, Ministry of Water and Electricity, Water Resources Development Department, Saudi Arabia, 2010.
[2] C. Kittas, T. Bartzanas, A. Jaffrin, Greenhouse evaporative cooling: Measurement and data analysis, Trans. ASAE 44(3) (2001) 683–689.
[3] A.S. Kassem, Energy and water management in evaporative cooling systems in Saudi Arabia. Resour. Conserv. Recycl. 12(3–4) (1994) 135–146.
[4] N.C. Sabeh, Evaluating and Minimizing Water Use by Greenhouse Evaporative Cooling Systems in a Semi-Arid Climate, PhD thesis, The University of Arizona, Tucson, AZ, 2007.
[5] R.W. Koca, W.C. Hughes, L.L. Christianson, Evaporative cooling pads: Test procedure and evaluation, Appl. Eng. Agric. 7(4) (1991) 485–490.
[6] G. Lahn, P. Stevens, Burning Oil to Keep Cool, The Hidden Energy Crisis in Saudi Arabia, Chattam House, London, 2011.
[7] K.S. Kumar, K.N. Tiwari, M.K. Jha, Design and technology for greenhouse cooling in tropical and subtropical regions: A review, Energy Build. 41(12) (2009) 1269–1275.
[8] C. Paton, P. Davies, The Seawater Greenhouse for Arid Lands, Mediterranean Conf. on Renewable Energy Sources for Water Production, Santorini, Greece, 1996.
[9] P. Davies, C. Paton, The seawater greenhouse in the United Arab Emirates: Thermal modelling and evaluation of design options, Desalination 173 (2004) 103–111.
[10] P.A. Davies, A.K. Hossain, G. Lychnos, C. Paton, Energy saving and solar electricity in fan-ventilated greenhouses, Acta Hortic. 797 (2008) 95–101.
[11] K. Kneifel, S. Nowak, W. Albrecht, R. Hilke, R. Just, K.V. Peinemann, Hollow fiber membrane contactor for air humidity control: Modules and membranes, J. Membr. Sci. 276 (2006) 241–251.
[12] A. Lowenstein, Review of liquid desiccant technology for HVAC applications, HVAC&R Res. 14(6) (2008) 819–839.
[13] A.Y. Khan, Sensitivity analysis and component modelling of a packed-type liquid desiccant system at partial load operating conditions, Int. J. Energy Res. 18(7) (1994) 643–655.
[14] E.B. Smith, Basic Chemical Thermodynamics, fifth ed., Imperial College Press, London, 2004.
[15] G. Lychnos, P.A. Davies, Modelling and experimental verification of a solar-powered liquid desiccant cooling system for greenhouse food production in hot climates, Energy 40 (2012) 116–130.
[16] A.S. Rattner, A.K. Nagavarapu, S. Garimella, T.F. Fuller, Modeling of a flat plate membrane-distillation system for liquid desiccant regeneration in air-conditioning applications, Int. J. Heat Mass Transfer 54 (2011) 3650–3660.

[17] F.H. Choo, M. KumJa, K. Zhao, A. Chakraborty, E.T.M. Dass, M. Prabu, B. Li, S. Dubey, Experimental study on the performance of membrane based multi-effect dehumidifier regenerator powered by solar energy, in: International Conference Solar Heating and Cooling for Buildings and Industry, Freiburg, Germany, Elsevier, 2014.
[18] F.A. Al-Sulaiman, P. Gandhidasan, S.M. Zubair, Liquid desiccant based two-stage evaporative cooling system using reverse osmosis (RO) process for regeneration, Appl. Therm. Eng. 27 (2007) 2449–2454.
[19] Q. Cheng, X. Zhang, Review of solar regeneration methods for liquid desiccant air-conditioning system, Energy Build. 67 (2013) 426–433.
[20] Q. Cheng, X.S. Zhang, X.W. Li, Performance analysis of a new desiccant pre-treatment electrodialysis regeneration system for liquid desiccant, Energy Build. 66 (2013) 1–15.
[21] H. Valdes-Gomez, S. Ortega-Farias, M. Argote, Evaluation of water requirements for a greenhouse tomato crop using the Priestley-Taylor method, Chil. J. Ag. Res. 69(1) (2009) 3–11.
[22] N. Castilla, E. Fereres, The climate and water requirements of tomatoes in unheated plastic greenhouses, Ag. Med. 120 (1990) 268–274.
[23] R.G. Snyder, Greenhouse Tomato Handbook, Mississippi State University, Starkville, MS, 1992.
[24] M. Vuscovich, Desarrollo de Un Modelo Para Determinar Evapotranspiracion En Invernaderos, In Facultad de Ingenieria Agricola, Universidad de Concepcion, Chillan, Chile, 2001, pp. 64.
[25] G. Lychnos, Feasibility of a Solar-Powered Liquid Desiccant Cooling System for Greenhouses, PhD Thesis, Aston University, Birmingham, 2010.
[26] P.A. Davies, P.R. Knowles, Seawater bitterns as a source of liquid desiccant for use in solar-cooled greenhouses, Desalination 196 (2006) 266–279.