

A RADIANT AIR-CONDITIONING SYSTEM USING SOLAR-DRIVEN LIQUID DESICCANT EVAPORATIVE WATER COOLER

S. A. ABDALLA*, KAMAL N. ABDALLA

Department of Mechanical Engineering, Faculty of Engineering, University of Khartoum,
SUDAN.

*Corresponding Author: saabdalla@uofk.edu

Abstract

Every air-conditioning system needs some fresh air to provide adequate ventilation air required to remove moisture, gases like ammonia and hydrogen sulphide, disease organisms, and heat from occupied spaces. However, natural ventilation is difficult to control because urban areas outside air is often polluted and cannot be supplied to inner spaces before being filtered.

Besides the high electrical demand of refrigerant compression units used by most air-conditioning systems, and fans used to transport the cool air through the thermal distribution system draw a significant amount of electrical energy in comparison with electrical energy used by the building thermal conditioning systems. Part of this electricity heats the cooled air; thereby add to the internal thermal cooling peak load. In addition, refrigerant compression has both direct and indirect negative effects on the environment on both local and global scales.

In seeking for innovative air-conditioning systems that maintain and improve indoor air quality under potentially more demanding performance criteria without increasing environmental impact, this paper presents radiant air-conditioning system which uses a solar-driven liquid desiccant evaporative cooler.

The paper describes the proposed solar-driven liquid desiccant evaporative cooling system and the method used for investigating its performance in providing cold water for a radiant air-conditioning system in Khartoum (Central Sudan). The results of the investigation show that the system can operate in humid as well as dry climates and that employing such a system reduces air-conditioning peak electrical demands as compared to vapour compression systems.

Keywords: Liquid Desiccant Evaporative Cooling, Liquid Desiccant Dehumidifier, Evaporative Cooling, Solar Collector Heater.

Nomenclatures

A_c	Area of the collector plate [m ²]
a	Mass transfer area per unit volume [m ² /m ³]
C	Mass specific heat product ($\dot{m} C_p$) [kW/°C]
C_p	Specific heat at constant pressure [kJ/kg.°C]
F_R	Collector heat removal factor
h_a	Convection heat transfer coefficient [kW/m ² .°C]
h_m	Convection mass transfer coefficient [kg/sec. m ²]
I_g	Latent heat of vaporization of the water [kJ/kg]
i_w	Specific enthalpy of saturated liquid vapor [kJ/kg]
i_g	Specific enthalpy of saturated water vapour [kJ/kg]
L	Chamber total height [m]
m	Mass flow-rate [kg/sec]
q	Rate of heat transfer [kW]
T_a	Ambient temperature [°C]

1. Introduction

Air-conditioning has been achieved reliably and efficiently over the last few decades due to the popularity gained by vapour compression machines as a result of halogenated hydrocarbons discovery. The need to conserve high grade energy and reducing the harm effects of halogenated hydrocarbons, such as; the contribution to the Earth's ozone layer depletion and global warming due to emissions of halogenated hydrocarbons during production and use, necessitate exploring alternative techniques.

Evaporative cooling, a very simple, robust and low cost cooling technology basically achieved by evaporation of water in air is one proposition. Evaporative water coolers (cooling towers) are devices utilizing the direct contact between water and atmospheric air to cool water by evaporating part of the sprayed water in the air. Despite its potential to reduce cooling energy and peak energy demand, cooling towers are not widely used in many areas because of their declining cooling capacity with increasing outdoor humidity.

In liquid desiccant evaporative cooling (LDEC) process air is used, dehumidified by a desiccant solution, to cool water by direct evaporative cooling (both require no refrigerant). LDEC is considered to be a modification of direct evaporative cooling that can cater for different climates.

Unlike vapour compression cooling which rely on high energy technology, desiccant evaporative cooling relies on desiccant dehumidification (low energy technology) to provide dry air required for ventilation and evaporative cooling. Solar energy or any other type of energy that might otherwise be wasted provides the heat

energy required for regenerating the desiccant used by the desiccant dehumidifier during the cooling season (summer) and heating the water circulated through the radiant system during the heating season (winter). This provides dry ventilation air and cold water for a radiant system, and thereby gives a solution to thermal environment control that significantly reduces electrical energy demands, greenhouse gas emissions and dependence on harmful refrigerants.

As an open heat driven cycle affording the opportunity to utilize heat that might otherwise be wasted, a liquid desiccant evaporative cooling cycle can be coupled with solar heating to produce dry ventilation air and cold water for a radiant system. This can significantly reduce cooling electrical energy demands in comparison with conventional vapour compression refrigeration, and should in theory be extremely environment friendly as it eliminates greenhouse gas emissions and dependence on harmful refrigerants. As it delivers cold water and dry air at relatively high COP, solar-operated liquid desiccant evaporative water cooling would be cost effective.

The objective of this paper is to study the performance of a solar-driven desiccant evaporative cooling system in providing cold water for a radiant air-conditioning system in Khartoum Sudan. In doing so, a computer program was used to simulate the solar-driven liquid desiccant evaporative cooler. The computer program was developed based on unit subroutines constituting the solar-operated liquid desiccant evaporative cooling system components governing equations.

2. System Description

The liquid desiccant evaporative water cooler, which is designed to serve as an open cycle absorption system operating with solar energy is shown schematically in Fig. 1. The cooler consists of nine major components: continuous fin tube type process air pre-cooler, air-to-water air cooler, an isothermal vertical tube type falling film absorber, adiabatic packed-bed tower regenerator, solution-to-solution strong solution pre-cooler and weak solution pre-heater, water-to-solution solution cooler, solution-to-thermal fluid solution heater, solar collector thermal fluid heater, counter-flow packed bed type evaporative water cooler and appropriate instruments for various measurements. Arabic numerals indicate working fluids states at specific locations; thick solid lines represent air flow, thin solid and dashed lines represent solution and water flow respectively.

The liquid desiccant system is connected in a flow arrangement that allows thermal fluid storage and is capable to work in two automatic modes as may be selected by the user. One automatic mode is for full system operation in which all components including the thermal fluid storage circuit operate, while the second is for solar heating only.

In the full automatic mode, pump 1 pumps absorbent solution from regenerator sump (state 13) through the solution-to-solution heat exchanger where it is pre-cooled by exchanging heat with cold solution leaving the absorber sump. The solution then flows through the solution-to-water heat exchanger where it is cooled to state 9 by water from the evaporative water cooler and supplied to the absorber distribution system. The cold solution to trickle down in counter flow to air stream and collects in the absorber sump. A fan draws ambient air through the air-to-air heat exchanger where it is pre-cooled to state 2

and through the air-to-water heat exchanger where it is cooled to state 3 to the absorber chamber. In the absorber, water vapour is removed from the sensibly cooled process air entering the bottom of the absorber (state 3) by being absorbed into the absorbent solution. Part of the dehumidified air leaving the absorber (state 4) is taken to facilitate ventilation purposes while the remainder is brought into direct contact with sprayed water in the evaporative cooler. The temperature of the absorbent solution in the absorber is maintained constant using a water-to-solution heat exchanger enclosed within the absorber chamber through which cold water from the evaporative water cooler is circulated.

To maintain the liquid desiccant at the proper concentration for moisture removal, pump 2 pumps weak solution from the absorber sump (state 10), through the solution-to-solution heat exchanger where it is pre-heated to state 11 by recovering heat from the hot solution leaving the regenerator. The pre-heated solution is then pumped through the solution-to-thermal fluid heat exchanger where it is heated to the required regeneration temperature (state 12). The hot solution then trickles down the regenerator distribution system in counter flow to atmospheric air entering at the bottom of the regenerator. The vapour-pressure difference between the ambient air and the hot solution causes ambient air to absorb water vapour from the solution (i.e. re-concentrate the absorbent to state 13). The hot air is discharged to the atmosphere while the re-concentrated solution (state 13) is pumped through the solution-to-solution pre-cooler and the solution-to-water cooler to the absorber distribution system.

During solar heating, pump 4 supplies the thermal fluid-solution heat exchanger with the required amount of the hot thermal fluid from the hot fluid storage tank. After it exchanges its heat with weak solution, the leaving warm solution is mixed with another amount of warm thermal fluid from the warm fluid storage tank and pumped through the solar collector heater to the hot thermal fluid storage tank. During night, pump 5 supplies the thermal fluid-solution heat exchanger with the required amount of hot thermal fluid from the hot thermal fluid storage and store the warm fluid in the warm fluid storage tank.

The regenerator and the associated flow system and components are all similar to what was shown at the absorber side. The system regeneration side is shut down if the thermal fluid storage tank cannot supply thermal fluid at sufficiently high temperature or if the absorbent solution concentration in the absorber pool rises above a set limit.

Figure 2 shows the psychrometric cycle of process air flowing through the solar-driven liquid desiccant evaporative water cooler employed solely to provide cold water for a radiant system. Lines 1-2, 2-3 represent the path of the process air (ambient air) through the air-to-air and air-to-water heat exchangers. Line 3-4 represents the path through the absorber and line 4-5 the pass through the evaporative water cooler.

3. System's Simulation

The simulation process constitutes description of the procedure used to model the system components and a main program that integrates these components. The main program calls the unit subroutines to link the components and form a complete cycle.

Mass and energy governing equations are written by taking each system component as a control volume and divide the domain of interest into a finite number of computational cells using finite difference technique. A mathematical solver solves simultaneously the system components governing equations.

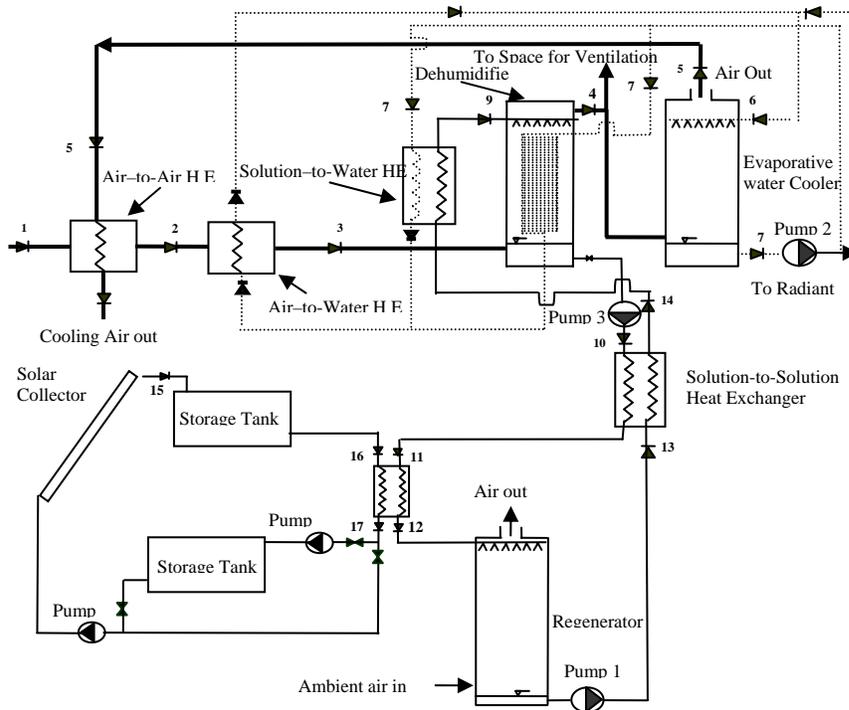


Fig. 1. Schematic of the solar-driven liquid desiccant evaporative water cooler design version.

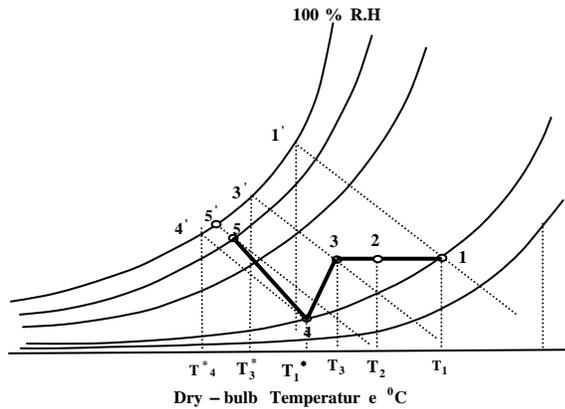


Fig. 2. Solar-driven desiccant evaporative cooling cycle.

A MatLab[®] computer code was developed specifically to simulate the system. The code is based on unit sub-programs containing the governing equations of the system's components. The components are linked together by a main program which calls the unit's sub-programs to form the complete cycle. A mathematical solver is employed to solve the entire cycle established equations. Property sub-programs were used to provide the thermodynamic properties of the different working fluids. An explanation of the main program aided by a detailed flow chart is shown in Fig. 8.

As detailed in the flow chart, the program begins with a definition of air-to-air heat exchanger constant parameters (ambient air conditions, process air mass flow rate, heat exchanger configuration and tube diameters). With the air-to-air heat exchanger constant parameters defined, the program determines the exchanger parameters and simulates the exchanger by calling the exchanger sub-routine. The same procedure applies for the air-to-water heat exchanger; the only difference is that the condition of air leaving the air-to-air heat exchanger replaces ambient air condition.

Next, the absorber constant parameters are defined. These include absorber cross sectional area, salt and air mass flow rates; concentration of salt-water solution entering the absorber, temperature by which salt-water solution and cooling water enters the absorber. Using these parameters the heat and mass transfer coefficients, and salt-water solution enthalpy are determined. Then a length for the absorber and is assumed and divided into smaller differential segments of equal lengths; this sets the total number of iterations to be performed by the absorber calculation loop. The absorber calculation loop is primarily based on two routines which are called within the absorber calculation loop. Routine 1 marching in the downward direction solves liquid differential segments, and routine 2 marches upward solves air and cooling water differential segments.

Next, the solution-to-solution heat exchanger constant parameters are defined (conditions and mass flow rate of solution leaving the absorber and the regenerator, heat exchanger configuration and tube diameters). With heat exchanger constant parameters defined, solution-to-solution heat exchanger sub-routine is called to simulate the exchanger. The same procedure applies for the thermal fluid-to-solution heat exchanger; the only difference is that the conditions of solution leaving the solution-to-solution heat exchanger and the storage tank leaving conditions are the heat exchanger inputs.

3.1. Heat exchangers models

The goal of the heat exchanger modelling is to relate the inlet and outlet temperatures, the overall heat transfer coefficient, and the geometry of the heat exchanger to the rate of heat transfer between the two fluids. This study employs the effectiveness number of transfer units ϵ -NTU approach for simulating the heat exchangers. The analysis begins by determining the overall heat transfer coefficient and then calculates the effectiveness ϵ as a function of the number of transfer units NTU and capacity rate

ratio Z. Cross and counter flow configurations effectiveness formulae can be given by equations 1 and 2 respectively [1-2]: -

$$\varepsilon = 1 - \exp \left[\left(\frac{1}{Z \eta} \exp [- NTU] Z \eta \right) - 1 \right] \tag{1}$$

$$\varepsilon = \frac{1 - \exp[- NTU(1 - Z)]}{1 - Z \exp[- NTU(1 - Z)]} \tag{2}$$

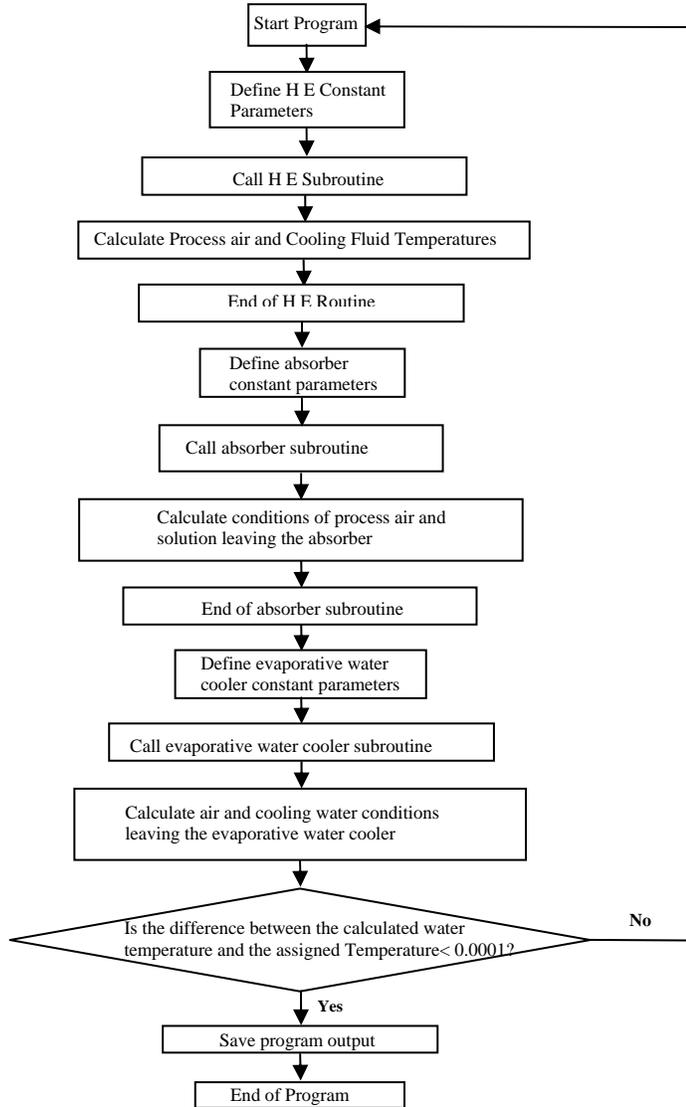


Fig. 3. Absorber simulation flow chart.

Once the effectiveness is known, temperature of the cold stream leaving the exchanger $T_{1, out}$, the heat transfer Q and the temperature of hot stream leaving the exchanger $T_{2, out}$ are also calculated as:-

$$T_{1, out} = T_{1, in} + \varepsilon \frac{\dot{C}_1}{\dot{C}_{min}} (T_{2, in} - T_{1, in}) \quad (3)$$

$$\dot{Q} = \dot{C}_1 (T_{1, out} - T_{1, in}) \quad (4)$$

$$T_{2, out} = T_{2, in} - \frac{\dot{Q}}{\dot{C}_2} \quad (5)$$

3.2. The absorber model

Many researchers [8- 13] dealt with desiccant systems and desiccant system modelling. The absorber modelling procedure employs finite difference method to derive the governing equations. In the analysis, it is assumed that only water is transferred between air and the desiccant; the chamber cross-section is assumed to be uniform, the solution temperature is assumed to remain constant, the liquid phase resistance to heat transfer is considered negligible (resulting in interfacial temperature equal to the desiccant bulk temperature). The principal resistance to heat and mass transfer is assumed to occur in the boundary layer of air surrounding solution droplet, and, heat conduction and mass diffusion in the solution and in the air normal to the flow direction are neglected.

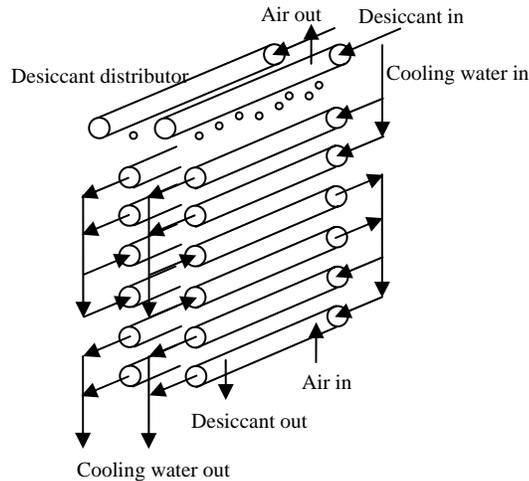


Fig. 4. Flow streams inside the absorber.

The governing equations are obtained by dividing the absorber height into small

segments. Each small segment is further divided into liquid desiccant, moist air, and cooling water differential segments. The model is primarily based on two routines; routine 1 marching in the downward direction solves the liquid desiccant differential segments and routine 2 marches upward to solve the moist air and cooling water differential segments. Due to the interlinking of the liquid desiccant, moist air and cooling water control volumes, moist air and cooling water conditions are initially required to solve the liquid region.

Steady state mass balance for the given absorber configuration as obtained from the typical control volume of Fig. 3 shows that the mass of salt in solution and mass of dry air in moist air are constant, and that mass transfer (mass of vapour added to the solution equals the mass of water vapour condensed from).

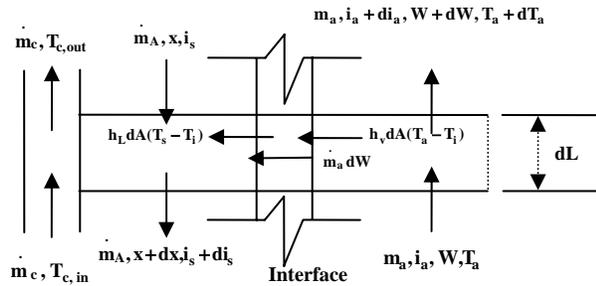


Fig. 5. Differential heat and mass transfer element used.

To derive absorber finite difference model equations the mass transfer is driven by the difference between air and solution partial vapour pressures determined by solution temperature and concentration. The rate of mass transfer across a differential length dL from a tower whose empty cross-sectional area A are given in terms of an assumed constant interfacial area per unit volume a . The differential interfacial area dA_i is often expressed in terms of a constant interfacial area per unit volume a , the absorber differential height dL and the cross sectional area of the empty absorber A as $dA_i = aAdL$. The mass flux is given by [3] as

$$\dot{m}_a dW = K_{m,v} dA_i (y_a - y_i) = K_{m,v} (y_a - y_i) aAdL \tag{6}$$

$$\dot{m}_A dx = K_{m,L} dA_i (x_i - x) = K_{m,L} (x_i - x) aAdL \tag{7}$$

Control volume steady state energy balance yields:

$$i_s \dot{m}_A + dq_c = \dot{m}_A (i_s + di_s) + h_s aA (T_i - T_s) dL + i_w \dot{m}_a dW \tag{8}$$

The liquid phase control volume energy balance depends on the location of the falling film on the cooling water tubes or between the tubes. For the control volume in the cooling water tubes region

$$i_s \dot{m}_A + dq_c = \dot{m}_A (i_s + di_s) + h_s aA (T_i - T_s) dL + i_w \dot{m}_a dW \quad (9)$$

The energy balance for the control volume between the tubes is given by

$$i_s \dot{m}_A = \dot{m}_A (i_s + di_s) + h_s aA (T_i - T_s) dL + i_w \dot{m}_a dW \quad (10)$$

An energy balance equation for the control volume of moist air gives

$$\dot{m}_a i_a = \dot{m}_a (i_a + di_a) + h_a aA (T_i - T_a) dL + i_v \dot{m}_a dW \quad (11)$$

An energy balance at the liquid desiccant-moist air interface gives

$$h_w aA (T_i - T_s) dL - i_w \dot{m}_a dW = h_v aA (T_a - T_i) dL + i_{fg} \dot{m}_a dW \quad (12)$$

An energy balance over the cooling water, cooling water-liquid desiccant interface and global control volumes gives: -

$$dq_c = -U (T_w - T_s) dA_i = -U aA (T_w - T_s) dL \quad (13)$$

$$dq_c = \dot{m}_{c,w} di_{c,w} \quad (14)$$

$$dq_c = \dot{m}_A (i_s + di_s) - \dot{m}_A i_s + \dot{m}_a di_a - \dot{m}_a (i_a - di_a) \quad (15)$$

The solution begins by taking the conditions of all moist air and cooling water differential segments equal to the inlet air and cooling water conditions respectively. The model iteratively loops over these two routines to achieve better approximations of the unknown solution; this is repeated until all unknown variables converge to stable values. In evaluating the optimum values of air and solution mass flow rates and solution concentration the pressure of water vapour in equilibrium with lithium chloride were obtained using correlations derived from the work of Conde [4]. The solution method can be summarized as follows:

1. Assume a length of the absorber and divide the absorber length into differential segments of incremental length dL .
2. Equate moist air and cooling water conditions of all incremental steps to inlet moist air and cooling water conditions respectively.
3. Solve the liquid control volume for each segment marching downward; moist air and cooling water conditions at each node are obtained from step 2 in case of first iteration or routine 2 in case of later subsequent iterations.

The details of the solution are given in Appendix 1.

3.3. The regenerator model

The regenerator model considers the coupled nature of heat and mass transfer processes accompanying the falling film desorption process and takes into account the mass transfer resistances in both the liquid and vapour phases; it uses packed bed correlations [3, 5 & 6] to predict the heat and mass transfer coefficients. The packed bed regenerator considered here consists of a metallic column shell lined with a protective membrane, mist eliminator, liquid distributors to wet the packing bed evenly and initiate uniform contact between the liquid and vapour, packing materials to provide a large wetted surface for the gas stream (maximizing the area available for mass transfer), and may include a packing strainer.

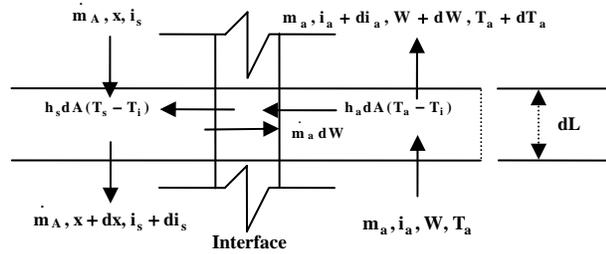


Fig. 6. Differential heat and mass transfer element used to derive regenerator finite difference model equations

Similar to the absorber, steady state mass flux of water absorbed in the differential element of length dL and interfacial area dA_i are given by: -

$$\dot{m}_a dW = K_{m,v} (y_a - y_i) a dL \tag{16}$$

$$\dot{m}_A dx = K_{m,L} (x_i - x) a dL \tag{17}$$

As for the absorber, assuming the weak solution-air interface to be at thermodynamic equilibrium, the two mixtures concentration, temperature and pressure will be coupled to each other at the saturation state and only two properties are sufficient to determine the mixture thermodynamic state. The energy balance for the liquid phase control volume gives

$$i_s \dot{m}_A = \dot{m}_A (i_s + di_s) + h_s a A (T_i - T_s) dL + i_w \dot{m}_a dW \tag{18}$$

An energy balance for the air control volume gives the total heat gained by the air due to transfer of heat from the weak solution-air interface as

$$\dot{m}_a (i_a + di_a) = \dot{m}_a i_a + h_a (T_i - T_a) dA_i + i_v \dot{m}_a dW \tag{19}$$

An energy balance at the weak solution-air interface gives

$$h_w aA(T_i - T_s)dL - i_{fg} \dot{m}_a dW = h_v aA(T_a - T_i)dL + i_w \dot{m}_a dW \quad (20)$$

3.4. Evaporative water cooler model

The evaporative water cooler used in this study consists of a packed bed chamber in which water to be cooled is pumped to distributor troughs at the top of the chamber, splashes down through the chamber and distributed over the packing; a fan draws air vertically upwards through the chamber in counter flow to the falling water. The packing (filling) retards the flow rate of the falling water and increases the water surface exposed to the air.

Finite difference method procedure is employed in this work to derive the packed-bed evaporative water cooler heat and mass transfer governing equations. Figure 5 shows the schematic diagram of the counter flow cooling tower used in this paper to derive the basic differential equations used for modeling the cooler performance using the finite difference method ignoring water lost by drift and heat transfer through cooler walls. In the analysis it is assumed that the cooler cross sectional area and the temperature throughout the water stream at any cross section are both uniform, that temperature normal to the direction of thickness is constant, and that the overall heat and mass transfer coefficients are constant.

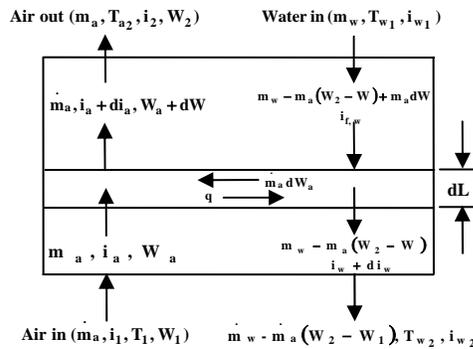


Fig. 7. Differential heat and mass transfer element used to derive cooler finite difference model equations

Considering the control volume of Fig. 7, steady state mass balance for the cooler configuration shows that the mass flow rate of dry air is constant, and that the mass transfer (mass of vapour added to the air) equals mass of water evaporated, hence

$$\dot{m}_a dW = K_m aA(W_{a,i} - W)dL \quad (21)$$

The solar heating system consists basically of a collector for heating the working fluid, a heat exchanger in which the working fluid exchanges heat with load, and working fluid storage tank, Fig. 7 shows a schematic diagram of the solar heating system. A heat exchanger between the solar collector and the load is sometimes provided for economics and safety reasons, material compatibility, to isolate the collector's working fluid from the load, and to prevent freezing of the working fluid. Depending upon the overall objective of the model, the complexity of the system can be increased to reflect the actual conditions by including the pipe losses, heat exchanger effectiveness, etc.

Assuming that all collector components have negligible heat capacity, the glass cover is thin and of negligible solar absorptivity, the collector plate solar absorptivity is close to unity and independent of the angle of incidence, the collector plate fins and back side have highly reflective surfaces, and radiation heat transfer from these surfaces to the insulation inside surface is negligible, the instantaneous total useful energy delivered by the flat plate collector is given by [7]: -

$$Q_u = A_c F_R [S - U_L (T_{fi} - T_a)] = \dot{m}_c C_p (T_{fo} - T_{fi}) \quad (25)$$

4. Results of Simulation

The simulation was accomplished using MatLab[®] code. Property sub-program for LiCl-water, the particular working fluid employed in this study, contains correlations derived from the work of Conde [4]. The investigation considers Khartoum (central Sudan) weather conditions to show the realistic potential of the solar-driven liquid desiccant evaporative water cooler as an alternative water cooling strategy in reducing primary energy consumption.

The computer simulation was arranged to yield air temperature, humidity ratio, and water temperature at the system outlets as well as heat duties of the various system components as functions of specified inlet and other operating conditions. 43^oC dry-bulb temperature, 23.4^oC wet-bulb temperature has been selected as a design case in conducting the simulation; values of the relevant parameters were varied around it. The values of the various parameters influencing the simulation for the reference case are listed in Table1; only one parameter was varied at a time, all others remained fixed at their design values.

Figures 9 and 10 represent the temperature and humidity ratio of the process air supplied to the evaporative water cooler, which has been dehumidified by the desiccant system, as a function of absorber height. As expected, dehumidification performance of the system increases, for the same air and solution mass flow rates, with increase in absorber height; the minimum humidity ratio attained corresponds to the solution vapour pressure. The slope of the curves tends to become flat after when the absorber becomes high. A similar behaviour is exhibited by the absorber cooling duty (heat of absorption removal) as a function of absorber height. To maintain

constant solution temperature, a cooling duty of about 11.2 kW is required at the reference condition.

Table 1. System parameters for reference case.

Parameter	Value
Ambient air conditions	
Dry-bulb temperature	43 ⁰ C
Wet-bulb temperature	23.4 ⁰ C
Return (room) air conditions	
Dry-bulb temperature	24 ⁰ C
Humidity	9.4g/kg
Heating thermal fluid temperature	85 ⁰ C
Cooling water temperature	25 ⁰ C
Cooling water mass flow rate through air-water HE	0.19 liters/sec
Cooling water mass flow rate through absorber	0.57 liters/sec
Air mass flow rate through absorber	1.6 liters/sec
Air mass flow rate through regenerator	1.14 kg/sec
Absorber solution pump mass flow rate	0.38 kg/sec
Regenerator solution mass flow rate	0.38 kg/sec
Solar collector heater pump mass flow rate	0.25 liters/sec

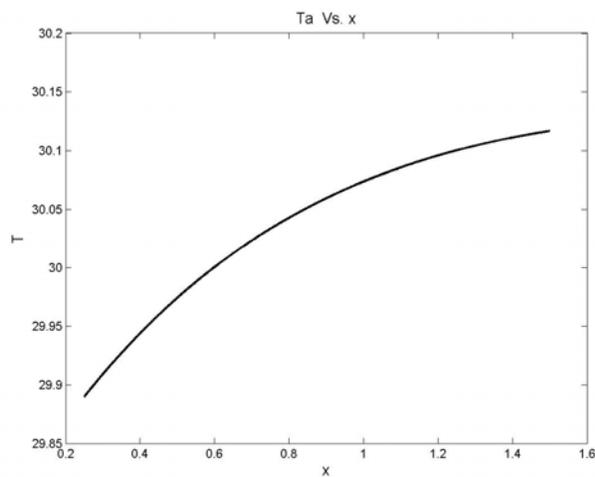


Fig. 9. Process air dry-bulb temperature versus absorber height.

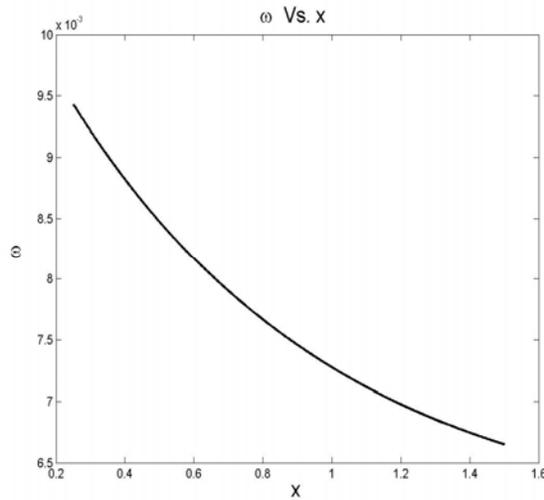


Fig. 10. Process air humidity ratio versus absorber height

5. Conclusions

The parametric study conducted to investigate the system performance shows that the system dehumidifies about 1.6 kg/sec of process air, at the design conditions defined earlier by this work (about 30.1⁰C dry-bulb temperature and 6.6g/kg humidity ratio, which is equivalent to 17⁰C wet-bulb temperature). At these conditions a 1.2 water-to-air ratio cooling tower (evaporative water cooler) using this process air can produce 1.92 kg/sec of water at about 22⁰C. This amount of cold water provides the cooling duties required in the air-to-water heat exchanger, the solution-to-water heat exchanger, the cooling duty required by the absorber as well as providing the cold water required by the radiant system.

This work resulted in a complete design and simulation of the a solar driven desiccant evaporative cooler. Comparing this system with vapour compression water cooling system the system reveals a number of advantages of the former over the latter. Some of the design features of the solar-driven desiccant evaporative water cooler which may be highlighted are

1. Refrigerant condensation is transferred from closed condenser to the environment.
2. Pressure-sealed units are avoided as the whole system operates at atmospheric pressure.
3. Water evaporation process in the regenerator is independent from dehumidification in the absorber, which means more flexibility during operation.
4. The efficient utilization of very low heat source temperatures is achieved by this system, which make it quite suitable for utilization of reject heat
5. The system is environmentally friendly, requires low high grade energy input, and

improves indoor air quality substantially in an energy efficient manner radiant air-conditioning.

6. Using solar-driven desiccant evaporative water cooler is the most promising thermal control technology option for future applications.

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Appendix 1

The following steps describe the solution

1. Guess the liquid desiccant-moist air interface temperature.
 2. Calculate liquid desiccant and moist air-side mass fraction of water at the interface.
 3. Guess moist air humidity ratio and calculate the mole fraction of water vapour in the bulk air.
 4. Calculate a new value of the humidity ratio.
 5. If the difference between the old and new humidity ratio values is less than the assumed convergence criterion, go to step 6 otherwise go to step 3.
 6. Calculate the mass flux of water vapour.
 7. Calculate a new value of the liquid desiccant-moist air interface temperature from an energy balance of interface control volume.
 8. If the difference between the old and new values of liquid desiccant-moist air interface temperature is less than the assumed convergence criterion, go to step iv, otherwise go to step i.
 9. Calculate next segment solution mass flow rate and concentration.
 10. Calculate next segment solution enthalpy from an energy balance of liquid control volume depending on the location of the falling film in the cooling water.
 11. Solve moist air and cooling water differential segments marching upward.
- a. When differential segment contains cooling water tubes
1. Guess the average temperature of cooling water in tube.
 2. Guess the liquid desiccant-moist air interface temperature.
 3. Calculate solution concentration and moist air humidity ratio at interface.
 4. Guess moist air humidity ratio and calculate the mole fraction of water vapour in the bulk air.
 5. Calculate a new value of moist air humidity ratio.
 6. If the difference between the old and new values of moist air humidity ratio is less than the assumed convergence criterion, go to step vii, otherwise go to step iv.
 7. Calculate next segment moist humidity ratio.
 8. Calculate the heat absorbed by the cooling water in the differential segment dq_c .
 9. Calculate next segment moist air enthalpy from an energy balance for the moist air control volume; if the next differential element contains no cooling tubes, go to step x, otherwise go to step ii to calculate next differential element conditions.
 10. Calculate the total amount of heat absorbed by cooling water in tube by adding all dq_c of the differential segments pertaining to that coolant tube.
 11. Calculate temperature of cooling water leaving the tube.
 12. Calculate a new value of cooling water temperature leaving the tube by taking the average temperatures of water entering and leaving the tube; if the

- difference between the old and new values is less than an assumed convergence criterion, go to step vii, otherwise go to 1.
13. If the next cooling tube assembly is not in parallel with the current tube assembly, equate next tube assembly inlet cooling temperature to the mean temperature of cooling water in current parallel tube assembly.
- b.** When the differential segment does not contain cooling water tubes
1. Guess the liquid desiccant-moist air interface temperature.
 2. Calculate moist air humidity ratio at interface.
 3. Guess moist air humidity ratio and calculate the mole fraction of water vapour in the bulk air.
 4. Calculate a new value of moist air humidity ratio; if the difference between the old and new values of moist air humidity ratio is less than the assumed convergence criterion, go to step v, otherwise go to step iii.
 5. Calculate the mass flux of water.
 6. Calculate a new value of interface temperature from the energy balance of the interface control volume; if the difference between the old and new values of interface temperature is less than the assumed convergence criterion, go to step vii, otherwise go to step i.
 7. Calculate moist air humidity ratio for the next segment.
 8. Calculate moist air enthalpy for the next segment from an energy balance for the moist air control volume.
 9. Repeat steps 3 and 4 until the rate of change of any variable between two successive iterations is less than the prescribed tolerance.
 10. Check the moist air humidity ratio at the absorber outlet. Decrease the absorber height and go to step 2 if it is less than the equilibrium humidity ratio of moist air at interface. Increase the absorber height and go to step 2 if the difference between moist air humidity ratio at the absorber inlet and outlet is a positive value and less than 75% of the difference between moist air humidity ratio at absorber inlet and equilibrium moist air humidity ratio at interface temperature. Assume convergence if difference between moist air humidity ratio at the absorber inlet and outlet is positive and lies between 75% and 80% of the difference between moist air humidity ratio at absorber inlet and equilibrium moist air humidity ratio at interface temperature.