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Theoretical modelling and optimization of bubble column dehumidifier for a solar driven humidification-dehumidification system

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Abstract. Availability and utilization of energy and water are the top most global challenges being faced by the new millennium. At the present state water scarcity has become a global as well as a regional challenge. 40 % of world population faces water shortage. Challenge of water scarcity can be tackled only with increase in water supply beyond what is obtained from hydrological cycle. This can be achieved either by desalinating the sea water or by reusing the waste water. High energy requirement need to be overcome for either of the two processes. Of many desalination technologies, humidification dehumidification (HDH) technology powered by solar energy is widely accepted for small scale production. Detailed optimization studies on system have the potential to effectively utilize the solar energy for brackish water desalination. Dehumidification technology, specifically, require further study because the dehumidifier effectiveness control the energetic performance of the entire HDH system. The reason attributes to the high resistance involved to diffuse dilute vapor through air in a dehumidifier. The present work intends to optimize the design of a bubble column dehumidifier for a solar energy driven desalination process. Optimization is carried out using Matlab simulation. Design process will identify the unique needs of a bubble column dehumidifier in HDH system.

1. Introduction

Renewable energies have a vital role in the domain of brackish and seawater desalination. This is due to the reason that potable water shortage is expected to be the major worldwide challenges of the next generation. Limited resources of fresh water remain under stress due to population increase as well as industrial developments. The problem is further amplified by pollution due to poor waste water treatment techniques. Among the various desalination techniques available, humidification dehumidification desalination process is considered as a promising technique for small capacity production plants. Many countries in the Gulf, Caribbean and Mediterranean region have adopted desalination as a sustainable source of fresh water [1]. The dehumidification process by humidification (HDH) is an attractive desalination process because of its simple layout that can be combined with

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solar energy. In addition, it can be designed to minimize the amount of energy discharged into the surroundings. The process proceeds by bringing unsaturated air into contact with warm saline water in a humidifier column to reach certain desired air humidity. This step is followed by condensing out the water vapour from the humidified air by passing it through a dehumidifier column [2]. From the foregoing considerations, it can be seen that the humidifying equipment should essentially comprise a heating device for the supply water and a humidifying apparatus for bringing them into contact. Schematic of a simple closed air open water HDH system is shown in figure 1 [3].

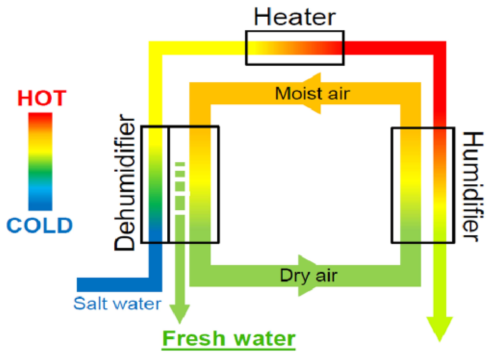

Figure 1. A simple CAOW HDH system.

Unlike other desalination systems, HDH can be easily synchronised with solar energy as the heating source due to its compatibility. However, the productivity of the whole system entirely depends on the dehumidifier efficiency. Water condensation rate in the dehumidifier in turn depends on the intensity of solar radiation which heats up the saline feed water. The dehumidification technique could be either a direct spray of fresh water into humidified air or a condensation of the water content from the humidified air by a heat exchanger. Sorbent and refrigerant dehumidifier is other commonly seen type of this category. But all these types of dehumidifier require external heat input to regenerate the pure water compared to a bubble column dehumidifier [3].

Bubble column dehumidifiers have advantages and disadvantages over other types of dehumidifiers. This can be explained in terms of both entropy generation and cost [4]. Bubbling mode of motion has higher absorption rates and lower stream to stream temperature difference compared to shell and tube type heat exchanger. The mixing of air stream with the water column at different temperatures increase entropy of the system. But the entropy generated is reduced with the low gas side resistance, high liquid hold up and high interfacial area in the column. Nevertheless, the asset of the bubble column dehumidifier lies in transferring the heat from bubble column to the saline water in the helical coil with high heat transfer coefficient in a small area. Earlier works of G P Narayanan et al. had shown that direct contact bubble column dehumidifier reduces the equipment size drastically. [4]. However, due to large volume of non-condensable gas, heat transfer coefficient is low in the apparatus. Mixture of non-condensable gas and vapor create a diffusion resistance for the vapor in the hot moist air. Heat and mass transfer processes are still not clearly defined in a bubble column though many works have been done in this respect for a bubble column reactor [5]. Numerous studies had been carried out to study the hydrodynamics of a bubble column [6-7]. Dehumidifier performance is governed by factors like bubble size, bubble-bubble interactions, bubble rise velocity, gas holdup, and the amount of interfacial area available for heat transfer. The air inlet flow conditions determine the type of flow regimes (homogeneous or heterogeneous). Various design procedures were carried out on the basis of heat transfer [8]. Aim of the present study is to develop a humidity profile of the hot-humid air along the bubble column using mass transfer principles. The results obtained can be used in obtaining the optimum dehumidifier sizing with maximum productivity for the system.

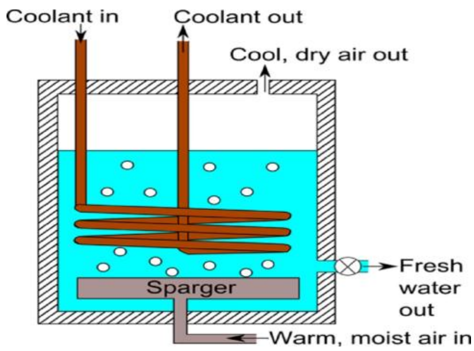

Figure 2. Schematic of a bubble column dehumidifier.

2. Theoretical modelling

As shown in figure 2 [3] hot humid air is bubbled through a column of fresh water and cold saline water flows through a helical coil. The concentration and temperature gradient between the center of the hot air bubble and the surface of bubble is the driving force for condensation in the dehumidifier. The heat liberated from the hot moist air is received by the helical coil. Helical coil intend to heat the cold saline water flowing through it.

2.1 Bubble column dehumidifier modeling

Bubble size depends on the sparger orifice size and rate of air flow. Bubble mean diameter is computed by equating the buoyant force on the bubble which take the bubble away from orifice and the force of surface tension which keep the bubble in orifice. The bubble diameter d_p can thus be expressed as follows [9],

$$d_p = \left(\frac{6 d_0 \sigma}{g \Delta \rho} \right)^{1/3} \tag{1}$$

where d_0 is the sparger orifice diameter (mm) , σ is the surface tension (N/m) , g is the acceleration due to gravity (m/sec^2) , Δρ is the change in density (kg/m^3)
Bubble rise velocity U_b, obtained by equating the buoyant force on the bubble and the drag force is given by [9],

$$U_b = \left(\frac{2\sigma}{d_p \rho_l} + \frac{g d_p}{2} \right)^{1/2} \tag{2}$$

Gas hold up ϵ is given by,

$$\varepsilon = \frac{U}{0.3 + 2U} \tag{3}$$

where U is the superficial gas velocity. Superficial gas velocity is obtained by dividing the mass flow rate of moist air by the cross sectional area of the bubble column. If the gas hold up and bubble diameter are known the interfacial area (a) available in the bubble column is given by,

$$a = \frac{6\varepsilon}{d_b} \tag{4}$$

For the bubble column modeling it is important to analyze heat and mass transfer process simultaneously. Hence energy and mass balance are being carried out to derive the relationship between the temperature of air at exit of dehumidifier and the column height. Balancing equations are developed for a small elemental volume of tower of height dz. Control volume which depicts the dehumidification process in a bubble column is shown in figure 3 [10]. Co-current type of flow is assumed between the sparged air and the cold saline water in the helical coil where ω and i denote the humidity ratio of moist air and enthalpy of saline water respectively [10].

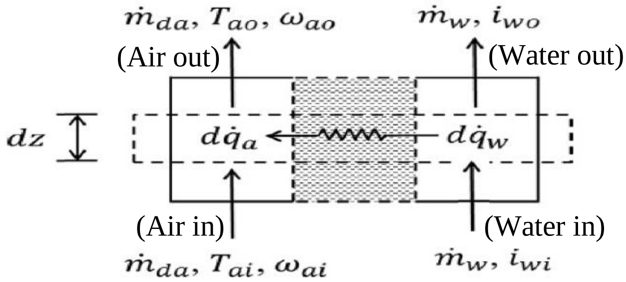

Figure 3. A small element in bubble column dehumidifier.

Certain assumptions are made in the process.

- Dehumidifier column is adiabatic.
- The temperature change in the small volume taken is considered to be linear throughout the length of the water column.
- Bubble motion is characterised by a homogenous flow regime.
- Temperature dependent properties like mass transfer coefficient vary along the column height with change in temperature.
- Direct condensation takes place in the dehumidifier.
- Chilton-colbourn analogy is assumed to hold good in the calculation to compute the mass transfer coefficient [11].

Taking energy balance on the air and water streams for the small element shown in figure 3, we get

$$d\dot{q}_a = d\dot{q}_w \tag{5}$$

Mass balance over the finite volume is given by,

$$d\omega_a = \frac{h_d a A_c}{\dot{m}_{da}}(\omega_{sw} - \omega_a)dz \tag{6}$$

where ω_{sw} is the humidity ratio of saturated air at water temperature.
Mass transfer coefficient h_d in the bubble column is given by [12],

$$h_d = D_{AB}\,0.105\left[\frac{D^{1.33} g^{1/3}(U - \varepsilon U_b)^{1/3}\rho}{\mu_{air}}\right]^{0.8}\left(\frac{\mu_{air}}{\rho D_{AB}}\right)^{1/3}\left(\frac{\mu_{air}}{\mu_w}\right)^{0.14} \tag{7}$$

where D is the column diameter, D_{AB} is the diffusion coefficient, μ is the viscosity (kg/msec).

2.2 Multi start helical coil

Heat transfer enhancement in multiphase flow is accomplished through various heat transfer augmentation techniques. Use of helical coils is a widely used augmentation technique in single phase flow. One of the main advantages of helical coil is its compactness i.e., it has more surface area for a given volume. Also thermal stresses induced can be minimized by the use of helically coiled tubes [13]. The secondary flow in helical coil will also enhance the pressure drop thereby increasing the pumping power.

Less information is available on multi start helical coil in a multiphase flow. In present study dehumidification is achieved by using multi start helical coil in bubble column. The inlet of cold fluid is given to a common header and from the header two concentric set of helical coils are starting with different coil diameters and with constant pitch. The outer and inner coil diameters are 20 cm and 15 cm respectively. Since the helical coils deals with corrosive fluid the material of construction is stainless steel. The coil tube diameter is 14 mm and length is 12 m. Two concentric set having 9 coils

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in which 5 coils start from outer and 4 coils start from inner. The pitch of outer and inner coil is 3.4 cm and 2.4 cm respectively. Schematic of the multi start helical coil is shown in figure 4.

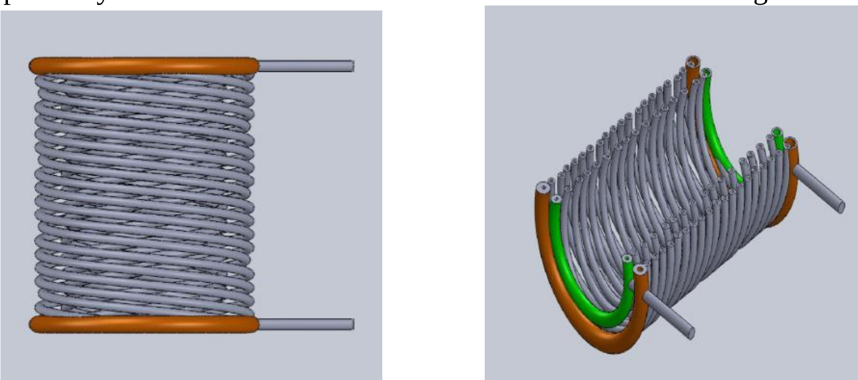

Figure 4. 3D model of multi start helical coil.

2.2.1 Heat transfer formulation. The total amount of heat transfer in the bubble column dehumidifier equipped with multi start helical coil can be calculated from either shell side or tube side. The shell side heat transfer rate can be calculated by using the following equation,
Heat lost by humid air:

$$Q_a = m_{da}(h_{ai} - h_{ao}) \tag{8}$$

The tube side heat transfer rate can be calculated by using the following equation,
Heat gained by cold fluid:

$$Q_w = m_w cp_w (T_{wo} - T_{wi}) \tag{9}$$

Heat balance for dehumidifier [3]:

$$Q_a = Q_w + Latent\ \ heat\ \ of\ condensation \tag{10}$$

The total thermal resistance in the bubble column dehumidifier is shown in figure 5.

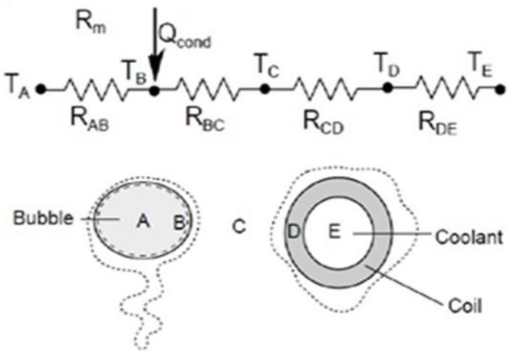

Figure 5. Thermal resistance network [5].

The overall heat transfer coefficient $\dot{Q}_w$ can be calculated using the equation,

$$\dot{Q}_w = U\ LMTD \tag{11}$$

where LMTD is the logarithmic temperature difference and U is the overall heat transfer coefficient. Thermal resistance through the tube wall can be neglected because of a thin stainless steel tube. Also it is assumed that heat transfer from humid air will be within the water column and the heat transfer between humid air and coil can be neglected.

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2.2.1.1 Tube side heat transfer coefficient. Laminar to turbulent transition is determined by critical Reynolds number for the helical coil. Srinivasan et al. [14] has researched on the critical Reynolds number and is related to the curvature ratio as follows,

$$\mathrm{Re}_{cr} = 2100\left[1+12\left(\frac{d_t}{D_c}\right)^{0.5}\right] \tag{12}$$

where d_t is the diameter of the coil and D_c is the diameter of the column.
Inside heat transfer coefficient can be calculated using the following correlation of kirpikov et al. [15] for Nusselt number in turbulent regime,

$$Nu(\mathrm{Pr})^{-0.4} = 0.0456\,\mathrm{Re}^{0.8}\left(\frac{d_t}{D_c}\right)^{0.21} \tag{13}$$

Heat transfer coefficient h can be calculated as,

$$h = Nu\,\frac{k}{d_h} \tag{14}$$

where d_h is the hydraulic diameter.

2.2.1.2 Shell side heat transfer coefficient. General shell side Nusselt number correlation is not suitable for a bubble column dehumidifier to find out the heat transfer coefficient. A widely used correlation by Deckwer [16] is limited to small cylindrical heat exchanging surfaces and is given by,

$$St = 0.1\left(\mathrm{Re}\,Fr\,\mathrm{Pr}^2\right)^{-0.25} \tag{15}$$

where St is the Stanton number , Fr is the Fourier number.
And the correlation developed by Saxena et al. [17] for two phase bubble column with internals is suitable for the present study. Shell side heat transfer coefficient h_o is given as follows from the correlation,

$$h_0 = 0.12\left(g^2\,\frac{\rho_l}{\mu_l}\right)^{1/6}\left[\frac{(\rho_l-\rho_g)}{\rho_l}\right]^{1/3}(k_l\rho_l cp_l)^{1/2} \tag{16}$$

3. Simulation Results and Discussion

The aforementioned equations were simultaneously solved using Matlab. The variation in properties like mass transfer coefficient, diffusion coefficient, air density, superficial gas velocity and bubble rising velocity with change in air temperature are accounted in the simulation program. From the developed program it was possible to study the effect of various parameters that influence the dehumidifier sizing. Optimization of dehumidifier sizing is also possible from the results obtained.

From figure 5 it can be seen that the air exit temperature decreases with increase in bubble column height till it reaches a height of 0.85 m. This is due to the reason that as the bubble column height increases the contact time of air bubble with the cold water increases. This increases the condensation rate thereby reduce the exit air temperature. But beyond a certain height even if the contact time increases the system attains a steady state such that no further condensation is progressed. The simulation is being carried out for air inlet temperature of 68 ^{0}C, air flow rate of 0.06 kg/sec and dehumidifier column diameter of 0.3 m. Hence the bubble column height can be optimized to value of 0.85 m.

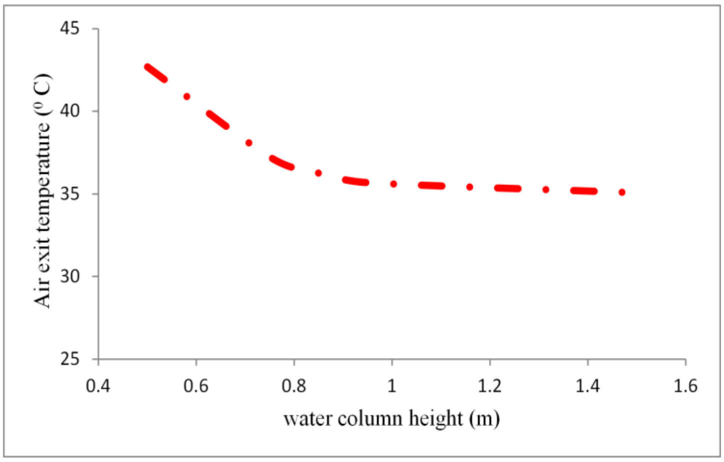

Figure 5. Influence of column height on air exit temperature.

Optimization of air flow rate in dehumidifier is also a necessary step in designing the whole HDH system. This is essential to determine the pipe sizing of the whole system. Consideration should be given to the influence of air flow rate on the condensation rate. The influence of flow rate of hot humid air on the air exit temperature is shown in figure 6.The air exit temperature maintain almost a steady value for a flow rate less than 0.04 kg/sec. After this value the exit temperature goes on increasing. Hence the flow rate can be optimized to value of 0.04 kg/sec. Here the simulation is carried out for column height of 0.85 m.

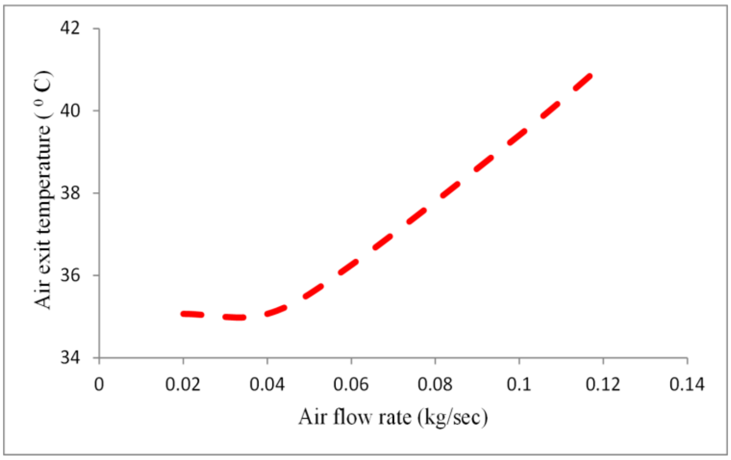

Figure 6. Influence of moist air flow rate on air exit temperature.

Sparger is an important internal accessory of a bubble column to determine the equipment efficiency. Sparger orifice diameter influences the bubble dynamics of the column. The superficial air velocity in the bubble column is determined by the sparger orifice. When the height to diameter ratio is lower in a bubble column the sparger design determines the efficiency of a bubble column dehumidifier. The orifice diameter control weeping condition in the bubble column. Weeping takes place when the kinetic energy of the moist air coming out of the sparger is not sufficient to support the liquid column height above to it. Pressure drop in the column is greatly influenced by the number of orifices and size of orifices. Hence the effectiveness of the equipment depends on sparger diameter to a large extent. From figure 7 it can be seen that the air exit temperature increases with increase in sparger diameter. From the figure it is also clear that, for a flow rate of 0.06 kg/sec and column height of 0.85 m, the sparger orifice diameter can be optimized to around 2 mm.

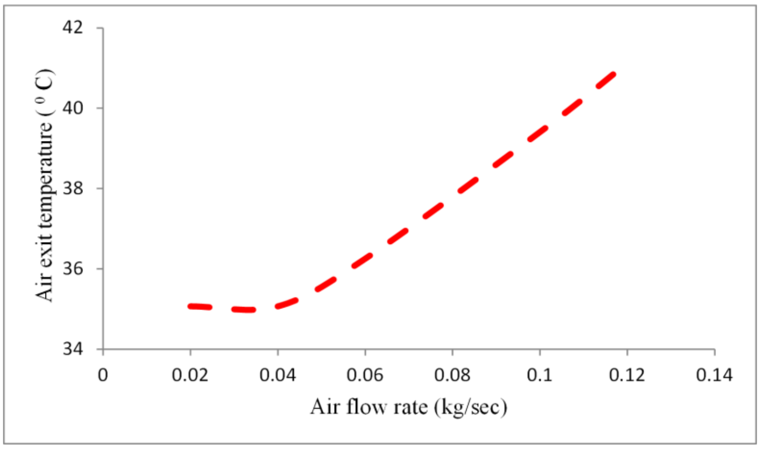

Figure 7. Exit air temperature variation with sparger diameter.

From figure 8, it is clear that the theoretical model value of exit air temperature agrees with the values obtained by Yi Li et al. Desalination (2006), 196 pp. 188-209 [18]. The slight change in the values seen, is due to the reason that in Yi Li et al. work the flow rate of moist air and other properties like diffusion coefficient and mass transfer coefficient was taken as constant along the column. But in the present study the temperature dependent variable tend to change slightly with temperature change as the moist air moves up in the bubble column.

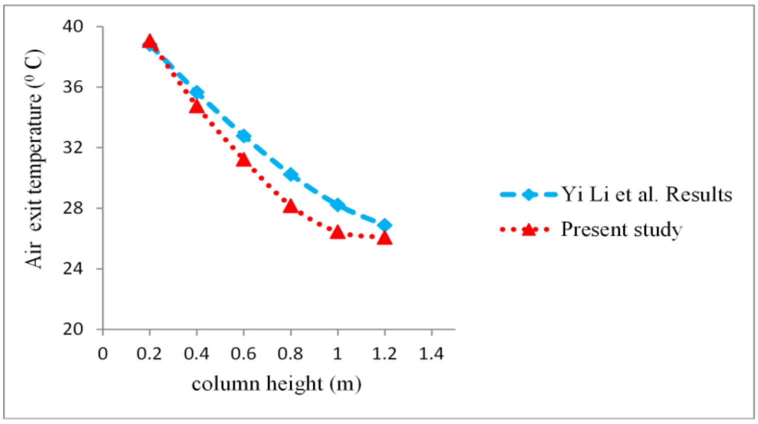

Figure 8. Comparison study between Li et al. (2006) result and present study.

4. Conclusion

Humidification dehumidification desalination technique driven by solar energy is one of the ideal and cost effective methods for decentralized small scale desalination systems. The effectiveness of the system can be controlled only with the optimization design on the system. Much more specifically, the optimization design of dehumidifier influences the overall efficiency of the system. As the moist air gets condensed in the dehumidifier, attention on the equipment design can improve the gross output ratio as well as recovery ratio of the system. Many methods have been well explained in designing a bubble column dehumidifier using heat transfer modelling and bubble dynamics. But a method based on the mass transfer is not well explained yet. Using the above calculation based on concentration gradient in the water vapour-air bubble, it is reliable to optimize the dehumidifier design in terms of the hot-humid air flow rate ,column height and sparger orifice diameter. In the present study the water column height in bubble column was optimized to 0.85 m depending on the air exit temperature. Air flow rate and sparger diameter for 0.3 m diameter bubble column was optimized to 0.04 kg/sec and 2 mm respectively. Hence further modification on the above discussed method can give way to optimization of much more variables involved in a HDH system. The high energy consumption and large sizing of the equipment are considered to be one of the disadvantages of a HDH system. Incorporating heat transfer and thermodynamic principles to the above calculations can bring down these disadvantages to a large extend. Hence the solar driven HDH system could be a boon to the coming generation.

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