International Journal of Current Engineering and Technology
E-ISSN 2277 – 4106, P-ISSN 2347 - 5161

Available at http://inpressco.com/category/ijcet

Research Article

Simplified Mathematical Modelling of Dehumidifier and Regenerator of Liquid Desiccant System

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Accepted 8 March 2014, Available online 01 April 2014, **Vol.4, No.2 (April 2014)**

Abstract

Mathematical model represents all the important features of the system for the purpose of deriving the mathematical equations governing the system's behavior. This paper describes a simple mathematical model for the preliminary design of an air dehumidification and regeneration process occurring in a structured packing using liquid desiccant cooling technology. With the help of control volumes of dehumidifier and regenerator, mass balance and energy balance equations have been derived for both the components of liquid desiccant system. These governing differential equations will be used in simulation of dehumidifier and regenerator. Performance measures of dehumidifier in terms of mass of condensation of the air and dehumidification effectiveness have been developed. Mass of evaporation of the air and regenerator effectiveness has also been developed to measure performance of regenerator.

Keywords*: Dehumidifier, regenerator, Mathematical model, Performance measures, mass of condensation*

1. Introduction

Humidity control is important in many engineering applications, such as space air conditioning, storage warehouses, process industries and many others. In space AC, where conventional vapor compression cooling is used, humidity control is often accomplished by cooling the air below its dew point. In hot and humid climates, this method is inefficient, since it should be followed by reheating the air to a safe temperature before it is introduced into the conditioned space. The evaporator in the vapor compression system operates at a lower temperature than is required to meet the sensible cooling load, leading to a lower coefficient of performance and, consequently, higher energy requirements. The effectiveness of evaporative cooling relies heavily on the existence of low relative humidity conditions, being higher as the relative humidity decreases. The humidity puts an extra load on the vapor compression AC systems and renders the evaporative cooling systems ineffective. Under high humidity conditions, energy efficient vapor compression systems operate at higher evaporator temperatures and found unable to maintain the indoor relative humidity within a comfortable range. This calls for dehumidifying the air prior to entering the evaporator.

Air conditioning in residential buildings is a large and growing market, almost exclusively covered by electrical compression systems. In order to supply this market with sustainable technologies the development of cost effective solar or waste heat driven cooling systems is necessary. The widely used closed absorption technology does not cover the low power range under 10 kW cooling capacity, so that there is currently no alternative to electrical split units available.

As buildings with low energy demand are often equipped with mechanical ventilation systems, it is obvious to consider air based thermal cooling technologies, such as open desiccant cooling systems, for low power applications. Small desiccant rotors, heat exchangers and humidifiers are market available for volume flows in the range of typical mechanical ventilation systems. Both conventional desiccant cooling units and liquid sorption systems dehumidify the outside air, which is then pre-cooled, humidified and injected into the rooms. The direct humidification of inlet air still causes concerns about hygiene, especially if maintenance is not guaranteed, which is generally the case in residential buildings.

To avoid any hygienic problems for low power cooling applications, a new system configuration is proposed in this work, which shifts the whole air treatment to the exhaust air side and only uses sensible cooling for the outside air stream. The room exhaust air is dehumidified by a desiccant wheel or by liquid desiccant systems, pre-cooled in a heat exchanger using an additional humidified air stream and further cooled by direct humidification. Finally the cooled return air is used to cool the supply air in an efficient heat exchanger.

Separating the control of humidity and temperature by means of desiccants could result in energy savings, as well as improved humidity control as explained by Oberg et al. (1998). Desiccants are chemicals with great affinity to

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moisture. They may be used effectively as a supplement to conventional vapor compression systems to remove the latent part of the cooling load. Desiccants are efficient in handling latent loads (reducing the humidity), whereas the evaporator of the vapor compression system is efficient in handling the sensible cooling loads.

As the desiccant vapor pressure increases due to the presence of the moisture that it has attracted, the desiccant material is transferred to a reactivation process. In the reactivation process, hot air is passed over the desiccant. The vapor pressure of the hot air is lower than the desiccant surface which forces the moisture to transfer from the desiccant surface into the hot air stream. The moist hot air is then exhausted from the system into the outdoor air. The desiccant material that has had the trapped moisture removed is now prepared to attract moisture as it is transferred back into the process air path. The dry process air leaving the desiccant is then passed over a conventional cooling coil which addresses the sensible cooling work required to meet the air specification of the conditioned space.

Air dehumidification can be achieved by two methods: (I) cooling the air below its dew point and removing moisture by condensation, or (2) sorption by a desiccant material. Desiccants in either solid or liquid forms have a natural affinity for removing moisture. As the desiccant removes the moisture from the air, desiccant releases heat and warms the air, i.e., latent heat becomes sensible heat. The dried warm air can then be cooled to desired comfort conditions by sensible coolers (e.g., evaporator coils, heat exchangers, or evaporative coolers.). To re-use the desiccant, it must be regenerated or reactivated through a process in which moisture is driven off by heat from an energy source such as electricity, waste heat, natural gas, or solar energy.

2. Literature Review

Air dehumidification is an important operation not only in industry but also in comfort cooling (Gandhidasan, 2004). Although many methods are available for this operation, liquid desiccant system is more attractive due to its flexibility in operation along with an added benefit of absorbing inorganic and organic contaminants in the air (Oberg and Goswami, 1998) and the ability of using a lower regeneration temperature coming for example, from solar energy. The strong, cool desiccant solution is sprayed at the top of the dehumidifier (absorber) after passing through a heat exchanger where it is precooled by the cooling medium (water). The ambient humid air enters the dehumidifier at the bottom. The moisture content of the process air in the unit is controlled by either controlling the desiccant temperature or by controlling both the temperature and the concentration of the desiccant. A low moisture content of the process air can be obtained at the outlet of the absorber by maintaining a low desiccant temperature or a high concentration at the inlet of the absorber (Khan, 1994).

Many literatures are dedicated to the investigation of performance of liquid desiccant dehumidifiers and regenerators (Stevens et al., 1989, Chung et al., 1996, Fumo and Goswami, 2002). One-dimensional differential heat and mass transfer models are well established and were frequently used to study the performances of packed bed dehumidifiers and regenerators. Factor and Grossman (1980) developed a theoretical model for a test column with LiBr solutions. The interface temperature and concentration were assumed to be the bulk liquid temperature and concentration.

The dehumidification process can be accomplished using the equipment of various configurations such as a finned-tube surface in a column (Johannsen, 1984; Mahmoud and Ball, 1988), coil-type absorber (Khan, 1998), spray tower (Scalabrin and Scaltriti, 1990; Chung and Wu, 1998) or packed tower (Factor and Grossman, 1980; Lof et al., 1984; Gandhidasan et al., 1987; Oberg and Goswami, 1998). Out of these basic configurations, the use of a packed tower configuration may be advantageous because of a large rate of heat and mass transfer per unit volume. The packings are of two major types namely random and regular. The regular packings offer the advantage of low pressure drop for the air stream at the expense of more costly installation than random packing. Hence many researchers, as mentioned earlier, had chosen random packing for their studies. The simultaneous heat and mass transfer between the air and the desiccant have a large impact on the dehumidifier performance.

The thermal performance of the dehumidifier is measured by the amount of water condensed from the humid air. Three different theoretical models, namely the finite difference model (Factor and Grossman, 1980, Gandhidasan et al., 1987, Oberg and Goswami, 1998), the effectiveness-NTU model (Stevens et al., 1989, Sadasivam and Balakrishnan, 1992), and the model based on fitted algebraic equations (Khan and Ball, 1992, Khan, 1994), have been developed for the analysis of packed bed absorption dehumidifiers. Out of these three models, the finite difference model has received more attention since this model gives accurate performance predictions based on fundamental equations. For this model, a step-wise heat and mass balance across the dehumidifier is used to determine the performance of the dehumidification process. Since the outlet conditions of the desiccant are unknown, an iterative solution is necessary until the results converge to the known inlet conditions at the top of the dehumidifier. Due to the high number of variables involved in the process, the analysis becomes increasingly complex and it mandates the use of a numerical solution to handle the iterative calculations.

Further, Fumo and Goswami (2002) made modifications to account for the higher surface tension of LiCl and higher water concentration in brines as compared to water concentration in TEG. A correction factor was used in their model to describe the reduction of area for mass transfer in the contact column. Gas phase heat transfer coefficient was also determined from known mass transfer coefficient by applying the heat and mass transfer analogy.

The model was used to predict experimental findings and the results of comparison were satisfactory. For simplification, overall gas-phase heat and mass transfer coefficients with different values of Lewis factor were utilized in simulation to study the performance of dehumidifiers/regenerators. Liquid phase heat and mass transfer resistances were usually much smaller than those in the gas phase. Overall gas-phase heat and mass transfer coefficients were measured (Chung et al., 1996; Chung and Wu, 2000) and utilized in simulation to study the performance of dehumidifiers/regenerators (Elsayed et al., 1993; Stevens et al., 1989). Due to the complexity in heat and mass interactions between the two phases with different packing materials, liquid properties and operating conditions, and so on, the overall volumetric heat and mass transfer coefficients did not necessarily follow the

Correlations for rate of vaporization in regenerators and rate of condensation in dehumidifiers were studied (Park et al., 1996; Patnaik et al., 1990). Simulation results or experimental data fitted correlations for the effectiveness of heat and mass transfer processes were available (Ullah et al., 1988; Chung, 1994; Martin and Goswami, 2000).

However, the validity of these correlations was found to be restricted to the type of equipments and operating conditions investigated. Though some correlations were based on the dimensionless groups obtained from dimensional analysis using Buckingham-Pi theorem, the theorem itself could not find out the exact functional forms of the correlations. Thus, accuracy will inevitably be compromised for broader ranges of validity and/or simplicity of the expressions. Charts of humidity and enthalpy effectiveness obtained from finite difference model as a function of design variables using a calcium chloride solution as the desiccant were presented by Elsayed et al. (1993), but no mathematical expression was provided.

3. Modeling of Dehumidifier

Mathematical model represents all the important features of the system for the purpose of deriving the mathematical equations governing the system's behavior. The mathematical modelling includes enough details to be able to describe the system in terms of equations without making it too complex. The mathematical model may be linear or nonlinear, depending on the behavior of the system's components. Linear models permit quick solutions and are simple to handle; however, nonlinear models sometimes reveal certain characteristics of the system that cannot be predicted using linear models.

We can use a very crude or elementary model to get a quick insight into the overall behavior of the system. The model is refined by including more components and details so that the behavior of the system can be observed more closely (Ritunesh et. al., 2009). Thus a great deal of engineering judgment is needed to come up with a suitable mathematical model of vibrating system.

Schematic diagram of an air dehumidification system with liquid desiccant is given in figure 1. A honeycomb celdek packing has been used in dehumidifier in present work. Strong desiccant solution flows in downward direction through celdek packing and air flows in upward direction through the packing.

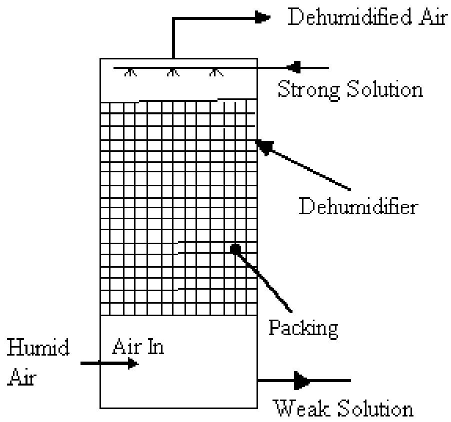

Fig. 1 A simlified schematic of air dehumidification with liquid solution

The desiccant solution is not able to stick on the inner side of packing due to reasons like turbulence, surface irregularity and impurity which might influence the surface tension. To account for this in modeling wetness factor f_{ab} has been employed as used by Jain et al. (2000). This is defined as the ratio of actual tube surface area wetted by the falling desiccant to the gross surface area of the flute of packing.

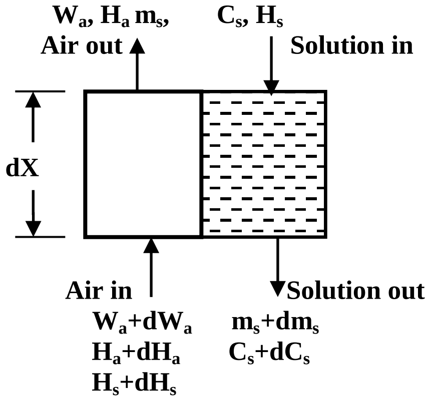

Fig 2 Control volume of Dehumidifier

To simulate the absorber which is a heat and mass exchanger, fundamental mass and energy equation have been written for air and desiccant solution. The steady state mass and energy equations for a control volume as shown in figure 2 are:

Mass Balance equation for air

Moisture present in the air leaving the control volume = Moisture present in the air entering the control volume + moisture added to the air due to mass transfer from liquid to air

$$m_a\ .W_a = m_a\ .(W_a + dW_a) + f_{ab}.h_m.P_a.dX\left(W_{eq} - W_a\right)$$

$$m_a\ .\frac{dW_a}{dX} = f_{ab}.h_m.P_a.\left(W_a - W_{eq}\right) \tag{1}$$

Mass Balance equation for desiccant

Mass of water in liquid desiccant leaving the control volume = Mass of water in liquid desiccant entering the control volume - Mass of water removed from the liquid desiccant due to mass transfer to air

$$(m_s + dm_s)(1 - C_s + dC_s) = m_s.(1 - C_s) - f_{ab}.h_m.P_a.dX\left(W_{eq} - W_a\right)$$

$$m_s.(1 - C_s) + m_s.dC_s + ddm_{s_{ms}}.(1 - C_s) - dm_s.dC_s = m_s.(1 - C_s) - f_{ab}.h_m.P_a.dX\left(W_{eq} - W_a\right)$$

Neglecting second order derivatives, we have

$$m_s.dC_s + dm_s.(1 - C_s) = f_{ab}.h_m.P_a.dX\left(W_a - W_{eq}\right)$$

$$\frac{d(m_s(1 - C_s))}{dX} = f_{ab}.h_m.P_a.dX\left(W_a - W_{eq}\right) \quad (2)$$

Energy Balance Equation for Air

Total enthalpy of the air leaving the control volume = Total enthalpy of the air entering the control volume + increase in energy of the air due to mass transfer from liquid desiccant to air + increase in enthalpy of the air due to heat transfer from liquid desiccant to air

$$m_a.H_a = m_a.(H_a + dH_a) + H_{fg}.h_m.P_a.dX\left(W_{eq} - W_a\right) + U_{as}.P_a.dX\,(T_s - T_a)$$

$$m_a.dH_a = H_{fg}.h_m.P_a.dX\left(W_a - W_{eq}\right) + U_{as}.P_a.dX\,(T_a - T_s)$$

$$m_a.\left(C_{pm}.dT_a + dW_a.H_{fg}\right) = H_{fg}.h_m.P_a.dX\left(W_a - W_{eq}\right) + U_{as}.P_a.dX\,(T_a - T_s)$$

$$m_a.\left(C_{pm}.dT_a\right) + m_a.dW_a.H_{fg} = H_{fg}.h_m.P_a.dX\left(W_a - W_{eq}\right) + U_{as}.P_a.dX\,(T_a - T_s)$$

Neglecting higher derivative, we have

$$m_a.C_{pm}.\frac{dT_a}{dX} = H_{fg}.h_m.P_a\left(W_a - W_{eq}\right) + U_{as}.P_a(T_a - T_s) \quad (3)$$

Energy Balance Equation for desiccant

Total enthalpy of the liquid desiccant solution leaving the control volume = Total enthalpy of the liquid desiccant solution entering the control volume -decrease in energy of the liquid desiccant solution due to mass transfer from liquid desiccant to air - decrease in energy of the liquid desiccant solution due to heat transfer from liquid desiccant to air + increase in energy due to heat transfer from water to liquid desiccant

$$(m_s .+ d_{ms})(H_s + dH_s) = m_s.H_s - H_{fg}.h_m.P_a.dX\left(W_{eq} - W_a\right) - U_{as}.P_a.dX\,(T_s - T_a) + U_{sw}.P_a.dX\,(T_w - T_s)$$

$$m_s.H_s + m_s.dH_s + d_{ms}.H_s + d_{ms}.dH_s = m_s.H_s - H_{fg}.h_m.P_a.dX\left(W_{eq} - W_a\right) - U_{as}.P_a.dX\,(T_s - T_a) + U_{sw}.P_a.dX\,(T_w - T_s)$$

$$m_s.dH_s + d_{ms}.H_s = H_{fg}.h_m.P_a.dX\left(W_{eq} - W_a\right) + U_{as}.P_a.dX\,(T_a - T_s) + U_{sw}.P_a.dX\,(T_s - T_w)$$

$$\frac{d(m_s + H_s)}{dX} = H_{fg}.h_m.P_a\left(W_{eq} - W_a\right) + U_{as}.P_a(T_a - T_s) + U_{sw}.P_a(T_s - T_w) \quad (4)$$

Energy Balance Equation for Water

Total enthalpy of the water leaving the control volume = Total enthalpy of the water entering the control volume - decrease in energy of the water due to heat transfer from the water to solution

$$m_w.H_w = m_w(H_w + dH_w) + U_{sw}.P_a.dX\,(T_w - T_s)$$

$$m_w.\frac{dH_w}{dX} = U_{sw}.P_a(T_s - T_w) \quad (5)$$

Where w_{eq} is equilibrium moisture contents of desiccant defined in terms of water vapour pressure (P_{eq}) of desiccant solution at the given concentration and temperature as

$$W_{eq} = 0.62\,\frac{P_{eq}}{P_{atm} - P_{eq}} \quad (6)$$

The equations are non linear coupled first order differential equations and have to be solved simultaneously.

The heat and mass transfer coefficients at any step are evaluated by using appropriate correlations discussed below:

Nusselt number is given by Gnuelinski correlation [Incropera&Dewitt (2000)]

$$Nu = \frac{\left(\frac{ff}{8}\right).(Re - 1000).Pr}{1 + 12.7\left(\frac{ff}{8}\right)^{0.5}\left(Pr^{2/3} - 1\right)} \quad (7)$$

0.5 <Pr< 2000 and 3000 < Re < 5× 10^6

Where for smooth tubes friction factor ff is given by equation

$$ff = (0.79\,log_n\,Re - 1.64)^{-2} \quad (8)$$

Air side heat transfer coefficient is calculated as

$$h_a = \frac{Nu.K_a}{D_h} \quad (9)$$

Using Reynolds analogy [Stocker (1968)],

$$\frac{h_a}{h_m.C_{pm}} = (Le)^{2/3} \quad (10)$$

The mass transfer coefficient is obtained as:

$$h_m = \frac{h_a}{C_{pm}.(Le)^{2/3}} \quad (11)$$

Solution side heat transfer coefficient for laminar flow of the film is obtained as following Ramm (1968):

$$Nu_s = 0.67\,Re_s^{0.11}\left(\frac{Pr_s\,S_T}{L_a}\right)^{0.33} \quad \text{For } Re_s < 2300 \quad (12)$$

Where $Re_s = \frac{4\Gamma}{\mu_s}$ and $\Gamma = \frac{m_s}{N_T.\,\pi.D_h}$

Film thickness for single phase flow is

$$S_o = \left(\frac{3\times\Gamma\times\mu_s}{\rho_s^2\times 9.81}\right)^{1/3} \quad (13)$$

For two phase flow it is given by

$$S = \left(1 - 0.22(V_s - 4)\right).S_o \qquad (14)$$

And reduced film thickness is calculated using S from the following equation

$$S_r = \frac{S}{0.9085 \times {Re_s}^{1/3}} \qquad (15)$$

Convective Heat Transfer coefficient can be determine by equation

$$h_s = \frac{Nu_s K_s}{S_r} \qquad (16)$$

Water side heat transfer coefficient h_w is obtained by Kern (1950)

$$Nu_w = 0.36\, Re_w^{0.55} Pr_w^{1/3}, \qquad (17)$$

For 2000 <Re_w< 1000000

$$h_w = \frac{Nu_s K_s}{D_h} \qquad (18)$$

$$Re_w = \frac{D_h \times M_w}{a_s \times \mu_w} \qquad (19)$$

Overall Heat Transfer coefficient for heat transfer from desiccant to water is given by:

$$\frac{1}{U_{sw}} = \frac{1}{h_w} + ln\left(\frac{D_o}{D_i}\right)\frac{D_o}{2K_t} + \frac{D_o}{D_i}\frac{1}{h_s} + \mathrm{R_{fs}} + \mathrm{R_{fw}} \qquad (20)$$

Overall Heat Transfer coefficient for heat transfer from desiccant to water is given by:

$$\frac{1}{U_{aw}} = \frac{1}{h_w} + ln\left(\frac{D_o}{D_i}\right)\frac{D_o}{2K_t} + \frac{D_o}{D_i}\frac{1}{h_a} + \mathrm{R_{fa}} + \mathrm{R_{fw}} \qquad (21)$$

4. Modeling of Regenerator

In regenerator, the desiccant solution is heated and it gets regenerated by losing its moisture to air. Schematic diagram of Regenerator with liquid desiccant is given in figure 3. A honeycomb celdek packing has been used in regenerator in present work. Weak desiccant solution flows in downward direction through the celdek packing and hot air flows in upward direction through the packing.

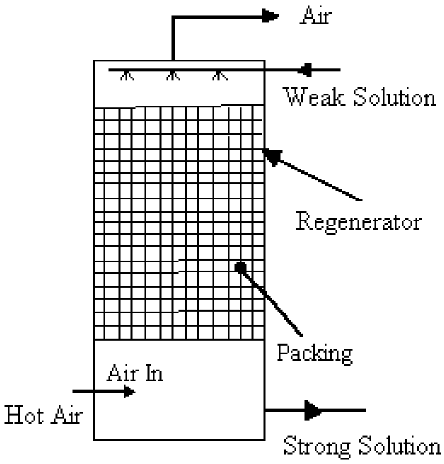

Fig. 3 Simlified schematic of Regenerator with liquid solution

As the configuration of regenerator is similar to that of dehumidifier, its modeling process is also similar. Only difference is that hot air will be used for regeneration.

Wetness factor f_{rg}has been employed to determine actual area of contact between air and desiccant for mass and heat transfer analysis. Control volume of regenerator is given in figure 4. To simulate regenerator, fundamental mass and energy conservation equations have been written for air and desiccant solution.

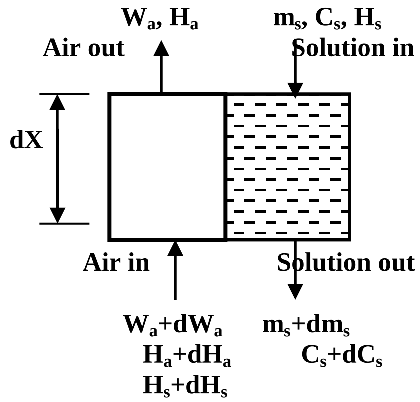

Fig. 4 Control volume of Regenerator

Governing Equations of Regenerator are written in similar manner as of dehumidifier. The final equations are as follows:

Mass balance equations for air

$$m_a \;.\frac{dW_a}{dX} = \; f_{rg}.h_m.P_a.\left(W_a - W_{eq}\right) \qquad (22)$$

Mass Balance equation for desiccant

$$\frac{d\left(m_s(1 - C_s)\right)}{dX} = \; f_{rg}.h_m.P_a.dX\left(W_a - W_{eq}\right) \qquad (23)$$

Energy Balance Equation for Air

$$m_a \;.C_{pm}.\frac{dT_a}{dX} = \; H_{fg}.h_m.P_a\left(W_a - W_{eq}\right) + U_{as}.P_a(T_a - T_s) \qquad (24)$$

Energy Balance Equation for desiccant

$$\frac{d(m_s H_s)}{dX} = \; H_{fg}.h_m.P_a\left(W_{eq} - W_a\right) + U_{as}.P_a(T_a - T_s) + U_{sw}.P_a(T_s - T_w) \qquad (25)$$

Heat and mass transfer coefficients (h_a,h_s,h_m) at any step are calculated using same equations as for dehumidifier.

5. Performance Measure of Dehumidifier and Regenerator

The performance of the dehumidifier is evaluated by calculating the moisture removal rate and the column effectiveness. The rate of moisture removal from the air (water condensation rate) is calculated from Eq. (26).

$$\dot{m}_{cond} = \;(W_{in} - \; W_{out}) * m_a * A \qquad (26)$$

The dehumidification effectiveness, ε_y is an air-side characteristic parameter which is related to the mass transfer effectiveness during the dehumidification operation.The effectiveness is defined as the ratio of the actual change in moisture content of the air leaving the absorber to the maximum possible change in moisture content under a given set of operating conditions. The dehumidifier effectiveness, ε, is expressed mathematically as:

$$\varepsilon = \frac{W_{in} - W_{out}}{W_{in} - W_{equ}} \quad (27)$$

W_{equ} is the absolute humidity of the air, which is at equilibrium with the desiccant solution at the inlet concentration and temperature.

The performance of the regenerator is evaluated by calculating the evaporation rate and the column effectiveness. The rate of evaporation from the air is calculated from Eq. (28).

$$\dot{m}_{evap} = (W_{out} - W_{in}) * m_a * A \quad (28)$$

The regeneration effectiveness, ε is defind in terms of humidity effectiveness. It is defined as the ratio of the actual change in specific humidity of the air flowing through regenerator to the maximum possible change under a given set of operating conditions. The regenerator effectiveness, ε_y, is expressed mathematically as:

$$\varepsilon = \frac{W_{out} - W_{in}}{W_{equ} - W_{in}} \quad (29)$$

Conclusion

Mathematical models for dehumidifier and regenerator have been derived from the control volumes of each component. The governing equations have been derived in differential form. These equations are non linear coupled first order differential equations and have to be solved simultaneously. Analytically closed form of solution of these equations is not possible. Thus these equations will be solved numerically by dividing the absorber/regenerator into a large number of steps. For the given input information at inlet of any step (solution, air temperature, mass flow rates, air humidity, solution concentration), the outlet conditions will be obtained assuming constant properties and heat and mass transfer coefficients within that step. Step by step calculations will be carried out from top to bottom of absorber/regenerator with appropriate step size depending upon the desired accuracy.

Acknowledgement

Authors are thankful to the Punjab Technical University, Jalandhar-Kapurthala Highway, Kapurthala, Head and faculty members of Mechanical Engineering Department of IET Bhaddal Technical Campus Ropar for their valuable support to carryout this research.

Nomenclature

A cross sectional area of dehumidifier/ regenerator [m^2]
H enthalpy [kJ/kg]
C concentration of desiccant [kg/kg of solution]
C_{pm} mean specific heat [kJ/kg.K]
D_h hydraulic diameter [m]
dX thickness of Control Volume [m]
f wetness factor [---]
h_m mass transfer coefficient [kg/m^2.s]
K thermal conductivity [W/m^2.K]
L length [m]
m mass Flow Rate [kg/s]
P pressure [bar]
S film thichness [m]
T temperature [K]
U overall heat transfer coefficient [W/m^2.K]
W specific humidity [Kg/Kg of dry air]
ρ density [m/s^2]
Nu Nusselt number [---]
ff friction factor [---]
Re Reynolds number [---]
Pr Prandlt number [---]
Le lewis number [---]
Sr Sherwood number [---]

Subscripts

a air
ab absorber
r,rg regenerator
atm atmosphere
s solution
eq,equ equillibrium
in inlet
out outlet
as air to solution
sw solution to water
r reduced
w water
U overall heat transfer coefficient
Cond condensate

Greek letter

ε effectiveness

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