Analysis of Thermal Effects in Packed Bed Liquid Desiccant Dehumidifiers

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Abstract

An analysis of the performance of a packed bed liquid desiccant dehumidifier is presented. The enthalpy and material balance equations written over a differential height of the bed incorporate the effect of the heat of dilution of the desiccant liquid, the liquid–air interface temperature and the fraction of the evolved heat that is accounted for in the air stream. Numerical results obtained are discussed in terms of the physical and operating parameters of the system.

Introduction

The most common thermodynamic cycle used for comfort conditioning of living space is the well-known vapour–compression cycle. If this cycle is to be used for dehumidification, the air to be dehumidified has to be cooled below its dew point. Obviously, this method of dehumidification is extremely energy intensive. Unlike the vapour–compression cycle, the open cycle or desiccant air-conditioning systems separate the sensible and latent heat loads, first by removing the moisture content to reduce the latent heat load and then by evaporative cooling to reduce the sensible heat load. Desiccant systems, using hydrophilic liquids or solids, utilize waste heat energy or other low grade energy sources, including solar thermal energy, for their regeneration.

In general, solid desiccants such as silica gel and molecular sieves achieve a higher degree of dehumidification than their liquid counterparts [1] and a number of investigators have studied these systems in detail. Shelpuk [2] has reviewed much of this work. Liquid desiccants (such as lithium chloride, calcium chloride, and various glycols) have, in general, lower drying capacities than the solids. However, systems employing liquid desiccants have significant advantages, including the possibility of cooling the air by manipulating the air and liquid flow rates and the liquid concentration and lower regeneration temperatures of the spent desiccant liquids. One of the earliest studies on open cycle air-conditioning systems using liquid desiccants was by Löf [3] in 1955. Gari *et al.* [4] noted the importance of regeneration of the liquid desiccant, since this operation was the primary energy consumer in the open cycle air-conditioning system. While Patnaik *et al.* [5] studied the dehumidification of air experimentally in a packed bed using a lithium bromide–water system with different types of distributors, Scalabrin [6] studied the regeneration of the lithium chloride–water liquid desiccant system.

In open cycle or liquid desiccant air-conditioning systems, the dehumidifier is the heart of the system [7]. The role of the dehumidifier is similar to the absorber in the absorption–refrigeration cycle. In the dehumidifier, the air and liquid desiccant can be contacted by any one of the following methods: (a) bubbling the air through the liquid, as in a tray column; (b) spraying the liquid in a fine dispersion in an upward air stream; (c) spraying the liquid over a bank of cooling tubes past which air is blown; and (d) passing the air and liquid streams through a packed bed. While each of the above contacting methods has its own advantages and disadvantages, the most promising appears to be the packed column. This is because a packed bed offers the greatest interfacial area between the air stream and the liquid stream, leading to high interchange rates of heat and mass between the two phases. The dehumidification of air in a packed bed with counter-current flow of the liquid desiccant is fundamentally similar to physical gas absorption, a transfer operation frequently encountered in the chemical process industries. This facilitates the transfer of information and procedures used in the analysis and design of gas absorption units to packed bed liquid desiccant dehumidification operations, with suitable changes.

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Previous work

Olander [8] developed the heat and mass transfer rate relationships for packed bed gas absorption operations. In this analysis there is no evolution of heat at the interface. Sherwood and Pigford [9] have described a design procedure for adiabatic gas absorption of a single solute in a packed tower. They neglect liquid-phase resistances to heat and mass transfer and also to change in liquid and gas flow rates through the tower. Treybal [10], however, following an approach similar to Olander [8], consider the heat evolved on absorption. The approximation in the Treybal approach [10], as subsequently discussed by Treybal himself [11], is that all the heat evolved on absorption is taken up by the liquid stream, thus neglecting the rise in gas temperature. This is considered to be a conservative approximation for relatively dilute solutions. Kelly *et al.* [12], reviewing the work on packed bed adiabatic absorbers, felt serious design errors may result when the magnitude of the thermal effects associated with heats of absorption is large.

Factor and Grossman [13], following a similar approach to Olander [8], neglected the liquid-side heat transfer coefficient by assuming the interfacial temperature to be the same as the liquid bulk temperature. Stevens *et al.* [14] developed an effectiveness model based on similar assumptions and reported that it compared well with the results of Factor and Grossman's [13] finite difference model. However, when compared with experimental data, their model predictions at a few data points appear to underpredict the air outlet temperature by as much as 20%. Moreover, the outlet humidity also differed from the model predictions. Gandhidasan *et al.* [15] used a model which considered both heat and mass transfer resistances in the gas phase, but only mass transfer resistance in the liquid phase. In all the studies described above, the heat of dilution is either neglected altogether, or the evolved heat is accounted for entirely in the liquid stream. As pointed out, in those studies where comparison with experimental studies have been made, there is an underprediction by the model of the outlet air temperature and some variation in the outlet air humidity. Treybal [11] points out that neglecting the rise in gas temperature will result in a taller column and over-design.

In view of the above, it is believed that the heat of dilution (or absorption) should be accounted for in the gas stream also. Failure to do so will lead to air temperature predictions that can give rise to considerable error in computing the coefficient of performance (COP) of the liquid desiccant air-conditioning system. In this paper, the development of a model which accounts for the heat of absorption in both the liquid and gas streams is described. The effect of various operating parameters on the packed bed is examined and discussed.

Analysis

The model presented below is based on Treybal's treatment [10, 11]. It follows therefore that all the assumptions made in that model are also valid in this study, except for one very important difference, namely, that the heat evolved on absorption is accounted for in the enthalpy of both the streams.

Figure 1 shows a schematic representation of a differential section of the packed column. In this differential height, dZ, the gas is flowing upward at a rate G and the liquid downward at a rate L. The column operation is assumed to be adiabatic and the gas and liquid properties are assumed to be constant in the radial direction, that is, gradients occur only in the Z direction. The implication of the schematic representation is that the analysis is based on the simplifying assumption of slug flow with the packing surface providing the heat and mass transfer interface. The differential element in Fig. 1 has three distinct control volumes: I representing the gas, II representing the liquid, and III the entire cross-section. The following processes take place within this differential section:

(1) mass transfer—from the gas phase to the liquid;

(2) heat transfer—(a) sensible heat transfer from the gas phase to the liquid through the interface; (b) latent heat of condensation from the vapour to the liquid; (c) heat of dilution evolved at the interface.

Moisture transfer can be expressed (by convention) as

$$N_A M_A a_m \, dZ = -G \, dY \tag{1}$$

where

$$N_A = F_G \ln\left(\frac{1 - p_{Ai}/P_T}{1 - \bar{p}_{AG}/P_T}\right)$$

from which the air humidity gradient can be written as

$$\frac{dY}{dZ} = -\frac{F_G M_A a_m}{G} \ln\left(\frac{1 - p_{Ai}/P_T}{1 - \bar{p}_{AG}/P_T}\right) \tag{2}$$

The Ackermann correction for simultaneous heat and mass transfer is applied only to the gas-side

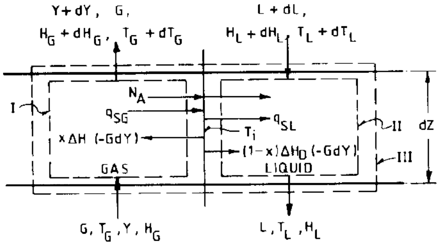

Fig. 1. Schematic representation of a differential section of the packed column.

resistance and is given by

$$h'_G a_H = \frac{N_A M_A C_A a_H}{1 - \exp(-N_A M_A C_A a_H / h_G a_H)} \tag{3}$$

Treybal [11] suggests that h_L is usually sufficiently large to make a correction unimportant for the liquid side. a_m and a_H, the transfer areas for mass transfer and heat transfer, are assumed to be equal, which is true if the packing is fully irrigated.

An enthalpy balance of the two streams applied over the entire differential volume (control volume III), under adiabatic conditions, yields

$$G\, dH_G = L\, dH_L + H_L\, dL \tag{4}$$

Similarly, an enthalpy balance is applied over the gas side alone in the differential section, to give

$$\begin{matrix}\text{enthalpy} \\ \text{out}\end{matrix} = \begin{matrix}\text{enthalpy} \\ \text{in}\end{matrix} - \begin{matrix}\text{sensible heat} \\ \text{transfer out}\end{matrix}$$

Here, 'enthalpy out' includes both the enthalpy of the outgoing air and the enthalpy of the vapour that is transferred to the liquid. Similarly, 'enthalpy in' includes the enthalpy of the incoming air and the fraction of the heat of dilution which enters the gas phase from the interface. Applying the enthalpy balance after introducing

$$dH_G = C_S\, dT_G + (\lambda_0 + C_A T_G)\, dY$$

where the humid heat C_S is given by

$$C_S = C_B + Y C_A$$

the following expression is obtained:

$$\frac{dT_G}{dZ} = -\frac{x\, \Delta H_D}{C_S}\frac{dY}{dZ} - \frac{h'_G a_H (T_G - T_i)}{G C_S} \tag{5}$$

Similarly, a liquid-side enthalpy balance (control volume II) yields

$$\begin{matrix}\text{enthalpy} \\ \text{out}\end{matrix} = \begin{matrix}\text{enthalpy} \\ \text{in}\end{matrix} + \begin{matrix}\text{sensible heat} \\ \text{transfer in}\end{matrix}$$

Here, 'enthalpy in' includes the enthalpy of the incoming liquid and the enthalpy of the transferred vapour, now a liquid, and the heat of dilution which enters the liquid from the interface. Applying the enthalpy balance and simplifying yields

$$T_i = \frac{G \dfrac{dY}{dZ}(\Delta H_D - \lambda_0 - C_A T_G) + h_L a_H T_L + h_G a_H T_G}{h_G a_H + h_L a_H - G \dfrac{dY}{dZ} C_{AL}} \tag{6}$$

Computational procedure

The equations developed above can be used both for rating and sizing the dehumidifier.

Rating problem

To determine the output condition of the air stream given the column height, air and liquid inlet conditions, the following procedure may be followed. The entire column is discretized into small sections of height ΔZ and the bottommost stage or segment is considered first. The outlet temperature and concentration of the liquid are assumed. Using eqns. (2) and (6) the interfacial temperature is determined. A trial and error procedure is required here as the vapour pressure of the liquid in eqn. (2) depends on the temperature. Equation (5) is then used to obtain the air temperature profile. From this the temperature of the air leaving the segment is determined. The outlet humidity of the air from this segment can be calculated using eqn. (2). Overall material and enthalpy balances over the discrete segment yield the liquid inlet flow rate, concentration and temperature.

The liquid inlet temperature so calculated will be the outlet condition for the next stage. This procedure is continued till the top of the column is reached and the given liquid inlet conditions are checked with the calculated values. If they do not match, new liquid outlet conditions have to be assumed and the iteration continued till convergence is achieved.

Sizing problem

To determine the column height required to achieve specified air outlet humidity conditions, the following procedure may be followed. From a material balance over the entire column, the outlet liquid flow rate and concentration are determined for a given air outlet humidity condition. The column is then divided into a finite number of segments of height ΔZ, which is as yet unknown. By assuming the outlet temperature of the liquid at the bottom of the column and using eqns. (2) and (6) the interfacial temperature is determined as before. Equation (5) gives the temperature gradient within this segment. From the humidity gradient determined by eqn. (2), ΔZ is determined and hence the air outlet temperature of this segment. The overall material and energy balance over this segment gives the liquid inlet conditions. This procedure is followed till the top of the column is reached and the given liquid inlet temperature is checked with the calculated value. If they do not match, a new liquid inlet temperature has to be assumed and the iteration continued till convergence is achieved.

Results and discussion

Numerical computations were performed to obtain the performance characteristics of a liquid desiccant packed bed dehumidifier using the algorithms described above. The liquid desiccant used in the computations was lithium bromide solution. The packing used in the column was 3/4 in. (19 mm) Intalox saddles. The mass transfer coefficients used were obtained from the correlation by Shulman *et al.* [16]. This correlation, while specific to Berl saddles, can also be used for Intalox saddles [12]. The heat transfer coefficients used were obtained from the mass transfer coefficients using the analogy between heat and mass transfer following the procedure described by Treybal [11]. The simulations were made for different air and liquid flow rates, inlet fluid conditions for a particular column height (rating problem) and for a particular air outlet humidity condition (sizing problem). Physical properties of the lithium bromide solution were obtained from the data of Patterson and Perez-Blanco [17].

Rating problem

Air temperature profiles within the column

For a given height of column (1 m), that is, the rating problem, the temperature profile of the air within the column is shown in Fig. 2 for L/G values of 0.5, 1.0 and 1.2. While Fig. 2(a) refers to air and liquid inlet temperatures of 25 °C each, Fig. 2(b) is for 33 and 25 °C and Fig. 2(c) for 30 and 40 °C, respectively. For each case, x values of 0.0, 0.5 and 1.0 are shown, that is, the fraction of the heat of dilution accounted for by the air stream.

In liquid desiccant heat pump systems the outlet air temperature of the dehumidifier is an important design factor. It is only on the basis of the outlet air temperature that sensible heat exchangers (coolers) or evaporative coolers can be designed. The general assumption that the heat of dilution is fully absorbed by the liquid stream obviously results in underprediction of the air outlet temperature.

At low L/G ratios the temperature profile of the air in the column exhibits a peak near the top of the column, resulting in greater dehumidification there. Proceeding down the column, as the liquid temperature increases, dehumidification will decrease (the vapour pressure of the liquid being greater, the concentration driving force for mass transfer is smaller). Thus, when the air temperature profile peak occurs near the top of the column, the dehumidification of the air is less. As L/G increases, the peak in the air temperature profile is shifted downward, that is, towards the lower part of the column. As this implies that the air is getting cooled in the region above this maximum, it means that the maximum dehumidification occurs in this region and above it the air loses its heat to the cooler liquid entering from the top of the column. The model of Stevens *et al.* [14] underpredicts the air outlet temperature by as much as 20%, that is, about 3–6 °C for their system. Figures 2(a) and 2(b) show that even at a value of $x = 1.0$, when all the heat evolved at the interface is accounted for by the gas, the additional temperature rise is only about 2–3 °C for the present system. This indicates that the value of x in the system considered by Stevens *et al.* must be fairly high. Furthermore, it is interesting to note that the difference between the temperature profiles for different x values is maximum where the peak of the air temperature profile occurs. Therefore a higher amount of dehumidification should occur at higher x values.

Liquid temperature profile within the column

Figure 3 shows the temperature profile of the liquid within the column for L/G values of 0.5, 1.0 and 1.2; x values of 0, 0.5 and 1.0, as in Fig. 2, were used. The Figure shows that at moderate L/G values the liquid is heated gradually as it progresses down the column (decreasing Z). On the other hand, at low L/G values the liquid temperature appears to increase sharply initially and then to level off. At low L/G values the quantity of liquid is relatively less and therefore the energy gained by the liquid will be seen as an increase in temperature only, whereas at higher L/G ratios, the quantity of liquid being larger, the gain in temperature will be less apparent. The value of x does not seem to have any effect at higher L/G values near the top of the column, irrespective of the fluid inlet conditions. This may

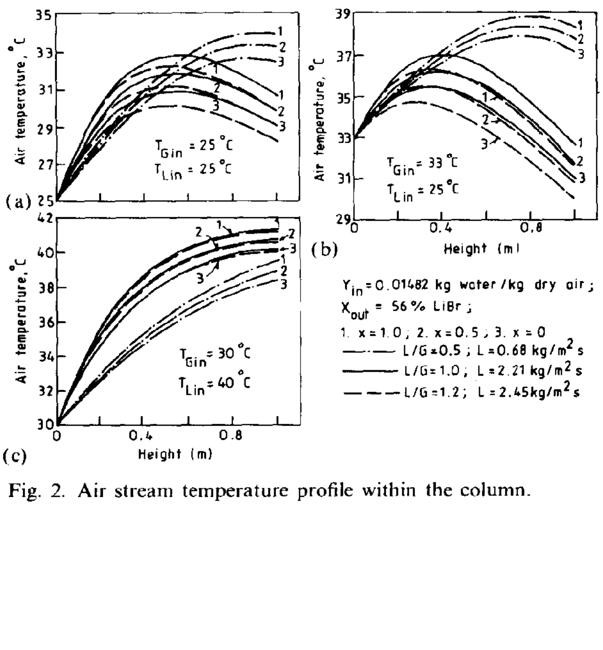

Fig. 2. Air stream temperature profile within the column.

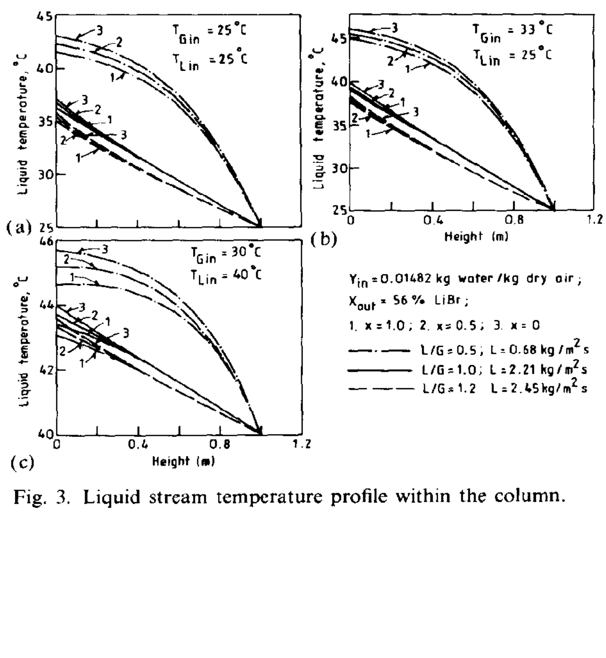

Fig. 3. Liquid stream temperature profile within the column.

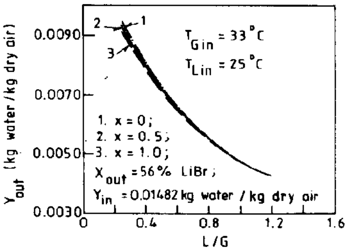

Fig. 4. Effect of L/G ratios on outlet humidity.

be explained as follows. At the top of the column the heat gained by the liquid is mainly from the latent heat of the vapour transferring from the air to the liquid phase. On the other hand, at the bottom of the column x has a marginal effect which may be due to the sensible heat transfer from the liquid to the air.

Outlet humidity

Figure 4 shows the effect of L/G ratios on the outlet humidity of the air for various inlet temperatures of the air and liquid. The effect of x seems to be minimal. However, as the difference in temperatures of the two inlet streams increases, the x values seem to have a greater influence.

As L/G ratios increase, the outlet humidity decreases. But the choice of an L/G ratio will also depend on the regeneration energy requirements. At high L/G ratios the outlet liquid temperatures fall (see Fig. 3), resulting in higher regeneration energy requirements. Therefore, as Factor and Grossman [13] have suggested, the optimal L/G ratio is likely to be around 0.6. Furthermore, it may be noted that at $x = 0$ and $T_i = T_L$ the present model collapses to the Factor and Grossman model.

Sizing problem

Effect of gas flow rate on packing height

Figure 5 shows that as G increases, the effect of x also increases, that is, as x increases, the required packing depth decreases. This is as it should be, since, for a given L, as G increases, L/G decreases. As explained earlier when Fig. 2 was described, the effect of x increases with lower L/G ratios.

Effect of liquid flow rate on packing height

Figure 5 shows that the packing height required decreases with increase in the liquid flow rate, as more liquid desiccant is available to dry the air. x has a moderate effect on the required packing height at low liquid flow rates and little effect at higher liquid flow rates. For a given G, as L increases, L/G increases. As explained earlier when Fig. 3 was described, the effect of x decreases with higher L/G ratios.

Effect of air inlet temperature on packing height

Figure 6 shows the effect of the air inlet temperature on the packing height. In this Figure, the importance of considering x, the fraction of the heat of dilution absorbed by the gas phase, is brought out dramatically. As the value of x increases, the packing height required for a particular dehumidification duty is reduced substantially. Furthermore, from the Figure it may be seen that as the air inlet temperature increases, the packing height required also increases. This is because at higher air inlet temperatures both sensible and latent heat will be gained by the liquid, thereby increasing its vapour pressure and decreasing the potential for mass transfer. Therefore, a taller column will be required. This result is, however, contrary to the conclusions

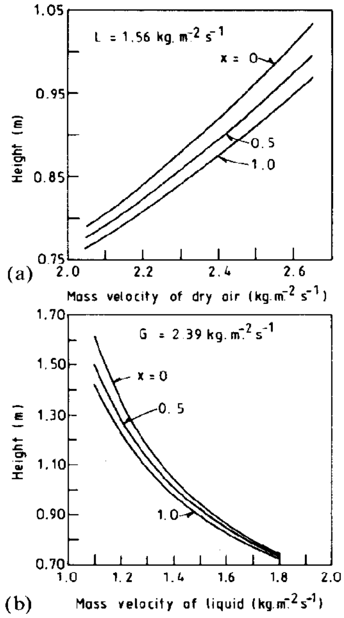

Fig. 5. Effect of flow rate on packing height. ($Y_{in} = 0.022$ kg water/kg dry air; $Y_{out} = 0.011$ kg water/kg dry air; $X_{in} = 55\%$ LiBr; $T_{Gin} = T_{Lin} = 30\,°C$.)

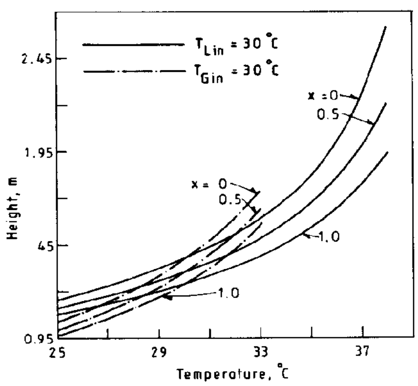

Fig. 6. Effect of fluid inlet temperatures on packing height. ($L = 1.158$ kg/m^2 s; $G = 2.316$ kg/m^2 s; $Y_{in} = 0.022$ kg water/kg dry air; $Y_{out} = 0.011$ kg water/kg dry air; $X_{in} = 55\%$ LiBr.)

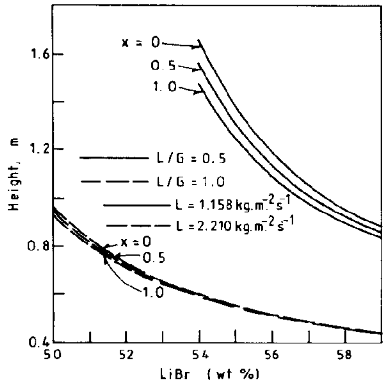

Fig. 7. Effect of liquid inlet concentration on packing height. ($Y_{in} = 0.022$ kg water/kg dry air; $Y_{out} = 0.011$ kg water/kg dry air; $T_{Gin} = T_{Lin} = 30$ °C.)

of Gandhidasan *et al.* [15] who said air inlet temperature has no effect on packing height. This is probably because the L/G values considered by them were large. However, as pointed out earlier in this paper and also by Factor and Grossman [13], the optimal L/G ratio should be close to around 0.6.

Effect of liquid inlet temperature on packing height
Figure 6 also shows the effect of the liquid inlet temperature on the packing height, namely, that the packing height required increases with liquid inlet temperature. This is, of course, an obvious result, since as the liquid temperature increases, its vapour pressure will also increase, resulting in a lower mass transfer driving force.

Effect of liquid inlet concentration
Figure 7 shows the effect of the liquid inlet concentration on the packing height. It can be seen that the packing height decreases with liquid concentration, which is explained by the fact that as the liquid concentration increases, the drying capacity of the liquid increases. The point to be noted here, however, is that at higher L/G values (around unity) the effect of x on the packing height is negligible, while at L/G ratios of around 0.5 the effect is significant.

Conclusions

A theoretical analysis has been performed to study the operation of a packed bed liquid desiccant dehumidifier. The model developed includes the effect of x, the fraction of the heat of dilution that is accounted for in the gas stream. The effect of x under various inlet conditions and flow rates has been studied for both the rating and sizing problem of the packed bed dehumidifier. It has been observed that x is more important at lower L/G ratios (<1.0). Moreover, as Factor and Grossman [13] have pointed out, optimal values of L/G ratios are around 0.5–0.7, so the effect of x in this region cannot be ignored. Furthermore, Stevens *et al.* [14] found their model underpredicts air outlet temperatures by up to 20% for some data points. This indicates that neglecting x will lead to design error. Treybal [11] also has pointed out that the column height calculated will be greater than actually required if the rise in air stream temperature is neglected.

Gandhidasan *et al.* [15], analysing the packed bed dehumidifier, concluded that the air stream inlet temperature does not affect the height of packing required. While this may be true for large L/G values, this study does not indicate such a conclusion for low L/G values.

The model presented here is general and can be used for any gas absorption operation which has large heat effects. In fact, the effect of x will be substantial in those cases where heat effects at the interface are much greater than in the present system. It may also be noted that at $x = 0$ and at $T_i = T_L$ the present model collapses to that of Factor and Grossman [13].

Nomenclature

a_H	specific interfacial area for heat transfer, m^2/m^3
a_m	specific interfacial area for mass transfer, m^2/m^3
C	heat capacity, J/kg °C
C_S	humid heat, J/kg °C
F	mass transfer coefficient, kmol/m^2 s
G	superficial mass velocity of dry air, kg/m^2 s
H	enthalpy, J/kg
ΔH_D	heat of dilution, J/kg
h	convective heat transfer coefficient, W/m^2 °C
h'	convective heat transfer coefficient corrected for mass transfer, W/m^2 °C
L	superficial mass velocity of liquid, kg/m^2 s
M	molecular weight, kg/kg mol
N	mass flux, kg mol/m^2 s
P_T	total pressure, Pa
p	vapour pressure, Pa
$\bar{p}$	partial pressure, Pa
q_S	sensible heat flux, W/m^2
T	temperature, °C
X	percentage concentration of liquid desiccant
x	fraction of heat of dilution accounted for in air stream
Y	absolute humidity, kg water vapour/kg dry air
Z	height of packed bed, m
λ_0	latent heat of vaporization, J/kg

Subscripts

A	water vapour
AL	water
B	air
G	gas
i	interface
L	liquid

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